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TRACKING CONTROL AND ROBUSTNESS STUDY OF SHIFTING PROCESS

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Abstract:

Heavy dump vehicles are usually working with big load changes and bad work environment, thus change the friction performance of transmission clutches, as well as great affect the shift quality seriously, which influence the vehicle performance. Many researchers developed a lot to design a useful automatic transmission control system. Using PID tracking control and Monte Carlo method, the controller based on an dynamic model was set up to analyze the shifting process of automatic transmission and its robustness in this paper. The shift process was divided into four stages, low-gear phase, torque phase, inertia phase and high-gear phase. The model presents the process from the first gear to the second gear when the torque has big change. Since the jerk and the friction work of clutch are both related to the speed of clutch which was easier to control, it was chose as the target to control the oil pressure for satisfying the requirement of shift quality. The simulation software, Maplesim and Simulink, were used to build the vehicle model and shifting controller for simulation under different working conditions, and the maximum jerk was changed from 34 m/s3 to 12 m/s3 after the optimization. In this paper the Monte Carlo has been used to quantize and evaluate the robustness of the closed-loop system for the friction coefficients and output torque of turbine variation leading by the friction feature parameters and throttle angle changed. Monte Carlo method was used to analyze the effectiveness and robustness of PID controller, which proves that it has good control effect when the throttle is ongoing minor fluctuations. When the throttle is full opening, a quadratic optimal controller based on disturbance is designed by the method of multi-objective optimization. When it changes within 20 percent, PID controller was designed under the guidance of tracking thoughts. The results also show that the controller could still obtain better effect when the friction coefficient ranged from -40 % to 40 % as well as engine torque changed from -20 % to 20 %, which indicates the robustness of controller.

Key words:
automatic transmission, shift quality, PID tracking control, Robustness, Monte Carlo method

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1. Introduction

The automatic transmission has been widely used in heavy vehicle for improving shifting quality and performance of vehicle. The smooth shifting is the great advantage of automatic transmission and many researchers developed shifting control for reducing the jerk during shift (Wieczorek, 2016). Since the ratio varied greatly during first gear shifting to second gear, the shifting process presents properly rough. Additionally, it leads to the control difficultly comparing with other shifting states. This paper develops the shifting process when the first gear shifts up to second one and describe the controller designation in subsequent parts.

There are three indicators for measuring shifting quality, the jerk, sliding friction work and shifting time, and it is interact among those elements (Zhang, 2015). Because of the length of shifting time, the tracking control has been used for smooth shift in this paper and it makes the multi-objective optimization problem changing to single target. For the single control variable the desire shifting quality is easy to achieve. Based on several papers, there are many variables as the tracking target for optimizing shifting quality. Li designed the PID controller of clutch speed error by stochastic algorithm (Li et al., 2013). Gao devised two freedom PID controller based on the speed of turbine toward rigid transmission shaft and it is effective for shifting quality and clutch service life (Gao, 2012). Mousavi designed the PID controller of motor speed to realize the pressure control of clutch utilizing the input/output feedback linearization technology (Mousavi et al., 2014). Yuan presented a method that combining model reference and transmission control, and they designed PID controller of clutch speed for limiting the change rate of engine and torque of clutch to optimize the shifting quality (Yuan et al., 2013). Zhong proposed the shifting control for a non-separable clutch based on feedforward PID controller for tracking engine speed (Zhong et al., 2012). It aims to acquire stabilize engine speed for stable torque. As mentioned above, all researchers have used PID for tracking control which is easy to go for improving shifting quality. Considering many disturbance from external environment in the real system, several researchers solve the uncertain control problem of system by Monte Carlo method. Chen and Almeida used the Monte Carlo method developing the effects of uncertainty parameter for flight control, motor control and intelligent control and they found the robustness and versatility of this method (Chen, 2016; Hou et al., 2016; Almeida et al., 2010).

This paper represented the controller robustness by two uncertainty parameters from internal and external automatic transmission utilizing Monte Carlo method based on the PID speed tracking control, and it receive good effects.

2. Dynamic analysis for shifting

For analyzing gear shifting, the moment balance has been discussed when the first gear shift up to the second gear about the planetary Pd, P2 and P1 with connective clutches Cd, C1 and C2. The moment balance diagram presented as below.

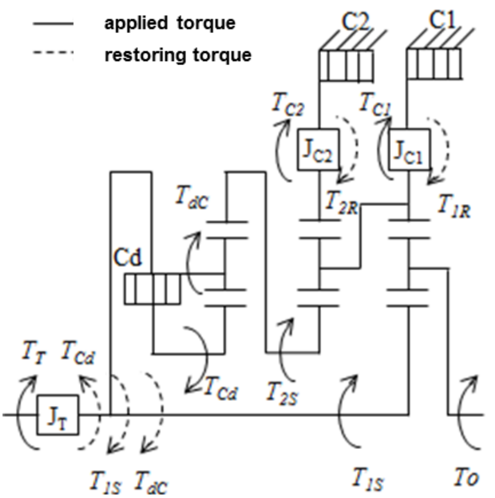

Fig. 1. The moment balance diagram during 1st shift up to 2nd

In generally, the process of shifting should go through four stages: low gear phase, torque phase, inertia phase and high gear phase (Murakami, 1974). The torque remained unchanged during low gear phase and high gear phase. While the torque transferring by clutch C1 would reduce the output torque during the torque phase, the clutch C1 pressure declines to zero at the end of torque phase. Since the turbine speed changed a lot in the inertia phase, the tracking control was used in this phase. According to the analysis shown in the Fig.1, the function of moment balance for studying shifting process presented as follows:

$$\begin{cases} T_T + T_{dS} - T_{1S} - T_{dC} = J_T \cdot \dot{\omega}_T \\ T_{C2} - T_{2R} = J_{C2} \cdot \dot{\omega}_{C2} \end{cases}, \quad (1)$$

where the T_T presents the output torque of turbine, T_{dS} *and* T_{dC} represents the output torque of sun gear and ring in the left planetary respectively, T_{1S}, T_{C2} and T_{2R} presents the output torque of sun gear in planetary P1, carrier in planetary P2 and ring gear in P2, J_T and J_{C2} present the inertia of turbine and carrier in P2, $\dot{\omega}_T$ and $\dot{\omega}_{C2}$ present the angular acceleration of turbine and carrier in P2.

Assuming the k as the planetary array parameters, the planetary system reveals performance representing by the follow equations:

$$n_S + k \cdot n_R - (1+k) \cdot n_C = 0, \quad (2)$$

$$T_S : T_R : T_C = 1 : k : (1+k), \quad (3)$$

where n_S, n_R and n_C present the speed of sun gear, ring gear and carrier respectively, T_S, T_R and T_C present the speed of sun gear, ring gear and carrier respectively.

The relationship between transferring torque and control pressure of clutch has shown as next equation:

$$T_C = \mu_C \cdot K_C \cdot F_C, \quad (4)$$

where μ_C presents the dynamic friction coefficient of clutch friction plates, F_C replaces the pressure of clutch, proportionality factor represents $K_C = zR_e$, z presents the number of friction pair and R_e represents the radius of friction plates.

When the vehicle drives in the level road without wind, the resistance equilibrium equation can be written as follows (Yu, 2009):

$$\frac{T_o \cdot i_0}{r} - F_x = \delta \cdot m \cdot \frac{dv}{dt}, \quad (5)$$

where T_O presents the output torque, Fx presents the resistance of vehicle, m represents the mass of vehicle, v means the velocity, δ takes the place of correction coefficient of rotating mass, i_0 presents the overall ratio both main retarder and wheel reductor and r represents the radius of wheel.
Based on the connected relation shown in Fig.1, the speed of clutch C2 was expressed by the next function:

$$\omega_{C2} = \frac{i_2 \cdot \omega_o}{i_2 - 1} - \frac{\omega_T}{i_2 - 1}, \quad (6)$$

where i_2 presents the ratio of second gear, ω_T and ω_{C2} present the angular speed of turbine and carrier in P2.

Because of the short shifting time, the resistance of vehicle maintains stable without considering the first derivative term of gradient for input speed and output speed. Considering above functions, the jerk and friction work during shifting was derived (Gao et al., 2015).

$$j = \frac{da}{dt} = \frac{1}{\delta \cdot m} \frac{d}{dt}\left(\frac{T_O \cdot i_0}{r} - Fx\right) \approx \frac{\mu_{C2} \cdot K_{C2} \cdot i_2 \cdot i_0}{\delta \cdot m \cdot r \cdot (i_2 - 1)} \frac{dF_{C2}}{dt}, \quad (7)$$

$$W = W_{C2} = \mu_{C2} \cdot K_{C2} \int_{t_1}^{t_2} F_{C2} \cdot |\omega_{C2}| dt, \quad (8)$$

where μ_{C2} presents the dynamic friction coefficient of clutch C2, K_{C2} represents the proportionality factor of clutch C2 and F_{C2} presents the pressure of clutch C2.

It is easy to be found that the jerk and fiction work are related with the speed and pressure of clutch C2. Therefore, when the speed tracking control of clutch C2 was suitable to use for acquiring the desired pressure, the coordination control would be utilized in optimizing shifting quality.

3. Speed tracking controller design with PID

According to the analysis about shifting process, the optimal problem of shifting quality would be changed to a tracking control problem. While the tracking target of clutch C2 should be qualified by some conditions in inertial phase. Because the jerk should be as small as possible at the end of inertia phase, the slope of target curve should be zero. To limit control variables, the slope of target curve would be zero at the beginning of inertial phase. The shifting time shouldn't be too long for minimizing the friction work during shifting. The desired speed curve is shown in the Fig.2 that meeting the above principles (Tao, 2002).

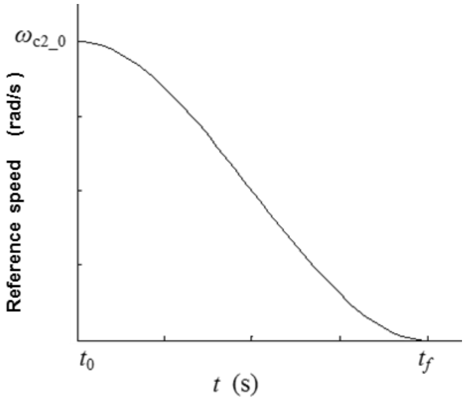

Fig. 2. The desire speed curve

Based on the desired curve, the target speed of clutch C2 could be represented as following equation:

$$\omega_{C2} = \frac{2\omega_{C2\_0}}{(t_f - t_0)^3}(t-t_0)^3 - \frac{3\omega_{C2\_0}}{(t_f - t_0)^2}(t-t_0)^2 + \omega_{C2\_0}, \quad (9)$$

where ω_{C2_0} presents the speed of clutch C2 at the beginning of torque phase, t_0 and t_f represents the starting point and ending point respectively.

4. Simulation results for pressure control of electro-hydraulic proportional valve

According to the shifting process from 1st gear to 2nd gear, the automatic transmission model has been built in the MapleSim software and the shifting controller has been designed though the Simulink for coordination control improving shifting quality (Tao, 2002). The pressure of clutch changed following the signal of shifting start, and it should enter the quick filling phase firstly in which the pressure of clutch C2 fills quickly and lasts about 0.1s. At the same time, the clutch C1 began to discharge achieving the Sliding friction critical oil pressure. The clutch C2 continued filling in torque phase, while the pressure of clutch C1 declined to zero at the end of torque phase. Based on several simulation tests, the desire torque phase time would be 0.1s. After that turning to the inertial phase, the clutch C2 continued to fill until the end of inertial phase which achieved the maximum pressure. It will last 0.4s and the pressure curve of clutch C2 was assured by the control variables from the tracking controller. For observing the simulation results easily, the pressure curve of clutch C1 and C2 for optimizing gear shifting shown in the Fig.3 were magnified by a custom coefficient, $f_{\text{normalized}}$.

$$f_{\text{normalized}} = \frac{F_{C2}}{F_{n\max}}, \quad (10)$$

where, the $F_{n\max}$ presented the maximum static friction force.

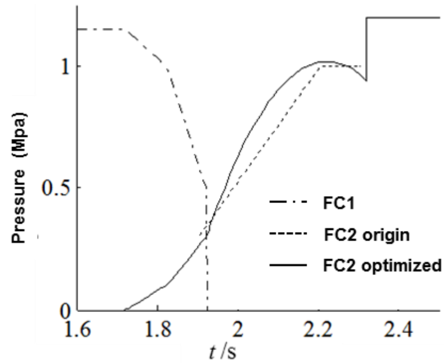

Fig. 3. The pressure curve of clutch C1 and C2

According to tracking the ideal speed curve, the simulation results demonstrated that the jerk during shifting declined a lot and the shifting time only postponed 0.04 s. The speed of clutch C2 and the error of speed are shown in the Fig.4. Comparing with desired speed, the optimized error of speed clutch C2 dropped to 16 % from 56 % without considering the ending point when the error turned to zero.

This paper compares the vehicle performance of the optimized shifting control with the original control method which is shown in the Fig.5. When the speed of turbine becomes to decline, it means the inertial phase starts and the controller begins to work. Based on the results, the vehicle speed was 1.43 m/s at this point. During the shifting, the maximum value of acceleration was 2.3 m/s^2 which is less than the maximum value of acceleration with the original control method, 0.25 m/s^2. Besides, the maximum of jerk dropped to 12 m/s^3 from 34 m/s^3 that achieving the automobile industry standard for transmission (Lagerberg, 2004).

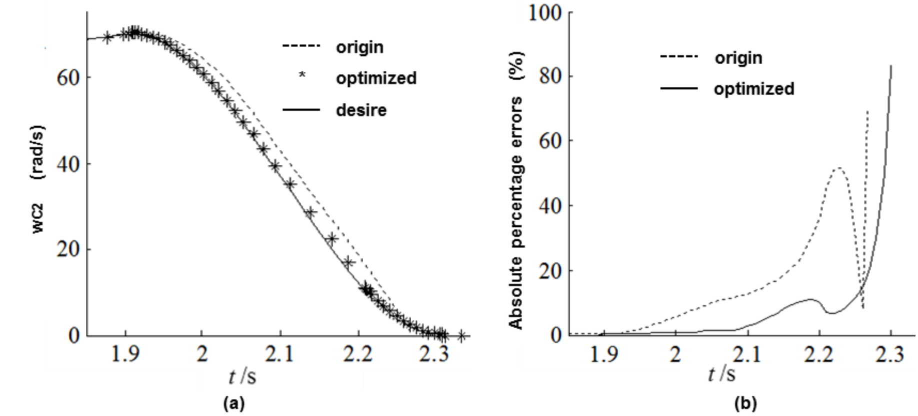

Fig. 4. The tracking speed and error of clutch C2
(a) The speed of clutch C2, (b) The comparison of speed error

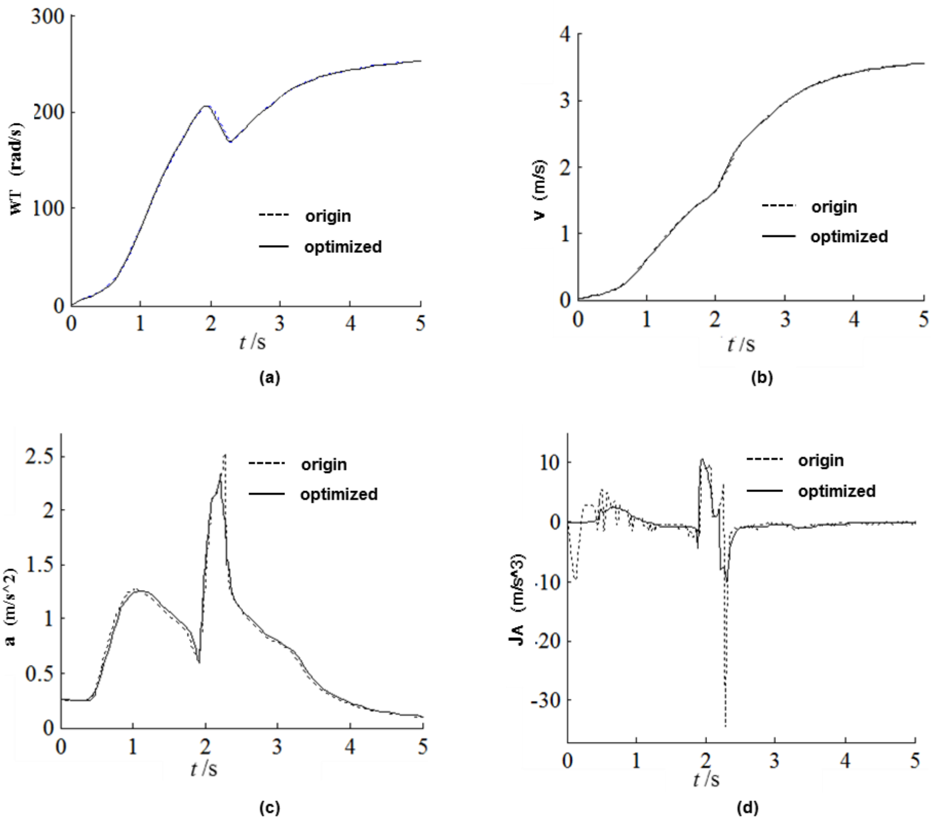

Fig. 5. The performance of vehicle
(a) Turbine output speed, (b) Velocity, (c) Acceleration, (d) Jerk

5. Robust analysis for controller

The simulation results demonstrate that the shifting quality could be improved by tracking feedback control based on specific driving conditions. The designation of controller should be robust when the external disturbance and parameters varied (Samanuhut, 2011).

In this paper the Monte Carlo has been used to quantize and evaluate the robustness of the closed-loop system for the friction coefficients and output torque of turbine variation leading by the friction feature parameters and throttle angle changed. This method could deduce the stability along with the stochastic disturbance for the shifting system. The main procedure includes several parts. The variation model was transformed to the probabilistic model which presents the change of friction coefficients and turbine output torque. Then the random number sequences of those parameters were input to the system, the test results were saved for next step after plenty of tests. Finally, those results are processed statistically and analyzed in detail.

5.1. Toward the varying friction coefficient

The relationship between friction coefficient and relative speed of clutch plate is exponential and the friction coefficient μ can present by following equation:

$$\mu = A + Bexp(-C \cdot \Delta\omega) . \tag{11}$$

where the coefficients of A, B and C represented by numbers 0.0993, 0.0307 and 0.0531 respectively, $\Delta\omega$ presents the relative speed of the clutch.

The A, B and C can be seen as random numbers of normal distribution for realizing friction performance variation. In addition, the random variables were produced by Monte Carlo and the standard deviation is 0.01. According to the way mentioned above, the variation range of the friction coefficient is between -40 % and 40 % which can cover the range of the coefficient during shifting process. Since the method needs amount of data based on a lot of tests, one thousand normal distributed test samples were provided in this paper, which are deduced by different A, B and C shown in Fig.6.

As can be seen the jerk during shifting was revealed in Fig.7. The jerk of vehicle influenced by the friction coefficient, while the probability of the maximum value of the jerk fulfilling the standard remained is about 81%. It demonstrates that the shifting controller has good robustness.

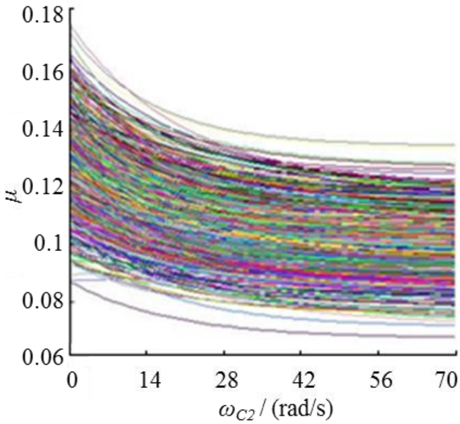

Fig. 6. The friction coefficients based on random sample

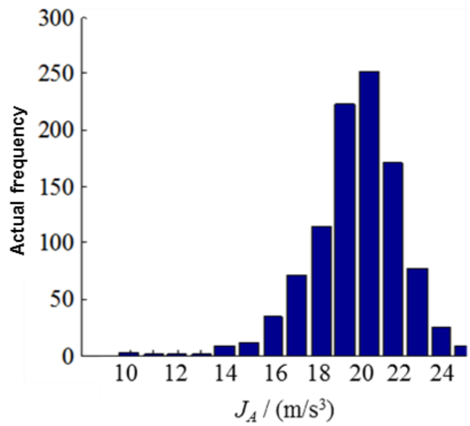

Fig. 7. The column diagram of maximum jerk with different friction coefficients

5.2. Toward the engine torque fluctuation

The output torque of engine was related with the throttle angle. As a result, the unstable driving operation would bring out the shake of throttle angle affecting the output torque fluctuation. Additional, the torque fluctuation deducing by engine work also made the output torque variation.

The engine and torque converter comprise a common device that output torque to the automatic transmission. The output features of this device could be expressed by a polynomial using turbine speed. Because the fluctuation of throttle angle is assumed between -20 % and 20 %, the turbine output torque was fit by the throttle angle and turbine speed.

$$\begin{cases} T_T = D \cdot n_T + E \\ D = -1.82 \cdot \alpha - 0.71 \\ E = 6935.65 \cdot \alpha + 755.74 \end{cases} , \tag{12}$$

where T_T represented the turbine torque, n_T presented turbine speed and D presented coefficients related with throttle angle, E presents the torque based on the partial engine characteristic.

By using Monte Carlo method acquiring random sample for throttle angle, the skewed distribution random number has been utilized with considering throttle angle less than 100 %, where the standard value is 0.95, the variance is 0.06, the skewness is -1.2 and the kurtosis is 3. The different turbine torque and the simulation results about the jerk are shown in Fig.8 and Fig.9 respectively. As can be seen that the turbine torque generated fluctuation when the throttle angle changed in the range of ±20 %. It impacted on the shifting quality, while the probability of the maximum value of jerk fulfilling the standard remained about 87 % which is below 20 m/s^3. The controller has robustness to fulfill the desire shifting quality with external distributions.

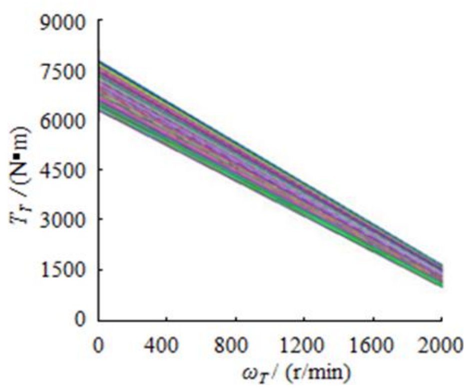

Fig. 8. The turbine torque based on random sample

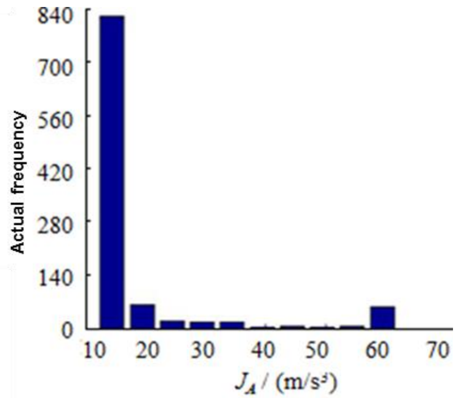

Fig. 9. The column diagram of maximum jerk with different throttle angle

6. Conclusion

(1)The PID tracking controller has been designed for optimizing the pressure during the inertial phase of shifting process, and it simplifies the multi targets problem as single target tracking problem which is easily to achieve.

(2)The simulation results by MapleSim and Simulink model represented that the tracking controller is well-behaved in robustness and real-time performance. The maximum value of jerk based on the simulation declined from 34 m/s^3 to 12 m/s^3 comparing with the results before optimized.

(3)It was developed the system robustness along with the variation of friction coefficient and engine torque by using the Monte Carlo method. The PID controller has stable performances when the range of friction coefficient variation achieves ±40 % and the fluctuation of throttle angle is up to ±20 %.

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