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WET CLUTCH SHIFTING PROCESS WITH ADAPTIVE FUZZY CONTROL

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The velocity and torque of clutch plates considerably changed during the inertia phase of shift, which leads to fluctuant output torque and unstable performance of the wet clutch. The accurate shift control is thus necessary to maintain the quality of gear shifting. Based on the mathematic model of the power upshift for wet clutch, the shifting dynamic model with four phases is built in multi-discipline simulation platform. A fuzzy adaptive controller is designed according to the dynamic model, in order to implement the closed loop control with tracking control theory for shifting. The controller created fluctuation towards the PID parameters, and it provided precise accurate quantity to be used for optimizing oil pressure force on clutch plates through accurate quantity. Comparing with the conventional PID controller, the simulation results showed that the maximum error for tracking speed of clutch was reduced by 0.062 rad/s via using the fuzzy controller with optimized parameters. Moreover, the hardware-in-the-loop simulation was conducted as well in this paper. The results indicated that margin of error for tracking speed was limited in 0.146 and the range of jerk was between -5.3 m/s^3 *and 9.57* m/s^3.

Keywords: shift controller; fuzzy adaptive resonance theory; wet clutch; Hardware-in-the-loop

1. Introduction

The wet clutch is usually utilized to change gear in an automatic transmission system. The planetary gear configurations are altered following system dynamics rules along with disengaging or coupling different clutches. Due to the smooth transfer torque, the wet clutch has been successfully used in most of the automatic transmission for several decades. Comparing with the dry clutch, the torque capacity of wet clutch is larger. As a result, the heavy duty truck and agricultural machine often utilizes a wet clutch in their automatic transmission. Working in harsh condition, it may lead to clutch failure easily in a short time. The desired shifting process would use the shortest time without any gear shift shock. However, the clutch dynamic model is highly nonlinear considering time-

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varying characteristic of the system. Therefore, the automobile industry has recently made significant investments in the shifting control of wet clutch [1-3]. Several researchers have derived full physical models for wet clutches[4-7]. These have been applied to the design of feedback controllers [8-9] and feedforward controllers [10]. This requires a large effort to get accurate models, typically consisting of white box modelling in combination with experimental parameter estimation. Complex models also complicate control design and often result in complicated control structures, unless simplified models are use, as in papers [4][8]. Using separate controllers for the shifting process can make things easier as well, as it is acceptable to develop a model for each phase separately. Advanced intelligent control theory has been applied to the torque phase, and a cost function for the optimal predictive model control needs to be well established before applying to off-line model of transmission system [11-13]. For the complicated intelligent control method, it has become a great challenge in real-time model. Besides, the output torque of transmission is hard to monitor during shifting unless building state observer. The controller with observer has achieved better results by an observer with high accuracy.

The nonlinear dynamic function of system would change when the on-going clutch start to engage. It’s necessary to establish the nonlinear clutch model in different phases for smooth gear shift. In this paper, the dynamic model including different shifting phases has been built with considering the characteristics of shifting process. Because of the huge torque fluctuation in initial phase, the fuzzy controller with adaptive algorithms was designed for optimizing shifting quality. Based on the simulation model, the hardware-in-the loop test would be used to verify effects of the controller.

2. Shifting process of wet clutch

2.1 Dynamic model

A wet clutch system consists of a multi-layer assembly of steel and friction plates. The wet clutch plates are submerged in transmission oil that lubricates the space between the friction and separator steel plates. When the clutch works in disengagement, the friction and steel plates rotate at different speeds (rpm). In the clutch engagement, the friction plates are pushed together and the oil is squeezed out of the gap and the porous friction material. Generally, the gear shift derives by the on-coming clutch and off-going clutch. The wet clutch model would be built according to the actual shifting process. This paper developed wet clutches process in a 6-speed automatic transmission. Assuming the system components as rigid bodies, the dynamic model of shifting has been simplified by following rules. The axial displacement would be neglected and the power loss resulted from mechanical efficiency be converted to driving resistance. The dynamic model represented in Fig.1. When the first gear changed to second gear, the engaging C1clutch would change to disengage while the B1clutch was from disengaging to engaging.

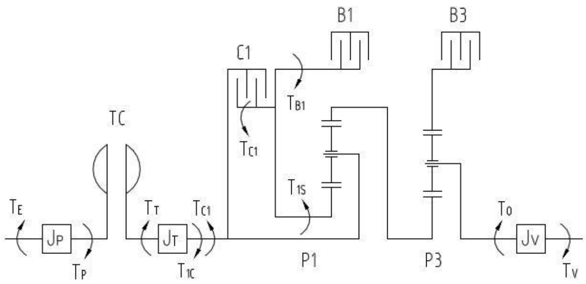


Fig. 1. The dynamic powertrain model for the first gear including 1-2 up shift (C1– clutch; B1, B3 – breaks; P1, P3 – planetary system; TC – torque converter; TE, TP, TT, T1C, TC1, TB1, T1S, TO, TV– torque)

(1) First gear dynamics

During the first gear phase, the C1 clutch is fully engaged and the B1clutch is disengaged. In this case, the speed of the sun gear and carries in P1 planetary system is same as the speed of ring gear. Applying to the dynamic equations describing the first gear, we can write the dynamic equation of the primary gear set during the first gear for the 6-speed transmission as follows.

$$\begin{cases} T_{C1} = \dfrac{i_{1p1}T_T}{p_1} = \dfrac{i_{1p1}KT_E}{p_1} \\ T_{B1} = 0 \\ \Delta\omega_{C1} = 0 \\ \Delta\omega_{B1} = \omega_T \\ T_O = i_1 T_T = i_1 K T_E \end{cases} \tag{1}$$

As mentioned, Tc1 and TB1 are the torques from the C1 and B1 clutches respectively, TE represents the transmission input torque, TT refers to the turbine torque calculated from the torque converter model, To represents the transmission output shaft torque, ΔωC1 represents the relative speed of C1 clutch, ΔωB1 represents the relative speed of B1 clutch, ωT represents the speed of turbine, i1 is the first gear ratio, i1p1 is the gear ratio of planetary system in the first gear, K is the torque ratio of converter, and p1 refers to the characteristic parameter of planetary P1.

(2) 1-2 up shift- Torque phase

When the clutches start to move the shifting turns to torque phase. The oil pressure on the B1 clutch plates begins to rise, therefore the B1 clutch transfers part of torque from engine which plates rub each other. Meanwhile the oil pressure on the B1 clutch plates declines but there is no sliding friction between its plates.

$$\begin{cases} T_{C1} = \dfrac{1}{p_1}\left(T_T - (p_1+1)\mu k_{B1} p_{B1} - J_T \dot{\omega}_T\right) \\ T_{B1} = \mu k_{B1} p_{B1} \\ \Delta\omega_{C1} = 0 \\ \Delta\omega_{B1} = \omega_{1S} = \omega_T \\ T_O = i_{3p}(T_T - \mu k_{B1} p_{B1} - J_T \dot{\omega}_T) \end{cases} \tag{2}$$

Where, μ represents the friction coefficient of friction plates, kB1 refers the proportionality coefficient based on the structure of B1clutch, pB1 represents the oil pressure on B1 clutch plates, JT refers the sum inertia of turbine, C1 clutch and carrier of P1planetary, ω1s represents the speed of sun gear of P1 planetary, and i3p is the gear ratio of P3 planetary system in the second gear.

(3) 1-2 up shift- Inertia phase

In the inertia phase both C1 clutch and B1 clutch work in slip stage.

$$\begin{cases} T_{C1} = \mu k_{C1} p_{C1} \\ T_{B1} = \mu k_{B1} p_{B1} \\ \Delta\omega_{C1} = \omega_{1C} - \omega_{1S} = p_1 i_{3p} \omega_O - p_1 \omega_T \\ \Delta\omega_{B1} = \omega_{1S} = \left(p_1+1\right)\omega_T - p_1 i_{3p} \omega_O \\ T_O = i_{3p} p_1 \mu (k_{B1} p_{B1} - k_{C1} p_{C1}) \end{cases} \tag{3}$$

As mentioned, kC1 refers the proportionality coefficient based on the structure of C1clutch, pC1 represents the oil pressure on C1 clutch plates, ω_{1c} represents the speed of carrier of P1 planetary, and ω_o represents the output shaft speed.

(4) Second gear dynamics

At the end of shifting phases, states of C1 and B1clutch are opposite comparing with the first gear phase. The torque form engine is transferred by clutch B1, so the gear ratio turns to the second gear. As mentioned the dynamic model of shifting should be built based on different phases.

$$\begin{cases} T_{C1} = 0 \\ T_{B1} = \dfrac{T_T}{1+p_1} = \dfrac{KT_E}{1+p_1} \\ \Delta\omega_{C1} = \omega_T \\ \Delta\omega_{B1} = 0 \\ T_O = i_2 T_T = i_2 K T_E \end{cases} \tag{4}$$

Where, the i_2 refers to the gear ration of the second gear.

2.2 Clutch pressure control in shifting

For all plate clutches, the model is a simple piston-cylinder model. The dynamics of clutch pressure depend on the difference between flow rates into and out of the clutch and accumulator cavity. However, before the clutch pressure starts to increase, the initial cavity has to be filled first. In the shifting process, the pressure needs to be adjusted in the torque phase and inertia phase. Therefore, these two phases are divided into four states, the filling state, the torque state, the inertia state and uplift state [14]. In the filling state, the C1 and B1 clutches begin to change working condition at the same time. Actually, the pressure of C1 clutch should be at a high value to assure the torque capacity in the first gear state. In this case, the pressure of C1 clutch decreases to transfer the torque considering output shaft load. The pressure of B1 clutch increases simultaneously to eliminate displacement between piston and plates, plates each other.

In torque state, the clutch system is in the 2-degree of freedom during shifting process. The output shaft torque becomes small when the friction torque of C1 clutch increases continuously. Since the friction torque of C1 clutch drop to 0 at the end of torque state, the torque would be maintained in the inertial state. Therefore, the torque and pressure of B1 clutch should be followed as below equations.

$$\begin{cases} T_{B1} = T_{B1}^{*} \\ p_{B1} = p_{B1}^{*} = \dfrac{T_{B1}^{*}}{\mu k_{B1}} \end{cases} \tag{5}$$

For the jerk especially comes up in inertial state, we develop the pressure control in this paper. At the end of inertial state, the driving plates and driven plates of B1 clutch has same speed. So the torque function was shown as below.

$$\begin{cases} T_{B1} = \dfrac{T_O}{i_{3p} p_1} = \dfrac{i_2 T_T}{i_{3p} p_1} \\ p_{B1} = \dfrac{i_2 T_T}{i_{3p} p_1 \mu k_{B1}} \end{cases} \tag{6}$$

When the ratio between the output shaft speed of transmission and the speed of turbine is equal to the second gear ratio, the inertial state is end off and the B1 clutch has fully engaged. However, the torque fluctuated greatly which may lead to that the plates come away. As a result, the pressure of B1 clutch should be increased to a high value to avoid the fiction off. The ideal pressure for two clutches was shown as below figur2.

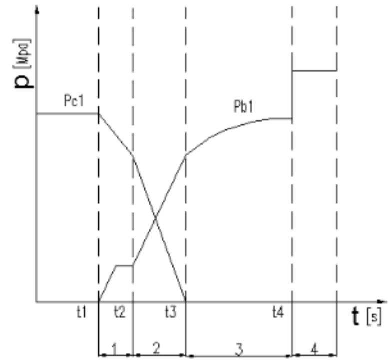


Fig. 2 Ideal pressure force curve

3. Precise tracking controller

3.1 Fuzzy PID adaptive controller

The previous parts describe in detail the importance of calibrating proper pressure proles on the relevant friction elements for good shift quality. It also demonstrates the difficulty in finding such a satisfactory pressure profile for a shift. In shifting process, there are two evaluating indicators, the jerk of vehicle and sliding friction work of plates. These two values should be small in ideal shifting process while it is trade-off between each other in actual shifting process. Therefore, the tracking control is a compromise method toward two indicators which is simplified and simple to implement.

This paper presents the application of tracking control methods to shift dynamics, which eliminates the need for calibrating pressure profile and generates a satisfactory pressure profile based on a formulation of a shift objective. The only input variable is the current of solenoid valve assuming that the throttle of engine depends on the load. So the current of solenoid would influence the speed of plates, the reasonable target speed would guarantee the shifting time suitable and decreasing the shifting shock, which avoids the friction damage by heat built up. It requires the speed of driving plate and driven plates to be the same in the synchronize moment. As a result, the tracking target profile in inertia state should follow some rules which are as below. Since the short shifting time would generate small fiction work, the $\left(t_f - t_0\right)$ need less than the specified shifting time. The speed of on-coming clutch should satisfy the condition without shock, which implies the slope of curve is zero at the end of moment, t_f .To avoid the saturated control variable, the slope of tracking target curve should be close to zero.

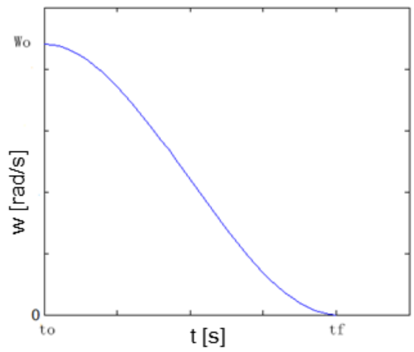


Fig.3 Reference trajectory for speed of clutch plates

To realize the target trajectory in the figure above, we use a cubic polynomial to fit the speed profile. The equations of profile are shown as below.

$$x_{1r} = \frac{2\omega_0}{\left(t_f - t_0\right)^3}\left(t - t_0\right)^3 - \frac{3\omega_0}{\left(t_f - t_0\right)^2}\left(t - t_0\right)^2 + \omega_0 \tag{7}$$

$$\dot{x}_{1r} = \frac{6\omega_0}{\left(t_f - t_0\right)^3}\left(t - t_0\right)^2 - \frac{6\omega_0}{\left(t_f - t_0\right)^2}\left(t - t_0\right) \tag{8}$$

Where, the t0 refers to the start moment of inertial state, the t_f represents the moment of clutch fully engaged, x_{1r} represents the target speed of friction plates during inertial state, and ω_0 refers to speed of friction plate at the start moment in inertial state.

Because the controller performance depends on the structure, the structure of controller is necessary to be designed before building the fuzzy controller. The structural design is to choose the input and output variables of the controller. And the fuzzy controller can be classified into three categories: one-dimensional, two-dimensional and multidimensional controller. The two-dimensional controller has two inputs and only one output, the fuzzy decision rules of controller were represented as follow.

Rn: *if* X_1 *is* A_1n *and* X_2 *is* A_2n, *then* Y *is* Bn

Where A1n, A2n, Bn refers to fuzzy subset in universes, X1 represents the error, and X2 represents the change rate of error. Usually, the X1 and X2 are used for the inputs for 2- dimensional controller which output is calculated considering the error and change rate of error. Since the results of 2- dimensional controller are better than the 1- dimensional controller and it is easy to carry out in application, the 2- dimensional controller has been wild used in many engineering fields. Comparing with mentioned controller, the structure of 2- dimensional controller has been applied in this paper, and the speed error and change rate of error of B1 clutch were chose as the inputs.

The fuzzy rules have become the central to design the controller in tracking control. It consists of three parts, choosing suitable fuzzy language variables, deciding membership function and building fuzzy rules. The amount of fuzzy variables should be moderate and located in universes equably during choosing fuzzy variables. In defining fuzzy subsets, the max value of membership function for variables is not too small, and it would decrease the sensitivity of system. Assuming, the basic universe of et is $[-e_u, e_u]$, and the universe is set as E=$[-6, 6]$ and the quantization factor is $k_e = 6/e_u$. Following the same rules, the universe of de(t) represents $EC = [-6, 6]$ and the quantization factor is $k_e = 6/ec_u$. The membership function of the input variable were shown in the below figure.

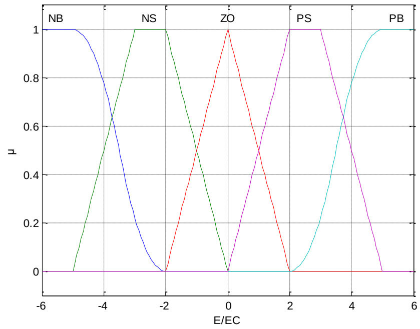


Fig.4 Subjection function distribution of input variable E/EC

Based on the subject function, the triangle-shape grade of membership function has been used with small error which makes the control sensitive. While the Trapezoid membership function was used in medium error, the sigmoid membership function has been applied for large error to improve system stability. Meanwhile, the fuzzy language almost covers the universe which the membership value is large than 0.5 at the points of intersection among these mentioned membership function. The ΔKp is between -3 and 3, which is the increment of Kp, the parameter of PID control. The five language states were presented by NB, NS, ZO, PS and PB. In this paper the membership functions of Ki and Kd are the same as the Kp. The control variables are fuzzy value deducing by fuzzy rules, so they are solved ambiguity and change to accurate quantity before applying to control actuators. The weighted average method was applied to solve ambiguity, and it is shown as below,

$$Z^* = \frac{\sum_{i=1}^{n} \mu_{c_i}(w_i) w_i}{\sum_{i=1}^{n} \mu_{c_i}(w_i)} \tag{9}$$

After solving ambiguity, the increment of parameter has been obtained by control variables and their scale factor.

3.2 Corresponding simulation for adaptive controller

The vehicle dynamic model was built by MapleSim which consist of engine model, automatic transmission model and vehicle longitudinal dynamic model. The fuzzy controller established in Matlab/Simulink and the diagram of shift controller includes automatic transmission modules, data processing module and fuzzy tracking controller. In corresponding simulation the automatic transmission model was seen as controller object according to the model in MapleSim. The inputs of controller module are throttle angle, the pressure profile of clutches, while the outputs are the speed and torque of engine, the speed of turbine and the output shaft speed and velocity of vehicle.

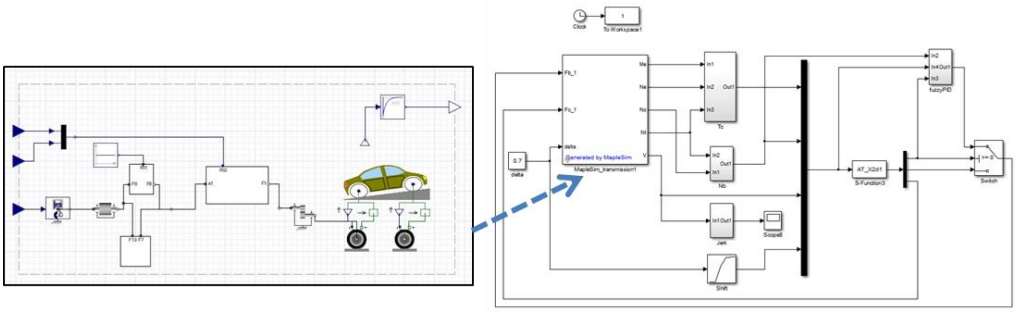


Fig.5 The Corresponding simulation model

In order to verify the adaptive performance of control police, the results are necessary to be analysed in different Internal and external variables. The control variables have been developed in different working conditions towards the external environmental variables.

The variation of the vehicle mass and the road slope results in the change of the vehicle dynamics. Therefore, we consider some major uncertainty and disturbance terms, such as the variations of road grade α and vehicle mass m. The corresponding simulation results in Fig.6 are given to indicate the fuzzy controller performance. Before 1.2 second the pressures of B1and C1clutches remain unchanged and the gear keeps in the first gear. As can be seen from Fig.6b that the gear changes at 1.2s when the pressure of the off-going clutch begins to decrease and velocity of vehicle is around 6.53Km/h. The rapid filling state is between 1.2 to 1.38s, and the capacity torque of C1 clutch decreases to adapting current torque at the same time. The pressure of B1 clutch maintains zero in this state while neglecting effect of the hydraulic system. During the torque state, between1.38s and 1.58s, the pressure of C1 decline to zero as a linear trend while the pressure of B1clutch rises to the target oil pressure linearly. In the inertial state the pressure of B1 clutch begin to change following the adaptive fuzzy control rules, which increases firstly then decline. After the inertial state, the pressure of B1 clutch rise to a high value to avoiding sliding by the torque shock. The shifting process lasts around 0.98s following the trend of profile in Fig.2.

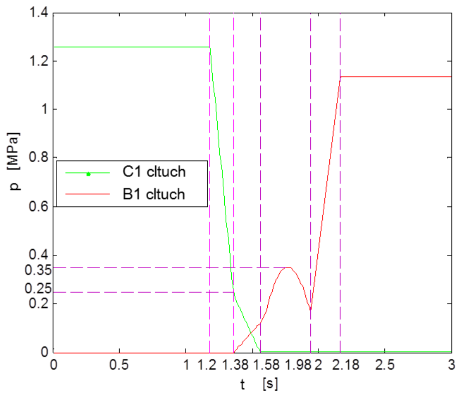


Fig.6a Pressure force curve of power upshift

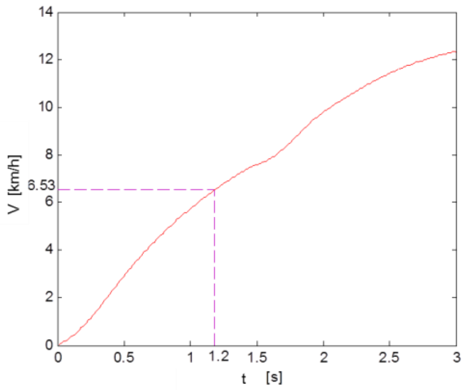
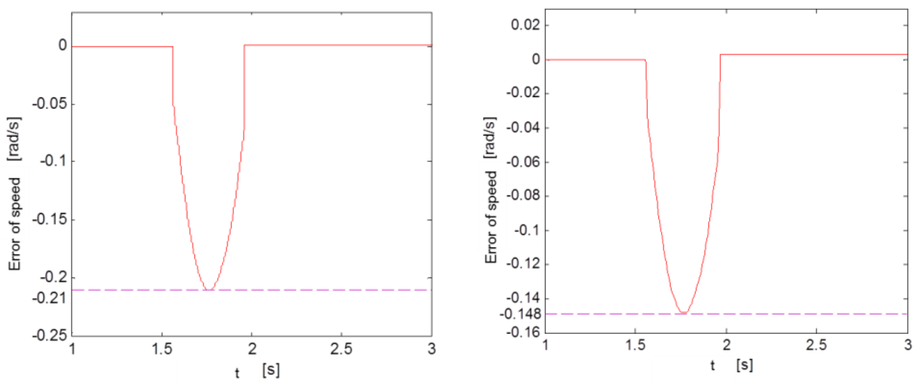


Fig.6b Velocity curve of power upshift

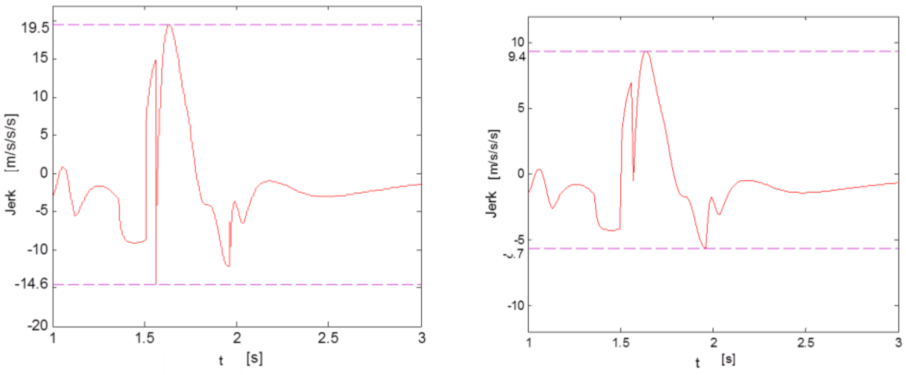
The Kp parameter of PID was tuned by the fuzzy controller after the inertial state of shifting process. The inertial value of Kp is -1, and the controller adjusts the value according to the fuzzy rules. As a result, the Kp decline to -1.045 at the end of inertial state. Comparing with normal PID method, the fuzzy PID method has advantages that tuning the control variables based on the error of tracking profile. The error of tracking profiles in normal PID and fuzzy PID method are shown in Fig.7a and Fig.7b respectively. Comparing to the results, the max value of error of tracking profile declines by 0.062rad/s after tuning parameters of PID.



a normal PID b fuzzy PID

Fig.7 Tracking error for velocity of brake plates during Inertial phase

Before tuning the parameters of PID, the jerk of shifting process is represented in Fig.8a. The max value of jerk peaks at the gap between torque state and inertial state, which induces by the output shaft torque varying greatly. The value is between -14.6m/s3 to 19.5m/s3 that exceeds the human perception threshold, 10m/s3. By tuning the Kp, the range of jerk is around -5.7m/s3 and 9.4m/s3, hence, the adaptive fuzzy method is better than the normal PID.



a normal PID b fuzzy PID

Fig.8 Jerk during the power upshift

4. Hardware-in-the-loop test

4.1 Building experimental platform

Hardware-in-the-loop test plays an important role in the 'V' development process in designing transmission control unit (TCU) at the late stage, which could be used to test the main function of TCU. During the test the TCU output real signals toward for the simulation model, hence the test save the cost in development process. Meanwhile, it makes easily for the test in extreme condition and failure injection test especially comparing with bench test. The hardware-in-the-loop test has been developed in this paper using the DFIII controller module and the DeskHIL simulation platform produced by Hengrun technology cooperation. The adaptive control algorithm has been written into the controller firstly, and the simulation model built in front part would be played in the DeskHIL. The model and controller communicate via the CAN bus, and the state of transmission model send to controller by the simulation platform and the shifting signal are outputted by the controller.

Table 1

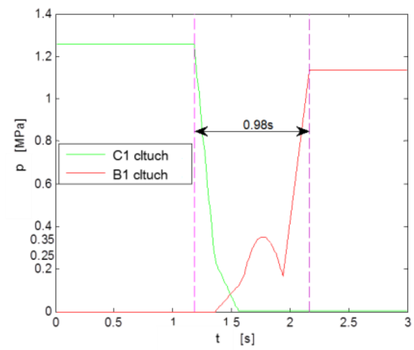
The fuzzy rules of discrete ΔKp

Mesh number	-6	3	0	-3	-6
-6	-3	-2	-2	-1	0
-3	-2	-1	0	1	2
0	-1	-1	0	1	2
3	-1	0	1	2	3
6	0	2	3	3	3

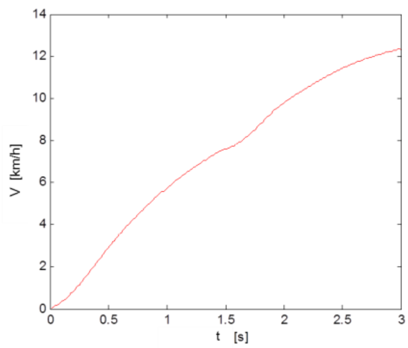
To improve the efficiency of controller, the membership functions for input and output variables would be discretized by writing the fuzzy adaptive method into the control unit. Then, the variables would be output after calculating according to the fuzzy rules. The parameters of PID, Kp, is discretized by fuzzy rules which is shown in the following table, and the other parameters are similar with Kp.

The hardware-in-the-loop test observers by HiGaleView in the host computer and the parameters are easily to be revised through the software. Besides, the output variables in the test could be checked by graphics window.

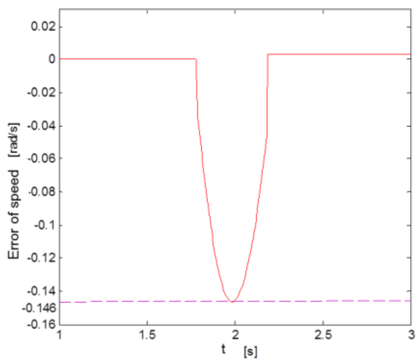
The simulation conditions are set as: the mass of vehicle is 5500kg, the percentage of throttle angle is 70% and the slope is 0.3. The hardware-in-the-loop test is conducted in the acceleration scenario. The simulation results are represented in Fig.9 that the pressure profile is shown in Fig.9a and velocity of vehicle is shown in Fig.9b. Based on the results in on-line test, the trend of velocity and shifting time is same as the corresponding simulation results.



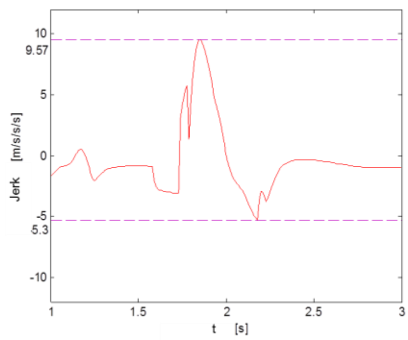
a The profile of pressure



b The velocity



c Tracking error by fuzzy PID



d The jerk by fuzzy PID

Fig.9 Simulation result of Hardware-in-the-loop

The tracking error using adaptive fuzzy controller is shown in Fig.9c which is less than 0.146. While the range of the jerk during shifting is between-

5.3m/s3 and 9.57m/s3, the deviation with off-line results is only 2%. Hence, the control strategy has effective and real-time performance based on the results of hardware-in-the-loop test. In conclusion, the adaptive fuzzy control strategy is a concise and effective way to improve the shifting quality and the controller provides the pressure profile of clutches to track the ideal speed profiles for decreasing the jerk during shifting.

5. Conclusions

1) The mathematic model of indicators for shifting quality has been built and the dynamic model of shifting for automatic transmission has been established in this paper. Using the simulation software, the powertrain system model was built considering the shifting process. Besides, the adaptive fuzzy controller has been designed for corresponding simulation with powertrain system model. Based on the simulation results, the control strategy for clutches has been verified effective for different vehicle mass and road slope.

2) During the hardware-in-the-loop test, the test results are comparable with the corresponding simulation results. The results using adaptive fuzzy control method illuminate that the indicators of shifting quality have been improved during shifting process. It implies that the controller is effective in real-time using the fuzzy control algorithm.

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