Temperature Variation of Wet Clutches in the DSG Vehicle during a 10km Driving Cycle

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Abstract. The temperature of wet clutches has a vital influence on the vehicle in terms of dynamic performance, service life, wear characteristics, etc. To investigate the temperature variations during the operation process, a detailed dynamic powertrain model of the DSG vehicle is established firstly to calculate the dynamic load of different clutches. Then, a thermal resistance model for the gearbox cooling system is built to predict the temperature of wet clutches. Integrating the dynamic model and the heat transfer model, a co-simulation model is finally obtained. The simulation results illustrate that the temperature of the clutch used for the launching process has a sharp increase from 100°C to 123°C; however, within a 10km driving cycle, the temperature of five clutches are all lower than 125°C, which meet the requirements of the vehicle.

1. Introduction

Combining advantages of automatic transmission and manual transmission, the direct shift gearbox (DSG), which also called dual clutch transmission (DCT), not only has the advantages of high efficiency, compactness and low weight, but also can achieve power shift process without power interruption and circulation [1]. Recently, DSG gets extensive and growing attention from the industry and is becoming widely used in vehicles. Similar to other transmissions, a lot of researches have been done to achieve excellent vehicle performance, including optimizing the clutch engagement process during the launching process and the torque transfer process during the gearshift process.

For the launching process, a sliding mode variable structure control strategy is proposed to improve the dynamic performance of DSG vehicle [2]. From the simulation results, the designed control strategy can not only reflect driving intention adequately, but also has strong robustness. However, launching the vehicle with only one clutch is not sufficient to utilize the structure advantage of DSG. A control method based on twin-clutch torque distribution is provided for the DSG vehicle [3]. The simulation results indicate that the proposed method could greatly shorten the launching time, share the friction work by two clutches and avoid possible power coupling and power cycling.

As for the DSG shift process, it can be divided into the torque phase and the inertia phase. During the torque phase, the engine torque is transferred from the leaving clutch to the coming clutch. To obtain a smooth transfer process, a detailed dynamic model of DSG vehicle is established and a control strategy is proposed in [4]. The results show that the vehicle can achieve good performance during the torque phase. Furthermore, to investigate the effects of clutch application timing, three different clutch pressure profiles are studied, and the optimized profile can obtain the best possible shift quality [5]. When it comes to the torque relationship between two clutches, a torque based
control strategy is proposed with only oncoming clutch slipping [6]. In terms of the inertia phase, the power-on upshift and power-off downshift process are optimized by the feedback control and torque based logic control [7]. In addition, an integral linear quadratic regulator is introduced by Guoqiang to regulate the relative speed between the engine and the slipping clutch during the inertia phase [8]. From the simulation results, the proposed controller can reduce jerk level and improve shift quality significantly, compared with conventional controllers.

Owing to these considerable researches, the performance of DSG vehicle has been improved significantly. However, when it comes to the thermal problems during the vehicle operation process, it is seldom considered, especially for the temperature variation of clutches during a continues driving cycle. To fill that gap in knowledge, the thermal status of the clutches in the DSG vehicle is investigated.

In the second part, a powertrain model for the DSG vehicle is proposed to calculate the dynamic load during the vehicle operating process. In the third part, a thermal resistance model for the gearbox cooling system is built. After that, the simulation results are presented in the forth part. Finally, the conclusions are presented.

2. Dynamic model of the DSG vehicle

![Powertrain system of the DSG vehicle](image)

The structure of the DSG vehicle is presented in Figure 1. As there are no synchronizers in this DSG, the launch and the shift processes are completed with different slipping clutches. By controlling clutches according to Table 1, various gear ratios can be achieved. The sign “●” presents that the corresponding clutch is engaged.

| Gear | CL | CM | CH | C1 | C2 |
|------|----|----|----|----|----|
| 1    | ●  | ●  | ●  | ●  | ●  |
| 2    | ●  | ●  | ●  | ●  | ●  |
| 3    | ●  | ●  | ●  | ●  | ●  |
| 4    | ●  | ●  | ●  | ●  | ●  |

2.1. Launching process description

During the launching process, C1 is pre-engaged, and only CL is slipping to start the vehicle. Then, the engine power is transferred from the input shaft to the output shaft through DSG. According to the Newton second law, the system dynamic can be deduced as follows.
\[ (I_v + I_m') \frac{d\omega_v}{dt} = M_v - M_i \]
\[ I_1' \frac{d\omega_1}{dt} = M_i - M_{CL} \]  \[ (2) \]
\[ I_2' \frac{d\omega_2}{dt} = M_{CL} i_{CL} - M_{C1} \]  \[ (3) \]
\[ I_3' \frac{d\omega_3}{dt} = M_{C1} i_{C1} - M_o \]  \[ (4) \]
\[ (I_v' + I_b') \frac{d\omega_i}{dt} = M_o - \frac{M_R}{i_b} \]  \[ (5) \]
\[ M_i = S_m (\theta_i - \theta_1) + C_m (\omega_i - \omega_1) \]  \[ (6) \]
\[ M_o = S_m (\theta_2 - \theta_o) + C_o (\omega_3 - \omega_o) \]  \[ (7) \]
\[ M_R = \left( \frac{1}{2} \rho C_v A \frac{v^2}{2} + mg \cos \beta + mg \sin \beta + \frac{\delta m}{dt} \right) r_e \]  \[ (8) \]

2.2. Upshift with two clutches

As the downshift process is the opposite process of the upshift process, only the upshift process is studied. Taking the 1st gear to the 2nd gear as an example, the clutch CL is released, and the clutch CM is applied. Then, equations (2,3) should be changed into equations (9,10).

\[ I_1' \frac{d\omega_1}{dt} = M_i - M_{CL} - M_{CM} \]  \[ (9) \]
\[ I_2' \frac{d\omega_2}{dt} = M_{CL} i_{CL} + M_{CM} i_{CM} - M_{C1} \]  \[ (10) \]

Before the torque phase and after the inertia phase, the system torque transferred by CL and CM are illustrated in equations (11, 12), respectively. During the torque phase, the relationship between these clutches is described by the equation (13).

\[ M_{CL} = \frac{M_m I_v + M_{CL} I_1' i_{CL}}{I_1' i_{CL} + I_2'} \]  \[ (11) \]
\[ M_{CM} = \frac{M_m I_1' + M_{CM} I_1' i_{CM}}{I_1' i_{CM} + I_2'} \]  \[ (12) \]
\[ M_{CL} = \frac{M_m I_1' - M_{CM} (I_1' + I_1' i_{CM}) + M_{CL} i_{CL}}{I_1' i_{CL} + I_2'} \]  \[ (13) \]

2.3. Upshift with four clutches

For the gearshifts between 1st and 3rd gear, only two clutches are involved; while for the gearshift between 3rd and 4th gear, four clutches are involved. Then, equations (2-4) should be changed into equations (14-16).

\[ I_1' \frac{d\omega_1}{dt} = M_i - M_{CH} - M_{CL} \]  \[ (14) \]
\[ I_2' \frac{d\omega_2}{dt} = M_{CH} i_{CH} + M_{CL} i_{CL} - M_{C1} - M_{C2} \]  \[ (15) \]
$$I_3 \frac{d\omega_o}{dt} = M_{c1}i_{c1} + M_{c2}i_{c2} - M_o$$

Similarly, the torque relationship between four clutches are optimized, shown in equations (17,18).

$$M_{c1} = \frac{M_o(I_1^2i_{c1}^2 + I_3^2) + M_{c1}(I_1^2c_{1d}i_{c1}^2 + I_1^2i_{c1}^2) + M_{c1}i_{c1}(i_{c1} - i_{c2})i_{c1}i_{c2} + I_3^2}{I_1^2c_{1d}^2 + I_2^2i_{c1}^2 + I_3^2}$$

$$M_{c1} = \frac{M_o(I_1^2i_{c1}^2 + M_{c1}(I_1^2c_{1d}i_{c1}^2 + I_1^2i_{c1}^2) + M_{c1}i_{c1}(i_{c1} - i_{c2})i_{c1}i_{c2} + I_3^2)}{I_1^2c_{1d}^2 + I_2^2i_{c1}^2 + I_3^2}$$

For the engine torque, it is modeled as a mean value torque generator [5]. As for the friction torque generated by clutches, it is expressed as follows.

$$M_{Clutch} = \mu NP A_f \frac{2}{3} \left( \frac{R_o^3 - R_i^3}{R_o^3 - R_i^3} \right)$$

3. Lumped thermal resistance model

At present, the Finite Element Method (FEM) is the most widely used numerical calculation method for engineering analysis of thermal problems. However, if this method is used to predict and evaluate the dynamic temperature of the friction element in real time, there are shortcomings such as too complicated theory, huge computational workload, long calculation time and difficulty in programming. Therefore, this section builds a thermal resistance model based on the lumped parameter method to predict the average temperature rise of the wet clutches [9]. As shown in Figure 2, the entire cooling system is divided into different lumped elements, including the operating environment (E), the radiator (R), the hydraulic fluid reservoir (F), the relief safety valve (RV), the pump (P), the pressure regulating valve (V) and clutches.

![Figure 2. Lumped thermal resistance heat transfer model.](image)

Considering the heat transfer of the components between elements, the heat transfer between the lubricants in each element, the heat exchange between the components and the lubricating oil, and the heat generation of the components in each element, etc., the temperature variations of the component and the lubrication oil in each element can be deduced by the following equation.

$$\begin{align*}
C_{i} \frac{dT_i}{dt} &= \Phi_i - K_{i,j_0}(T_i - T_{j_0}) - \sum_j K_{i,j}(T_j - T_k) \\
C_{io} \frac{dT_{io}}{dt} &= \Phi_{i_0} - K_{i_0,j_0}(T_{i_0} - T_{j_0}) - \sum_j K_{i_0,j_0}(T_{i_0} - T_{j_0})
\end{align*}$$

with $C_{i} = \rho_{i} c_{i} \cdot V_{i}$ and $K$ is the equivalent heat transfer coefficient [10].
Taking the clutch CL as an example to investigate its temperature variation, the heat state equation of clutch CL can be further expressed as follows according to the equation (20).

\[
\begin{align*}
C_{CL} \frac{dT_{CL}}{dt} &= \dot{Q}_{CL} - K_{CL,CL}(T_{CL} - T_{E}) - K_{CL,CLo}(T_{CL} - T_{CLo}) \\
C_{CLo} \frac{dT_{CLo}}{dt} &= -K_{CLo,CLo}(T_{CLo} - T_{Vo}) - K_{CLo,Vo}(T_{CLo} - T_{Vo}) \\
&\quad - K_{CLo,CL}(T_{CLo} - T_{CL}) - K_{CLo,E}(T_{CLo} - T_{E})
\end{align*}
\]

(21)

It is amused that all the generated friction heat is absorbed by the lubrication oil and the heat flux can be described as follows.

\[
\dot{Q}_{CL} = M_{Clutch} \Delta \omega_{Clutch}
\]

(22)

4. Thermal condition simulation and analysis

With the established dynamic model to calculate the friction torque during the launch and shift process and the lumped thermal model to predict the temperature of wet clutches, the thermal conditions of different clutches can be simulated. In this paper, a 10km driving cycle, shown in Figure 3, is taken as an example to investigate the temperature variations of clutches.

![Figure 3. 10km driving cycle.](image)

![Figure 4. Dynamic temperature variation of clutches.](image)

The simulation results are shown in Figure 4. After the clutch slips, its temperature rises, then decreases due to the cooling of the lubricating oil. In comparison, CM and CH are used frequently, and each time there is a certain temperature rise during the friction process. However, since the time interval between shifts is relatively long and the cooling is enough, the temperature always varies at a low level. As for the CL, C1, and C2, although the slipping frequency is low, a large amount of friction work is generated, resulting into the high temperature rise. Particularly, during the launching process, the temperature of the clutch CL increases from 100°C to 123°C directly, suffering a great dynamic load. However, in general, when operating in accordance with the shift cycle, the temperature of each clutch changes between 100°C and 125°C, within the controllable range [9].
5. Conclusions
In this paper, the dynamic temperature variations of wet clutches are investigated for the DSG vehicle. To achieve this outcome, a detailed dynamic model is established to calculate the dynamic load of different clutches within a 10km driving cycle. Then, to predict the temperature of clutches, a lumped thermal model is developed. From simulation results, the clutch used for the launching process suffers a large heat flux and its temperature increase significantly. As for the clutches used for the shift process, the temperature varies wavily. While, on the whole, the temperature of clutches is under 125°C, satisfying the requirements of vehicle.

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Appendix

| Parameters | Meanings |
|------------|----------|
| A          | front area of vehicle |
| Af         | surface area of friction pairs |
| C_D        | drag coefficient |
| I          | moment of inertia of components |
| M          | torque transferred by components |
| N          | number of friction pairs |
| P          | pressure applied on clutch |
| R          | radius of friction disc |
| Q_CL       | generated heat flux in CL |
| S          | stiffness coefficient of the shaft |
| T          | temperature of components |
| V_s(o)     | volume |
| Q_s(o)     | heat source |

| Parameters | Meanings |
|------------|----------|
| θ          | rotation angle |
| f          | friction coefficient of road |
| g          | acceleration of gravity |
| i          | gear ratio |
| m          | vehicle weight |
| v_e        | vehicle speed |
| r_w        | radius of wheel |
| β          | road grade angle |
| μ          | friction coefficient |
| ω          | angular speed of components |
| C_i(o)     | specific heat capacity |
| ρ_i(o)     | density |
| δ          | increasing coefficient of mass |