The wear characteristics of wet clutches in the DSG vehicle in a 10km driving cycle

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Abstract. In order to investigate the wear characteristics of wet clutches in the DSG vehicle during shift process, this paper presents a dynamic analytical model and an experimental test of wet friction pair for simulation and analysis. A detailed dynamic model is established firstly to calculate the dynamic load of the clutches during the shift process. The wet friction pair experiment is conducted to explore the wear behaviours under different conditions. Then, the wear coefficient is deduced. The experimental results indicate that, when the temperature is constant, the slipping speed has little influence on the wear coefficient, while the contact pressure is on the contrary. Then, an empirical formula is obtained to describe the relationship between the wear coefficient and the contact pressure. Based on the simulation results, the wear amount of the clutches fixed on the input shaft is about 0.8mg, while for clutches on the middle shaft, it is only 0.2mg within a 10km driving cycle.

1. Introduction

The purpose of installing the transmission is to expand the limited performance of the internal combustion engine, so as to fulfill the propulsion demand for the normal daily driving in terms of the torque and speed. To obtain good dynamic performance and energy saving, many types of transmission have been developed, including manual transmissions, automated manual transmissions, automatic transmissions, continuous variable transmissions, and direct shift gearbox (DSG) [1]. DSG, which is also called DCT, can achieve high transmission efficiency and good shift quality. It is assumed that DSG is an important developing tendency in transmissions field in the near future [2].

Due to the special structure of DSG, the whole shift process can be divided into the torque phase and the inertia phase. During the torque phase, the engine torque is transferred from the off-going clutch to the on-coming clutch. A lot of researches have been done by Goetz M to analyse the dynamic characteristics of DSG during the torque phase [3]. Through establishing a detailed dynamic model of the DSG vehicle, a control strategy was proposed, and the simulation results indicated that a smooth gearshift during the torque phase was obtained. However, if both two clutches slipped during the torque phase, a certain power interruption or power circulation occurred. To address this problem, a torque based control strategy was put forward to complete the torque exchange process with only oncoming clutch slipping [4]. As for the inertia phase, feedback control and torque based logic control have been adopted to optimize the power-on upshift and power-off downshift process [5,6]. In recent paper, Guoqiang Li proposed an integral linear quadratic regulator for the inertia phase. By choosing the jerk level as the control objective, the simulation results showed that the jerk could be reduced during the entire shift process, indicating a good shift performance [1].

Owing to the valuable research, the DSG vehicle performance has been improved significantly. However, when it comes to the wear problems of the wet clutch, which may result in a reduction of service life and decrease of carrying capacity, it is seldom considered. To fill that gap in knowledge, the wear status of the DSG vehicle is investigated in a 10km driving cycle based on the wear characteristics of the wet clutch.

In the second section, the dynamic shift process of the DSG vehicle are studied. Then, in the third section, the wear characteristics of the wet clutch are investigated by a wet pin-on-disc experiment. As for the fourth section, the wear condition of different clutches in the DSG is simulated. Finally, the conclusions are presented.

2. Shift process of the DSG vehicle

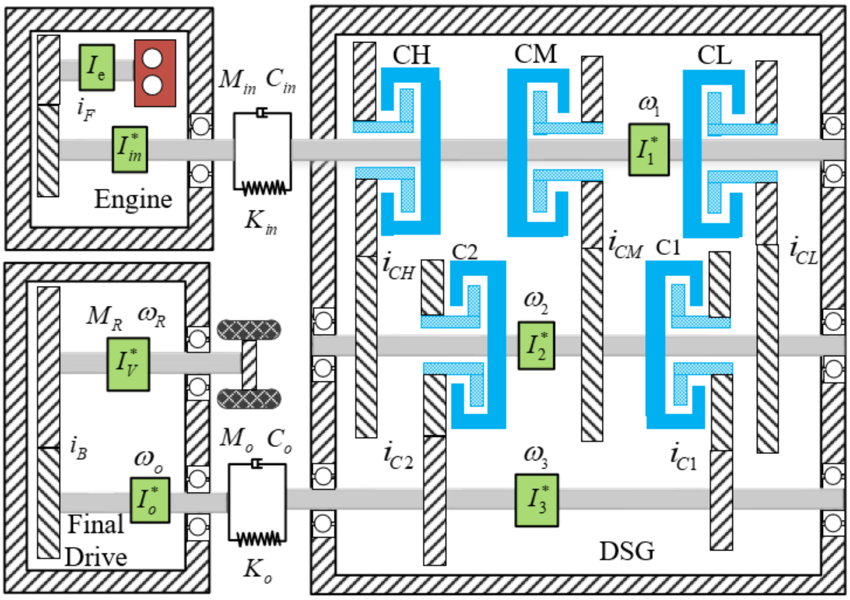

Figure 1. Powertrain system of the DSG vehicle

The simplified structure of the DSG is shown in Figure 1, consisted of five wet clutches and three shafts. Compared with the conventional DSG, the synchronizer is abandoned in this DSG, and, during the shift process, there is no pre-engagement process. By applying different clutches, various gear ratios can be achieved directly. To shift DSG, the leaving clutch is released and the coming clutch is engaged according to the Table 1, in which "●" represents that the corresponding clutch is engaged.

Table 1. Clutch schedule for each gear

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

From the 1st gear to the 3rd gear, two clutches are involved, while during the shift process between the 3rd gear and 4th gear, four clutches are involved simultaneously. As the downshift process can be considered as the opposite process of the upshift process, only the upshift process is analysed in this paper, including the 1st gear to the 2nd gear and the 3rd gear to the 4th gear.

2.1. Upshift with two clutches

During the upshift process from the 1st gear to the 2nd gear, the clutch CL is disengaged and the clutch CM is engaged. As shown in Figure 2, the upshift process goes through from the torque phase to the inertia phase and the engine power is transferred from CL to CM. According to the Newton second law, the system dynamic can be deduced as follows.

$$(I_e+\frac{I_{in}^*}{i_F})\frac{d\omega_e}{dt}=M_e-\frac{M_{in}}{i_F} \tag{1}$$

$$I_1^*\frac{d\omega_1}{dt}=M_{in}-M_{CL}-M_{CM} \tag{2}$$

$$I_2^*\frac{d\omega_2}{dt}=M_{CL}i_{CL}+M_{CM}i_{CM}-M_{C1} \tag{3}$$

$$I_3^*\frac{d\omega_3}{dt}=M_{C1}i_{C1}-M_{\mathrm{o}} \tag{4}$$

$$(I_o^*+\frac{I_V^*}{{i_B}^2})\frac{d\omega_o}{dt}=M_{\mathrm{o}}-\frac{M_{\mathrm{R}}}{i_B} \tag{5}$$

$$M_{in}=K_{in}(\theta_{in}-\theta_1)+C_{in}(\omega_{in}-\omega_1) \tag{6}$$

$$M_o=K_o(\theta_3-\theta_o)+C_o(\omega_3-\omega_o) \tag{7}$$

$$M_R=\left(\underbrace{\frac{1}{2}\rho_{air}C_DA{v_e}^2}_{\text{air resistance}}+\underbrace{mgf\cos\beta}_{\text{wheel resistance}}+\underbrace{mg\sin\beta}_{\text{gradient resistance}}+\underbrace{\delta m\frac{dv_e}{dt}}_{\text{acceleration resistance}}\right)r_w \tag{8}$$

During the torque phase, the relationship between these clutches is described by Equation (9).

$$M_{CL}=\frac{M_{in}I_2^*-M_{CM}(I_2^*+I_1^*i_{CL}i_{CM})+M_{C1}I_1^*i_{CL}}{I_1^*i_{CL}^2+I_2^*} \tag{9}$$

2.2. Upshift with four clutches

As mentioned above, during the shift process between 3rd gear and 4th gear, four clutches are involved. Thus, the Equations (2-4) should be changed into Equations (10-12).

$$I_1^*\frac{d\omega_1}{dt}=M_{in}-M_{CH}-M_{CL} \tag{10}$$

$$I_2^*\frac{d\omega_2}{dt}=M_{CH}i_{CH}+M_{CL}i_{CL}-M_{C1}-M_{C2} \tag{11}$$

$$I_3^*\frac{d\omega_3}{dt}=M_{C1}i_{C1}+M_{C2}i_{C2}-M_{\mathrm{o}} \tag{12}$$

Similarly, to transfer the engine torque fluently, the torque relationship between four clutches are optimized and is shown in Equation (13,14).

$$M_{CH}=\frac{M_{in}(I_2^*i_{C1}^2+I_3^*)+M_oI_1^*i_{CH}i_{C1}-M_{CL}(I_1^*i_{CH}i_{CL}i_{C1}^2+I_2^*i_{C1}^2+I_3^*)+M_{C2}I_1^*(i_{C1}-i_{C2})i_{C1}i_{CH}}{I_1^*i_{C1}^2i_{CH}^2+I_2^*i_{C1}^2+I_3^*} \tag{13}$$

$$M_{C1}=\frac{M_{in}I_3^*i_{CH}+M_o(I_1^*i_{CH}^2i_{C1}+I_2^*i_{C1})+M_{CL}I_3^*(i_{CL}-i_{CH})-M_{C2}[(I_1^*i_{CH}^2+I_2^*)i_{C1}i_{C2}+I_3^*]}{I_1^*i_{C1}^2i_{CH}^2+I_2^*i_{C1}^2+I_3^*} \tag{14}$$

The engine output torque, it is modeled as a mean value torque generator [4]. For the slipping clutches, the generated friction torque can be expressed as follows.

$$M_{\mathrm{Clutch}}=\mu NP\,A_f\,\frac{2}{3}\left(\frac{R_o^3-R_i^3}{R_o^2-R_i^2}\right) \tag{15}$$

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3. Wear characteristics of the wet friction pair

Figure 2 shows the friction and wear testing machine UMT Tribolab. Before the test, the pin plate was fully running-in. In the experiments, the following steps are carried out: (1) clean the lubricating oil pool and inject 30mL new lubricating oil; (2) heat the specimen to the target temperature; (3) adjust the motor to the target speed, and use the loading module to engage the friction pair; (4) separate the friction pair, collect all the lubricating oil and clean the test bench. Repeat the above steps according to the conditions listed in the Table 2. In DSG, the cooling system maintains the temperature of the entire transmission around 90℃~110℃, thus in experiments, the temperature is set as 90℃.

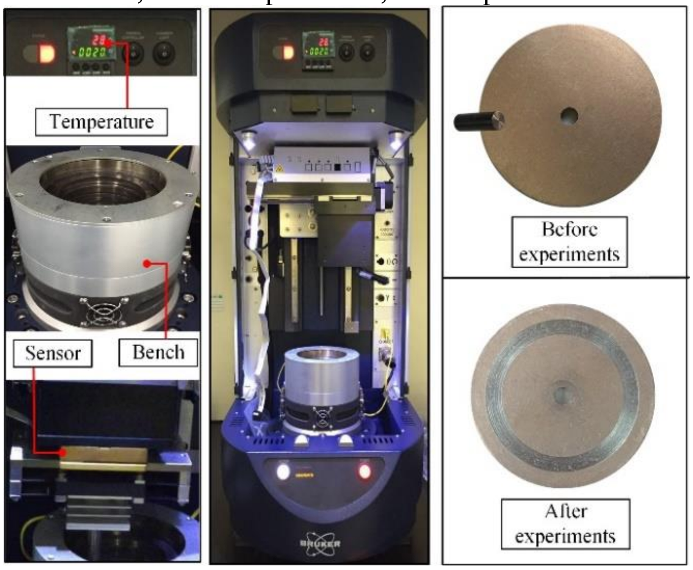

Figure 2. The high temperature tribometer UMT Tribolab

Table 2. Test conditions

Parameters	Value
Average interface contact pressure (MPa)	2.1,2.8,3.5,4.2,4.9
Velocity (mm/s)	1047, 1309, 1571, 1833, 2085

There is no copper element in the new lubricating oil before the slipping test, while, after each experiment, the copper-base powder material from the clutch will be absorbed by the lubricating oil. By measuring the composition of the copper element with the multielement oil analysis spectrometer, the wear amount of the wet friction pair can be deduced. After that, the Archard law is introduced and the wear coefficient of the wet friction pair can be further obtained.

$$K_{\delta} = \frac{\phi_{test}}{A_{test} p_c S} \tag{16}$$

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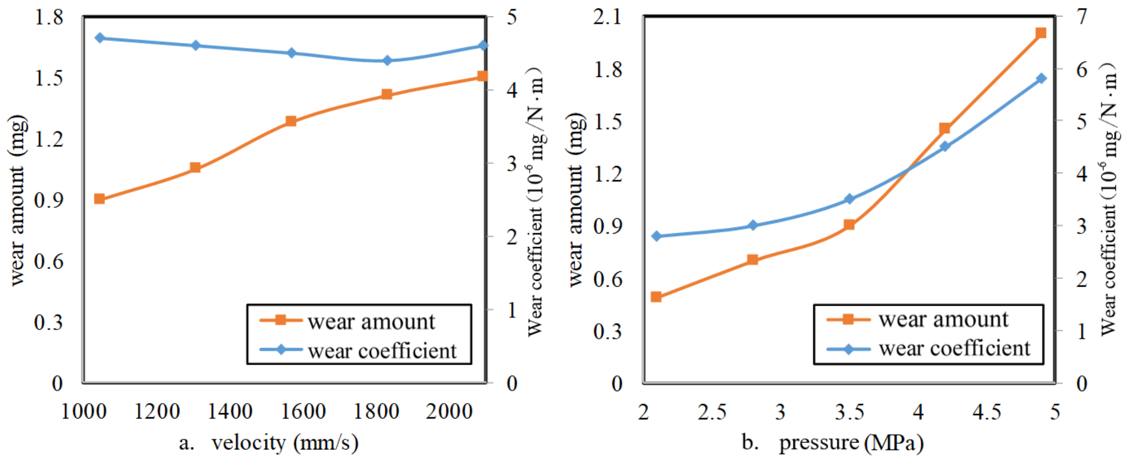

Figure 3. Effect of velocity and pressure on the wear coefficient

Figure 3 illustrates part of the obtained experimental results and the calculated wear coefficient. As shown in Figure 3.a, with the increased slipping speed, the wear amount increases significantly, while the wear coefficient has little fluctuation, even the slipping speed varies from 1047mm/s to 3927 mm/s. As for the influence of the engage pressure, it can be observed from the Figure 3.b. With the increase of the pressure from 2.1MPa to 4.9MPa, the wear amount increased from 2.8mg to 5.8mg. Furthermore, the wear coefficient shows an obvious growing up trend. According to the above analysis, it can be concluded that the slipping speed has little influence on the wear coefficient. Then, with the method of polynomial curve fitting, the relationship between the wear coefficient and the applied pressure for the wet friction pair is deduced as follows.

$$K_\delta = 0.1183P^3 - 0.8664P^2 + 2.5966P + 0.0054 \tag{17}$$

4. Wear condition analysis

With the established DSG shift model, the dynamic load of the wet clutch during the shift process can be calculated. Then, according to the Equation (23), the wear amount of different clutch can be further obtained. For the vehicle driven on the structured road, the new European driving cycle is always adopted to test the vehicle's performance. However, this DSG is equipped in a heavy tracked vehicle, which runs on the unstructured road. Thus, a 10km driving cycle is calculated according to the actual experimental data [7]. The obtained shift cycle is shown in Figure 4.

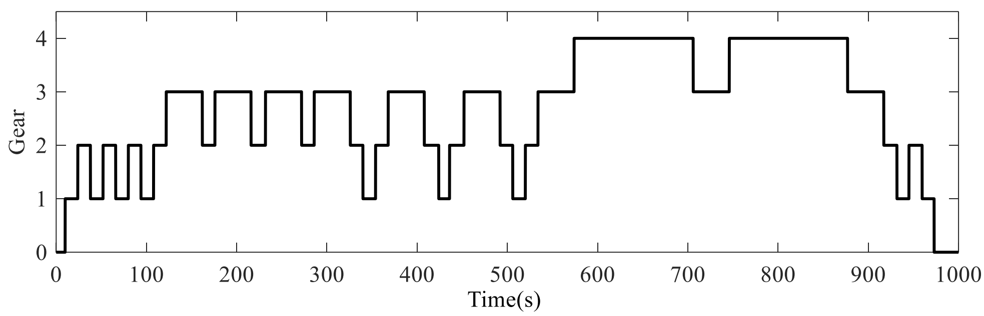

Figure 4. Shift cycle in a 10km driving cycle

With the 10km driving cycle, the simulation results of wear amount of each clutch is obtained, as shown in Figure 5. For CL, CM and CH clutches fixed on the input shaft, the wear amount is roughly the same, about 0.8mg. As for clutches C1 and C2, they are only utilized during the downshift process from the 4th gear to the 3rd gear, so the wear amount is 0.18mg and 0.22mg respectively. According to the material parameters of the friction plate used in DSG, when half of the friction material wears

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down, the quality is 42656mg. For CL, CM and CH, they can serve for 490,300 km, 618,200 km and 546,900 km respectively. For the vehicle driven on the unstructured road, the clutches meet the requirements in terms of the wear of friction plate.

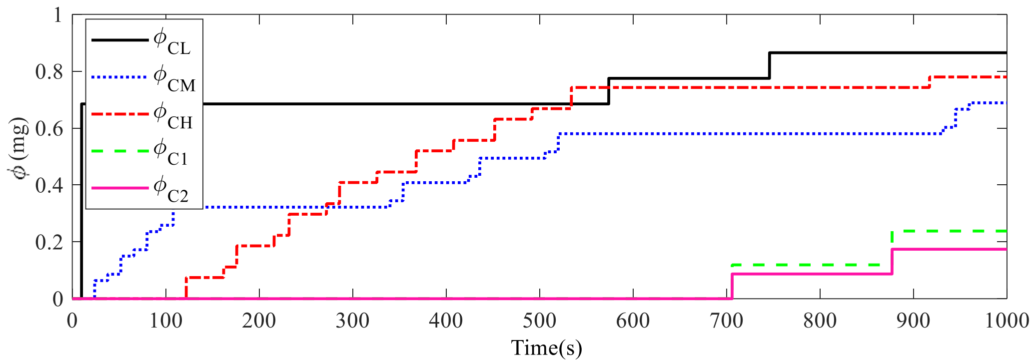

Figure 5. Wear amount of different clutches

5. Conclusions

In this paper, the wear characteristics of wet clutches in the DSG vehicle are investigated. To achieve this outcome, an integrated dynamic model is established for the DSG vehicle, describing the shift process with two and four clutches respectively, and the wet pin-on-disc experiment is conducted. On the basis of the experimental data, the slipping speed between the friction pairs has little influence on the wear coefficient; however, with the increase of the contact pressure, the wear coefficient also grows up and the relationship is described by an empirical formula. Within a 10km driving cycle, the wear amount of clutches on the input shaft is almost four times larger than that of clutches on the middle shaft. Even so, all the clutches can meet the requirements of the DSG vehicle in terms of friction wear.

Appendix

Parameters	Meanings	Parameters	Meanings
A	front area of vehicle	g	acceleration of gravity
A_f	surface area of friction pairs	i	gear ratio
A_{test}	contact area	m	vehicle weight
C_D	drag coefficient	p_c	Pressure applied on discs
I	moment of inertia of components	v_e	vehicle speed
K_δ	wear coefficient	r_w	radius of wheel
M	torque transferred by components	β	road grade angle
N	number of friction pairs	μ	friction coefficient
P	pressure applied on cluthes	ω	angular speed of components
R_i	inside radius of friction disc	ϕ	wear amount
R_o	outside radius of friction disc	ρ_{air}	density of air
S	Slipping distance		

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