Application of Driving Style Recognition in the Shift Control of a Two-speed DCT for Pure Electric Vehicles

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Abstract. The driving style recognition algorithm can be used to obtain the driver's current driving style in real time. Thus, the algorithm is combined with the transmission control logic to make the shifting operation of the transmission adaptive to drivers of different driving styles. Firstly, by introducing the dynamic coefficient, the comprehensive shift schedule that match different driving styles is calculated. Next, the two transmission shift evaluation indexes, i.e., shift time and degree of jerk are determined, with the corresponding objective function established. Using this function as the constraint function of the NSGA-II multi-objective genetic optimization algorithm, the Pareto optimal solution set of the engagement speed control parameters of the two clutches under different motor output torques is obtained. The transmission shift control strategy proposed herewith is validated by the co-simulation of MATLAB/Simulink and MSC Carsim software. The co-simulation results show that the transmission can automatically adjust the shift parameters according to different types of driving styles, so that the shift schedule and shift quality of the two-speed dual-clutch transmission are in accordance with the needs of drivers with different driving styles.

1. Introduction
Driving styles can usually be divided into three types, namely mild, normal, and radical\textsuperscript{[1]}. The application of driving style recognition to the shift schedule of a transmission can improve the fuel economy of hybrid electric vehicles, or improve the energy efficiency of pure electric vehicles\textsuperscript{[2][3][4]}. In this work, for a dual-clutch transmission (DCT), the shift schedule and shift control strategies that match different driving styles are determined, so that the shift quality can meet the expectations of drivers with different driving styles.

2. Shift dynamic model of the DCT
The dual clutch transmission has one input shaft, two output shafts, three pairs of meshing helical gears, two clutches, actuators, hydraulic circuits, and other components. Since this is a multi-degree-of-freedom, multi-mass system, the equivalent concentrating mass method is used to simplify the entire transmission dynamics system in building the shift dynamic model of the DCT. The schematic diagram of the shift dynamic model of the dual clutch transmission is shown in Figure 1.
3. Research on comprehensive shift schedule (CSS) in accordance with driving style

3.1. Selection of shift parameters

At present, the two-parameter shifting schedule is adopted in most cars equipped with automatic transmissions. The two parameters usually refer to the vehicle speed and the accelerator pedal opening. The driving intention of the driver can often be expressed by these two parameters, as the accelerator pedal opening can reflect the desired torque of the driver, while the speed of the vehicle can reflect the demand of the driver on performance or economy of the vehicle. Therefore, the transmission response to the intention of the driver can be improved by taking different balance between the performance and economy for drivers of different driving style.

3.2. Economy-oriented shift schedule

The economy-oriented shifting schedule is designed to make the motor work in the most energy efficient way while meeting the performance demand. There are many factors that can affect the energy economy of electric vehicles, such as electric motors, batteries, drivetrains, working conditions, and so on. Given the battery and the drivetrain, the motor efficiency $\eta_m$ is the single most significant variable influencing the energy economy, which is therefore taken as the reference index for the economy-oriented shift schedule.

The relationship between the vehicle speed and the efficiency of the electric motor is firstly determined for the transmission in different gears under the same accelerator pedal opening. Then, for a given accelerator pedal opening degree, the vehicle speed corresponding to the intersection point of the first and the second gear motor efficiency curves is taken as the optimal economy-oriented shift speed. The optimal economy-oriented upshift curve can be obtained through interpolation of above-mentioned shift speed points of different accelerator pedal openings, as shown in Figure 2. To avoid the reduction in dynamic performance due to frequent gear shifts, the delay in downshift should be appropriately set. In this work, an equal delay of 7km/h in downshift schedule is adopted.
3.3. Performance-oriented shift schedule

Based on the vehicle parameters, the relationship between the driving force in the two gears and the vehicle speed for 100% accelerator pedal opening can be drawn as shown in Figure 3, in which the acceleration resistance is included in the driving resistance curve. Apparently, the upshift speed can be selected between 61.2 and 117 km/h. In consideration of drivers’ driving habits, it is deemed reasonable to take the speed of the performance-oriented upshift greater than the maximum of that of the economy-oriented upshift. Since the maximum speed of the economy-oriented upshift is 72.3 km/h according to the previous discussion, the upshift speed of 75 km/h is chosen for performance-oriented schedule. Similarly, in order to avoid frequent gear shifts affecting ride comfort, the downshift speed of 68 km/h is taken for the performance-oriented schedule, with an equal delay of 7 km/h in speed. The final result of the performance-oriented shift schedule is shown in Figure 4.

3.4. Comprehensive shift schedule (CSS)

According to Reference [6], a comprehensive shift schedule (CSS) can be formulated by introducing a performance coefficient $\beta$ associated with the accelerator pedal opening, and then the comprehensive shifting point based on the performance coefficient can be determined with the following formulas.
\[ \beta = \frac{(\alpha - \alpha_{\text{min}})}{(\alpha_{\text{max}} - \alpha_{\text{min}})} \]  
\[ v_m = \beta v_p + (1 - \beta) v_e \]

where \( \alpha, \alpha_{\text{max}}, \alpha_{\text{min}} \) represent instantaneous, upper limit, and lower limit of accelerator pedal openings, respectively, with the upper and lower limits being driving style dependent; \( v_m, v_p, v_e \) represent comprehensive, performance-oriented, economy-oriented shift speeds in km/h, respectively.

For a given driving style, the comprehensive shift speed can be adjusted by taking different weighting factors between the economy and performance-oriented shift speeds according to the opening degree of the accelerator pedal. However, the comprehensive shift speed determined by such a methodology is instantaneous, while the accelerator pedal under the driver’s foot cannot always maintain a stable opening during the actual driving. If the delay logic is not set in the shift program, this would not only add the burden to the TCU in actual operation, but also be harmful to the improvement in the robustness of the shift control program. Therefore, it is reckoned necessary that specific performance coefficients be determined according to different driving styles to facilitate the actual shift control of the transmission, with the performance coefficient being a constant for a given driving style.

The CSS is designed by taking the following methodology. The above-mentioned economy-oriented, comprehensive, and performance-oriented shift schedules are taken respectively when the accelerator pedal opening is between 0 and \( \alpha_{\text{min}} \), between \( \alpha_{\text{min}} \) and \( \alpha_{\text{max}} \), as well as between \( \alpha_{\text{max}} \) and 100%. The values of \( \alpha_{\text{min}} \) and \( \alpha_{\text{max}} \) are determined according to the driving style, as shown in Figure 5.

For the moderate driving style, the performance coefficient \( \beta \) is taken as 0.05 since emphasis should be placed on energy economy for the CSS, as shown in Figure 6:

For the normal driving style, the performance coefficient \( \beta \) is taken as 0.3 since a balance between energy economy and dynamic performance should be implemented for the CSS, as shown in Figure 7.
For the radical driving style, the performance coefficient $\beta$ is taken as 0.9 since the weight should be put on the dynamic performance for the comprehensive shift schedule, as shown in Figure 8.

![Figure 8. CSS for the radical driving style.](image)

4. Research on the control of clutch engagement rate in accordance with driving style

4.1. Shift quality evaluation indexes of the DCT

The shift time $t$ and the degree of jerk $j$ are usually selected as indexes to evaluate the shift quality of the DCT [7].

4.1.1 Shift time $t$.

Taking the positive torque upshift process as an example, the shifting time of the DCT can be expressed as $t = t_1 + t_2 + t_3$, where $t_1$ - shift preparation time (s); $t_2$ - torque phase time (s); $t_3$ - inertial phase time (s).

Let the oil pressure change rates of clutch C1 in torque phase and inertia phase be $k_a, k_b$ respectively, and let the oil pressure change rates of clutch C2 in torque phase and inertia phase be $k_c, k_d$ respectively.

Then, by solving the dynamic equations, $t_2, t_3$ related to $k_a, k_b, k_c, k_d$ can be obtained dynamic equations.

\[
\begin{align*}
t_2 &= f_1(k_a, k_c, T_m) \\
t_3 &= f_2(k_a, k_b, k_c, k_d, T_m, N_m)
\end{align*}
\]

(3)

4.1.2 Degree of jerk $j$.

The formula for calculating the degree of jerk is

\[
j = \frac{1}{\delta m_{R_{tire}} \eta_t} \frac{d(T_{out_{t0}} \eta_t - T_r)}{dt}
\]

(4)

Where $\delta$ - vehicle rotation mass conversion factor, according to the empirical formula in reference [8]; $\eta_t$ - total mechanical efficiency of the transmission system; $T_r$ - resistance torque of the vehicle from the ground. It is assumed that $\delta = 1.08$ and $\eta_t = 96\%$ in this work.

For the convenience of calculation, and in considering the very short shift time of usually less than 1s, it can be assumed that $T_r$ is basically unchanged during shift. By combining the dynamic equations of the transmission with the formula of the degree of jerk, the formula for the degree of jerk can be rewritten as
Then the relationship between the degree of jerk and \(k_a, k_b, k_c, k_d\) in the torque phase and the inertia phase can be obtained respectively:

\[
\begin{align*}
  j_1 &= g_1(k_c) \\
  j_2 &= g_2(k_a, k_b, k_c, k_d, T_m) \\
  j_{\text{max}} &= \max(j_1, j_2)
\end{align*}
\]

Referring to the above-mentioned formulas of the shift time \(t\) and the degree of jerk \(j\), it can be found that under the same shift condition, the degree of jerk increases as the shift time decreases. On the other hand, the demand for shift quality can be different for different type of drivers. For example, conservative drivers would prefer a gentle and comfortable driving style, with smooth gear shifts. Therefore, when setting the clutch engagement speed for a conservative driver, it should be ensured that the shift time is as short as possible as long as the degree of jerk does not exceed the widely accepted upper limit.

4.2. The NSGA- II algorithm for solving the rate of change in oil pressure for clutch control

Based on the two dependent variables of shift time \(t\) and degree of jerk \(j\), the selection of \(k_a, k_b, k_c, k_d\) is a typical multi-objective optimization problem. A genetic optimization algorithm based on Pareto optimal solution is proposed, which can greatly improve the computational speed and accuracy of the multi-objective genetic optimization algorithm \[9\]. Therefore, the second-generation non-dominated sorting genetic algorithm with elite strategy (NSGA-II) is employed in this work.

(1) Optimization variables

The rates of change in oil pressure \(k_a, k_b, k_c, k_d\) of the clutch C1 and the clutch C2 in the torque phase and the inertia phase are selected as the variable to be optimized with the NSGA-II multi-objective genetic optimization algorithm.

(2) Objective function

Next, taking the positive torque upshift as an example, the objective function of the NSGA-II multi-objective genetic optimization algorithm is determined.

The first objective function is the shift time \(t\). According to the dynamic equations of the torque phase and inertia phase, the first objective function can be derived as

\[
  f_{\text{object}1} = t = \frac{90\%T_m}{\lambda_2 k_c} + \max \left( \frac{P_{1\text{max}} - 90\%T_m}{\lambda_1 k_c}, \frac{P_{2\text{max}} - 90\%T_m}{\lambda_2 k_d} \right)
\]

Where, \(\lambda_1, \lambda_2\) - coefficients dependent on the size and material of clutch C1 and clutch C2 respectively; \(P_{1\text{max}}, P_{2\text{max}}\) - maximum oil pressure in the hydraulic circuits for controlling clutches C1 and C2 respectively.

The second objective function is the degree of jerk \(j\) in torque phase. Since the degree of jerk is a vector, in order to avoid missing a good group in the genetic optimization algorithm, according to Equation 4, the absolute value of the degree of jerk should be used as the objective function. Thus, the second objective function can be derived as

\[
  f_{\text{object}2} = |j_1| = \left| \frac{i_0 \eta_t}{\delta m R_{\text{tire}}} \frac{d(i_1 T_{C1} + i_2 T_{C2})}{dt} \right|
\]

The third objective function is the degree of jerk \(j\) in inertia phase, which can also be derived as

\[
  f_{\text{object}3} = |j_2| = \Phi \left| \frac{T_m k_d}{\theta_1 - 90\%T_m}, \frac{T_m k_b k_c}{\theta_2 k_c - 90\%T_m k_a} \right|
\]

Where, \(\theta_1, \theta_2\) - coefficients dependent on the size and material of clutch C1 and clutch C2 respectively; \(\Phi\) - coefficient related to vehicle dynamics.
(3) Constraint conditions
Since the upper limit of the acceptable degree of jerk is different for different countries, the upper limit of 17.64 m/s$^3$, a recommended value in China, is taken in this work. Therefore, the constraint for the degree of jerk can be expressed as $\max(j_1, j_2) < 17.64 m/s^3$.

Although there is no standard or recommended value for the shift time, the shift time of several DCTs currently in production can be taken as references. The effect of friction work on the service life of clutch plates should also be considered when restricting shift time. Based on engineering experience and generally accepted criterion, the upper limit of acceptable shift time of 0.6s is assumed, namely: $t < 0.6$.

For clutch C1, in the torque phase, the clutch plate remains in the engagement state. Once it enters the inertia phase, it should quickly separate. In order to ensure the separation of the clutch C1 successfully during the shift, the oil pressure change rate in the inertia phase must be higher than that in the torque phase: $k_a < k_b$.

(4) Optimization Results
In the multi-objective genetic optimization algorithm, the optimization result improves theoretically as the number of the population $N$ becomes large. However, this would significantly increase the number of iterations and the calculation time. According to the empirical formula given in Reference [10], the number of populations is selected as $N=100$, and the maximum hereditary algebra is 500. Then, with the working conditions simulated for the output torque of electric motor before shift ranging from 10N·m to 300N·m, the final optimization results are obtained. Part of the optimization results are shown in the Tables 3-1 through 3-2.

| Table 3-1 Optimization results of $T_m = 90N \cdot m$ |
|---|---|---|---|---|---|---|
| NO. | $k_a$ | $k_b$ | $k_c$ | $k_d$ | $j_1(m/s^3)$ | $j_2(m/s^3)$ | $t(s)$ |
| 1 | 2.54 | 18.56 | 0.85 | 18.92 | -1.25 | 10.96 | 0.47 |
| 2 | 3.36 | 19.00 | 1.41 | 18.68 | -2.06 | 8.81 | 0.32 |
| ...... | | | | | | | |
| 99 | 3.02 | 19.47 | 1.56 | 19.41 | -2.29 | 7.23 | 0.30 |
| 100 | 2.03 | 18.56 | 0.73 | 18.89 | -1.06 | 10.07 | 0.54 |

| Table 3-2 Optimization results of $T_m = 180N \cdot m$ |
|---|---|---|---|---|---|---|
| NO. | $k_a$ | $k_b$ | $k_c$ | $k_d$ | $j_1(m/s^3)$ | $j_2(m/s^3)$ | $t(s)$ |
| 1 | 3.04 | 14.27 | 1.86 | 13.39 | -2.72 | 17.62 | 0.45 |
| 2 | 11.33 | 12.17 | 8.32 | 20 | -12.16 | 1.11 | 0.15 |
| ...... | | | | | | | |
| 99 | 3.32 | 14.21 | 2.09 | 13.35 | -3.05 | 16.56 | 0.41 |
| 100 | 9.82 | 12.21 | 5.19 | 19.61 | -7.58 | 10.46 | 0.20 |

Part of optimization results are shown in Figure 9, as compared with those results obtained without optimization.
With the abscissa being the shift time $t$ and the ordinate being the absolute value of the maximum degree of jerk, the larger circles in Fig. 3.1 represent the shift quality evaluation indexes of the transmission before optimizing the rate of change in oil pressure of clutches for various output torque conditions of the electric motor. By connecting the circles through an interpolation curve, the change of the transmission quality evaluation indexes with the output torque of the electric motor before optimization can be roughly expressed. It is clear that when the clutches C1 and C2 are both working under the fixed rate of change in oil pressure, both the shift time and the degree of jerk increase with the increase in motor torque before shift. Therefore, in order to ensure that the shift quality meets the expectations of different types of drivers, the optimal results with lower degree of jerk should be selected for conservative drivers under different motor torque conditions. For aggressive drivers, the optimal results with less shift time should be selected. For ordinary drivers, the optimal results can be selected near the diagonal line passing the coordinate origin.

5. Simulation results and analysis

In order to verify the application of driving style recognition in transmission control, in particular, the driving style based CSS and clutch engagement rate control, the time trace data of vehicle speed from a verification driver is imported into the Drive Cycle module in Matlab/Simulink model for co-simulation. In order to demonstrate the effectiveness of the transmission control strategy proposed clearly, the results of co-simulation for the CSS and clutch engagement rate control without consideration of driving style are taken for comparison. The two-parameter economy-oriented shift schedule formulated in section 3.2 is adopted.

5.1. Validation of the driving style based on CSS

From the results of the co-simulation shown in Figure 10, it is obvious that the radical driving style is well reflected. In comparison to the case without application of the CSS, the downshift takes place earlier from about 376s to 371s, while the upshift time is delayed from about 384s to 386s, as shown in Figure 10(a). On the other hand, the speed of electric motor in the case with CSS is higher than that without CSS, as shown in Figure 10(b). After driving the same distance, the battery SOC for the vehicle with CSS is 85.55%, which is slightly lower than that without CSS 86.09%, which is an indication of slightly more energy consumption in response to the radical driving style of the driver.
The results of co-simulation show that the CSS proposed in this work can improve the response of the vehicle to the driver's radical driving style.

5.2. Validation of the driving style based clutch engagement rate control

In order to validate the optimal results of degree of jerk and shift time through the co-simulation, the following procedures are taken:

1. The initial vehicle speed is set to zero;
2. The dynamic shift schedule described in section 2.3 is applied;
3. The accelerator pedal opening is kept at 100% to accelerate the vehicle;
4. The degree of jerk and shift time during the shift process are recorded for the three types of driving styles.

In the co-simulation result shown in Figure 11, the vehicle performs a positive torque upshift when the vehicle speed reaches 75 km/h at about 8.86s. The shift time is 0.28s, 0.36s, and 0.48s, and the maximum degree of jerk is -16.74m/s³, -10.22m/s³, and 7.46m/s³ for the drivers with radical, normal, and mild driving styles, respectively. It is indicated that with the driving style based optimization of the clutch engagement rate parameters by the NSGA-II multi-objective genetic optimization algorithm, the
transmission can well match the intention of the driver automatically by adjusting the engagement and release rates of the two clutch plates according to different types of driving styles.

6. Conclusion

Based on the results from driving style recognition, a comprehensive shift schedule (CSS) and an optimization method of clutch engagement rate are proposed. Through co-simulation the following conclusions can be made:

(1) With the driving style based CSS, the response of the two-speed DCT to the intention of the driver can be improved through adjustment of the shift timing according to different types of driving styles.

(2) With the driving style based optimization of the clutch engagement rate by NSGA-II genetic optimization algorithm, the two-speed DCT can adapt to the type of driving style of the driver by automatically altering the shift time and shift quality through oil pressure control.

7. References

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