Cooperation mode optimization between two clutches for dual clutch transmission during the torque phase

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Abstract. The study focuses on the control method optimization for the Dual Clutch Transmission upshift process during the torque phase. The upshift process and two power flow paths are analyzed in detail. According to the analysis results, if two clutches slip together during the DCT upshift process, either torque hole or power circulation generates. To avoid these two conditions, the torque relationship between two clutches is optimized to enable the off-going clutch to disengage without slippage. The Matlab/Simulink platform is used to establish the DCT dynamic model. The obtained simulation results illustrate that, compared with the control methods based on two clutches slippage, the optimized one can not only avoid torque hole and power circulation, but achieve the highest system efficiency during the torque phase, which means the least friction work and the extension of clutch life.

1. Introduction

Combining the advantages of the Manual Transmission (MT) and the conventional Automatic Transmission (AT), DCT becomes the latest trend in transmissions and attracts extensive development interests in the automotive industry over the last few years. For all kinds of power transmission units, when fitted into vehicles, associated technologies should be studied and it is same with DCT. However, this new piece of technology for DCT still needs to be improved.

From a kinematic point of view, gearshifts in DCT are much similar to clutch-to-clutch shifts in AT [1, 2]. Hence, just like the control methods developed for AT [3-6], the fuzzy control and the optimal control are usually adopted to balance the jerk intensity and friction work during the launch and shift processes [7]. Zhao [8] proposes the fuzzy time decision and the model-based torque coordinating control strategy. From the simulation results and test data, the control strategy can not only meet the shift quality requirements, but also adapt to the various shift intentions, with a strong robustness. As for the optimal control, the jerk intensity and friction work are usually chosen to develop the quadratic performance index function, and based upon the extremum value theorem, the clutch dynamic optimal engagement curves can be obtained in a number of typical working conditions [9, 10]. Moreover, other methods are also developed to achieve precise torque control and obtain launch and shift smoothness for DCT vehicles. In [11], a model-based nonlinear gearshift controller is designed with the backstepping method to improve the shift quality and the effectiveness of the designed controller is validated by Hardware in the Loop simulation results. For the dry DCT, Zhao [12] proposes a self-adaptive
optimal control method based on the minimum principle. With the optimal and engine torques and rotating speed, the minimum jerk intensity and friction work are obtained.

Owing to the considerable work, DCT performance has been improved significantly. However, throughout the current control strategies, there is an obvious distinction. Some authors assert that two clutches should slip together to transfer the engine torque from the off-going clutch to the on-coming clutch. However, other researchers hold the view that DCT can achieve a smooth torque transfer process with only one slipping clutch (on-coming). This distinction indicates that further investigation about the relationship between two clutches is required to improve the DCT shift quality.

The study begins with an in-depth analysis of the DCT upshift process and two power flow paths during the torque phase. According to the analysis results, an optimized torque relationship between two clutches during the torque phase is proposed to obtain the highest system efficiency. Then, the Matlab/Simulink platform is used to develop the dynamic model. From the simulation results, the effectiveness of the proposed control method is validated.

2. DCT powertrain model development and problem statement

2.1. Integrated DCT powertrain model development

The DCT structure is shown schematically in Fig.1 and the working principle of DCT vehicle can be found in [1]. To develop the DCT powertrain model, major components of the vehicle should be modeled firstly. As shown in Fig. 1, the DCT vehicle consists of several complicated components, including the engine, clutches and the driving resistance. These component models are described as follows.

![Figure 1. Structure and dynamic model for DCT.](image1)

![Figure 2. Diesel engine torque output.](image2)
As shown in Fig. 2, the engine is modeled as a mean value torque generator, which is determined by the engine speed and the throttle opening.

The clutches in DCT are the essential gear changing components and described as Coulomb friction elements. The friction torque transferred by clutches in the slipping status depends on the cylinder's pressure and can be calculated according to the geometry and friction characteristics of the clutch.

\[
T = \mu P c A n \left( 2 \frac{R_0^3 - R_1^3}{R_0^2 - R_1^2} \right) \tag{1}
\]

Where \( \mu = \mu_s + (\mu_l - \mu_s) e^{-k_{\mu} t} \).

The driving resistance, which accounts for the road slope resistance, rolling resistance and aerodynamic resistance, is formulated in the equation (2).

\[
T_R = \frac{1}{2} \rho C_d A v^2 + mgf \cos \beta + mg \sin \beta \left( C_d + C_s \right) r_w \tag{2}
\]

2.2. DCT upshift process description

With the component models, the DCT upshift process can be analyzed and described. Similar to AT, the DCT upshift process consists of two phases: the torque phase when the engine torque is transferred between clutches and the inertia phase when the on-coming clutch is gradually synchronized. Fig. 3 shows the different stages of the 4-5 upshift process.

![Figure 3. Different stages of upshifting process.](image)

Due to the change of the clutch status during the DCT upshift process, different sets of equations are required to describe the system dynamics. When DCT operates in fourth gear, only CL is engaged. According to the dynamic balance relationship, the equations for DCT can be derived as follows.

\[
T_e - T_{in} = I_e \omega_e \tag{3}
\]

\[
T_{in} - T_{CL} = I_{in} \omega_{in} \tag{4}
\]
\[ T_{CL}I_4 - T_o = I_{eq}^4 \dot{\Theta}_r \] (5)

\[ T_o - \frac{T_R}{I_o} = I_{eq} \dot{\Theta}_p \] (6)

Where \( I_{eq}^4 \) is the equivalent mass moment of inertia for \( I_1 - I_3 \) in the output side of the output shaft, \( I_{eq} \) is equivalent mass moment of inertia of vehicle body \( I_R \) and \( I_o \) in the input side of the final drive. The input shaft torque \( T_{in} \), the output shaft torque \( T_o \), \( I_{eq}^4 \) and \( I_{eq} \) are expressed by following equations.

\[ T_{in} = k_i (\Theta_e - \Theta_m) + c_i (\omega_e - \omega_m) \] (7)

\[ T_o = k_o (\Theta_o - \Theta_m) + c_o (\omega_o - \omega_m) \] (8)

\[ I_{eq}^4 = I_1 \cdot \dot{\Theta}_e^2 + I_2 \cdot \dot{\Theta}_e^2 + I_3 \] (9)

\[ I_{eq} = I_o + \frac{I_R}{I_o^2} \] (10)

When the DCT receives the upshift signal, the torque phase starts. During this phase, the pressure applied on the off-going clutch is released gradually. While the pressure on the on-coming clutch increases. Then, the power from the engine is redistributed between two clutches gradually. The equations (4) and (5) should be changed and the rest of the equations are the same as those in the 4th gear.

\[ T_{in} - T_{CL} - T_{CH} = I_{in} \dot{\Theta}_m \] (11)

\[ T_{CL}I_4 + T_{CH}I_5 - T_o = I_{eq}^4 \dot{\Theta}_p \] (12)

After the pressure applied on CL decreases to zero, the inertia phase \((t_2 \leq t \leq t_3)\) comes. During this phase, the pressure applied on CH continues increasing and the engine torque is transferred by slipping CH. Correspondingly, equations (11) and (12) should be changed into equations (13) and (14).

\[ T_{in} - T_{CH} = I_{in} \dot{\Theta}_m \] (13)

\[ T_{CH}I_5 - T_o = I_{eq}^4 \dot{\Theta}_p \] (14)
When the relative angular velocity between CH driving and driven discs reduces to zero, the gear ratio changes to the fifth gear. With the analysis of the DCT upshift process, an integrated powertrain model can be developed. However, the optimal relationship between two clutches during the torque phase is still not clear. To reveal the mechanism of the torque interaction process and find an optimal way to accomplish this process, the method of power flow analysis is introduced.

3. Power flow analysis during the torque phase
In essence, the torque phase in the DCT shifting process is a power redistribution process from the off-going clutch to the on-coming clutch. Analyzing the power flows conditions is beneficial to improve the DCT performance.

3.1. Power flow path I
In Fig. 4, the solid line indicates the power flow path I. There is no circulating power, however torque hole exists probably. The power of the corresponding points are denoted as follows, input power \( P_{\text{in}} \), power from CL and CH on the output shaft \( P_{1}, P_{2} \), output power \( P_{o} \), power on driving and driven sides of CL, CH \( P_{LL}, P_{LR}, P_{HL}, P_{HR} \), \( \eta_{4} \) and \( \eta_{5} \) are the mechanical efficiency of fourth and fifth gears respectively. When the engine power is transferred by this path, it can be described as follows.

\[
P_{\text{in}} = P_{LL} + P_{HL}
\]

\[
P_{LR} = \frac{\omega_{1}}{\omega_{m}} P_{LL}
\]

\[
P_{HR} = \frac{\omega_{2}}{\omega_{m}} P_{HL}
\]

\[
P_{1} = \eta_{4} P_{LR}
\]
\[ P_2 = \eta_s P_{\text{in}} \]  
\[ P_r = P' + P_2 \]  

Define the ratio between the output and input power of off-going clutch:
\[ i_L = \frac{\omega_L}{\omega_{\text{in}}} \]  

Power distribution coefficient:
\[ \zeta' = \frac{P_{\text{ll}}}{P_w} \]  

Then, the DCT system efficiency under this condition can be deduced from equations (15-22).
\[ \eta_L = i_L \left( \eta_s (1 - \zeta') + \eta_s \frac{i_L}{i_L} \zeta' \right) \]  

3.2. Power flow path II

When DCT transfer the engine power according to the dotted line in Fig. 4, the circulating power exists.

And the equations for path II are presented as follows.
\[ P_{\text{ll}} = P_{\text{ll}} + P_{\text{in}} \]  
\[ P_{\text{ll}} = \frac{\omega_L}{\omega_1} P_{\text{lr}} \]  
\[ P_{\text{lr}} = \frac{\omega_L}{\omega_{\text{in}}} P_{\text{rl}} \]  
\[ P_{\text{lr}} = \eta_s P_1 \]  
\[ P_2 = \eta_s P_{\text{lr}} \]
According to equations (21, 22, 24-29), the DCT system efficiency can be deduced as follows.

\[ P_2 = P_1 + P_\nu \]  

(29)

Then, the DCT system efficiency during the torque phase can be described by the equation (31).

\[ \eta = \eta_L \left\{ \frac{1}{\eta_L} \left( 1 - \zeta \right) + \eta_i \frac{i_L}{L} \zeta \right\} \]  

(30)

(31)

3.3. Analysis of power flow conditions

In the equation (31), \( \zeta \) varies with the charging pressure from 0 to 1 during the torque phase. Thus, except the configuration parameters \( (i_4 \eta_4 i_5 \eta_5) \), the DCT system efficiency has a strong relationship with \( i_L \). To obtain an optimal status for two clutches during the torque phase, different conditions are analyzed.

As shown in Fig. 5, the condition of \( \omega_m < \omega_1 \) indicates that CL transfers the negative torque, resulting in the generation of circulating power. Obviously, during the upshift process, this condition wastes the system power, reduces the system efficiency and may damage the DCT system. Hence, it is avoided by the current proposed control strategies.

When it comes to the condition of \( \omega_m > \omega_1 \), both two clutches are under slipping status and the power interruption generates as \( i_L < 1 \). For most current control strategies which are proposed based on two slipping clutches, DCT is usually under this status. Although the weakness of this torque transfer method is not obvious when taking drag torque, time delay and many other factors into account during the upshift process, the power interruption does exist. Then, the shift time will be extended and the friction work increases due to the reduction of the system efficiency.

Thus, according to the above analysis, if two clutches slip together during the torque phase, either the power interruption or the power circulation generates. To avoid these two kinds of power loss, CL should be under the status of \( \omega_m = \omega_1 \), which means the off-going clutch does not slip during the torque
phase. Consequently, DCT obtains the highest system efficiency and the least friction work. According to the above analysis, the torque relationship between two clutches is deduced as follows.

\[
T_{CL} = \frac{T_{in}^45 - T_{CH} \left( I_{eq}^45 + I_{eq}^3i_4^3 \right) + I_{in}^45i_4^4}{I_{in}^45 + I_{eq}^45}
\]  

(32)

4. Simulation and analysis

To validate the effectiveness of the proposed control method, the simulation model of the DCT powertrain system is established on the Matlab/Simulink platform. The main parameters of DCT are shown in Table 1.

| Parameters | Value      | Parameters | Value      |
|------------|------------|------------|------------|
| \(k_i\)    | 283429 N\(\cdot\)m/rad | \(c_i\)    | 2 N\(\cdot\)m\(\cdot\)s/rad |
| \(k_o\)    | 54513 N\(\cdot\)m/rad  | \(c_o\)    | 2 N\(\cdot\)m\(\cdot\)s/rad |
| \(I_e\)    | 3.65 kg\(\cdot\)m\(^2\) | \(I_m\)    | 0.01637 kg\(\cdot\)m\(^2\) |
| \(I_o\)    | 0.001535 kg\(\cdot\)m\(^2\) | \(I_R\)    | 18.9021 kg\(\cdot\)m\(^2\) |
| \(I_1\)    | 0.2102 kg\(\cdot\)m\(^2\) | \(I_2\)    | 0.2084 kg\(\cdot\)m\(^2\) |
| \(I_3\)    | 0.2523 kg\(\cdot\)m\(^2\) | \(i_4\)    | 1.29 |
| \(i_5\)    | 0.775      | \(d_c\)    | 5.97e-5 |
| \(\mu_s\)  | 0.131      | \(\mu_k\)  | 0.051 |

As shown in Fig. 6, \(P_{CH}\) is adopted to engage the on-coming clutch. Curve \(P_2\), obtained from the proposed control strategy, presents the optimized effective pressure applied on CL. To make a comparison with the proposed control strategy, curve \(P_1\), obtained according to the control strategy based on two clutches slippage [8], is introduced. For this releasing pressure profile, it is reduced as the pressure on the oncoming clutch increases during the torque phase. In terms of the condition with power circulation, it is also taken into account (curve \(P_3\)) to investigate the system efficiency.

Simulation results of the upshift process based on the proposed control method are shown in Fig. 7(a) and 7(b). As the excellent match of the releasing and charging pressure, \(T_o\) shown in Fig. 7(a) has nearly no change during the torque phase, which means constant torque is transferred. From Fig. 7(b), the relative angular velocity between driving and driven discs of CL keeps zero. Thus, as designed, only the on-coming clutch slips during the entire upshift process. And in terms of the gear ratio shown in Fig. 7(b), it maintains 1.29 during the torque phase and then varies from 1.29 to 0.775 smoothly.
Compared with Fig. 7(a), the output torque in Fig. 8(a) has an obvious decrease, caused by the reduction of the torque capacity of the off-going clutch. Then, with the disengagement of CL, the engine speed is slightly higher than CL speed and two clutches slip together to transfer the engine torque. The power interruption generates, resulting in the decline of the output speed and the increase of the gear ratio in Fig. 8(b).
In terms of $P_3$, CL begins to transfer negative torque from 0.9s, resulting in the drastic change in $T_o$ (Fig. 9(a)). And, as the generation of power circulation, the output speed declines. Meanwhile, the gear ratio increases from 1.29 to 1.4, and then falls to 0.775.

**Figure 9(a).** Torques when $P_3$.

**Figure 9(b).** Gear ratio and speeds when $P_3$.

Fig.10 shows that the gearbox obtains the highest system efficiency when it upshifts with the proposed control strategy. And, of course, the friction work generated during the upshift process is the least (6982J) compared with two other conditions (8054J and 8450J) as shown in Fig.11. As for the results in Fig.12, it shows that both control strategies can satisfy the requirements of shift quality control. However, compared with the control strategy proposed based two clutches slippage, the proposed one reduces the jerk intensity from 4.6 m/s$^3$ to 2.5 m/s$^3$ effectively during the torque phase. Therefore, from the simulation results, the proposed control strategy for DCT upshift has a better performance and leads to excellent smoothness and responsiveness.

**Figure 10.** System efficiency
5. Conclusion

1) The working principles of the DCT upshift process has been investigated, meanwhile, two different power flow paths during the torque phase are analyzed in detail. According to the analysis, if two clutches slip together during the upshift process, either torque hole or power circulation generates. To avoid these two conditions, the torque relationship between two clutches is optimized.

2) According to the optimal corporation mode, a dynamic model integrating the engine, the transmission and the vehicle environment has been established on the Matlab/Simulink platform. From the simulation results, the optimized control method can not only obtain the highest system efficiency, but reduce the shift time and smooth the shift process effectively. The effectiveness and the correctness of the proposed cooperation mode between two clutches is confirmed.

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