Creeping Control Strategy for Direct Shift Gearbox Based on the Investigation of Temperature Variation of the Wet Clutch

Biao Ma, Jikai Liu, Man Chen, Jianpeng Wu, Liyong Wang, Changsong Zheng

Abstract—Proposing an appropriate control strategy is an effective and practical way to address the overheat problems of the wet multi-plate clutch in Direct Shift Gearbox under the long-time creeping condition. To do so, the temperature variation of the wet multi-plate clutch is investigated firstly by establishing a thermal resistance model for the gearbox cooling system. To calculate the generated heat flux and predict the clutch temperature precisely, the friction torque model is optimized by introducing an improved friction coefficient, which is related to the pressure, the relative speed and the temperature. After that, the heat transfer model and the reasonable friction torque model are employed by the vehicle powertrain model to construct a comprehensive co-simulation model for the Direct Shift Gearbox (DSG) vehicle. A creeping control strategy is then proposed and, to evaluate the vehicle performance, the safety temperature (250 ℃) is particularly adopted as an important metric. During the creeping process, the temperature of two clutches is always under the safety value (250 ℃), which demonstrates the effectiveness of the proposed control strategy in avoiding the thermal failures of clutches.

Keywords—Creeping control strategy, direct shift gearbox, temperature variation, wet clutch.

I. INTRODUCTION

DSG, sharing a similar work principle with Dual Clutch Transmission, is gaining more popularity among vehicle designers as a consequence of combining the advantages of manual and automatic transmissions in terms of gear efficiency and functionality. Beyond that, in contrast with the automatic mechanical transmission (AMT), DSG can achieve swiftly and seamlessly gearshifts without power interruption by the cooperation between two clutches [1]. Although DSG seems to have all these advantages, further enhancements are still required to enable DSG to possess the same or higher level of vehicle performances regarding ride comfort and drivability, compared with conventional transmissions. Thus, the dynamics and control of the clutch behaviors during the shift and launch processes become the key concerns in the development of DSG technology.

A typical gearshift process of DSG is accomplished by selectively engaging and disengaging clutches, proceeding

B. Ma, J. Liu, J. Wu, and C. Zheng are with Beijing Institute of Technology, Beijing, CO 100081 China (e-mail: mabiao@bit.edu.cn, jackylauno1@gmail.com, 15811319103@163.com, zhengchangsong@bit.edu.cn).
M. Chen is with Beijing Institute of Technology, Beijing, CO 100081 China (corresponding author, phone: +86-01068918637; e-mail: turb911@bit.edu.cn).
L. Wang is with Beijing Information Science and Technology University, Beijing, CO 100192 China (e-mail: wangliyong@bistu.edu.cn)

through the torque phase and the inertia phase. During the torque phase, the engine torque is transferred between two clutches and the timing to release and engage them is of vital importance to achieve a smooth shift without power interruption and circulation [2], [3]. As for the inertia phase, due to the synchronization of the on-coming clutch, not only the torque but also rotational speeds change intensively, so the shift quality has a strong relationship with this duration and much attention has been paid to this phase [4], [5]. When it comes to the launch process, DSG has a similar working process with AMT under the condition of launching with only one clutch involved. However, launching with only one clutch is not sufficient to utilize the advantages of DSG. Thus, an optimal torque coordinating control with two clutches simultaneously involved is proposed by Zhao [6]. Both simulation results and experiment data indicate that the proposed launching control strategy can not only reflect the driver's intention effectively, but also achieve excellent launching qualities.

Owing to the considerable research work, valuable technical references with regard to modelling, simulation and experiment are created and, under most conditions, the DSG vehicle can acquire excellent performance. However, when it comes to the thermal problem of the clutch, which may result in large clutch wear amount, reduction of service life and decrease of carrying capacity, it is seldom considered by the current proposed control strategies. The purpose of this paper is to fill that gap in knowledge, taking the creeping condition as an example and proposing a control strategy based on characteristics of the temperature variation of the clutch.

II. DSG STRUCTURE AND THERMAL PROBLEM

The schematic diagram of the DSG vehicle, including the hydraulic system, is shown in Fig. 1. Two clutches are mounted between the engine and the input shaft of the transmission respectively. Under the control of the hydraulic system and return springs, the corresponding clutch is engaged or disengaged. When the clutch CL is engaged and the clutch CH disengaged, the vehicle operates in low gears; on the contrary, the vehicle is driven in high gears. In addition to control the oil pressure, the hydraulic system also provides the clutches with the lubrication (cooling) oil to dissipate the heat generated during the slipping process. Generally, the temperature rise of two clutches is under control during the normal shifting or launching processes, when the slipping time lasts from 0.5 s to 2.0 s. However, if there is a traffic jam, the DSG vehicle has to creep at low speed, with a gear ratio even larger than the first

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gear. Under this condition, due to the lack of torque convertor, two clutches slip to drive the vehicle. Usually, the creeping process lasts for more than 5.0 s, depending on the road condition. Consequently, great thermal power generates in plates, which may cause thermal failures of the clutch. To avoid

the overheat condition and obtain a comfortable driving experience, it is necessary to study the temperature variations in clutches and propose an appropriate control strategy for the creeping process.
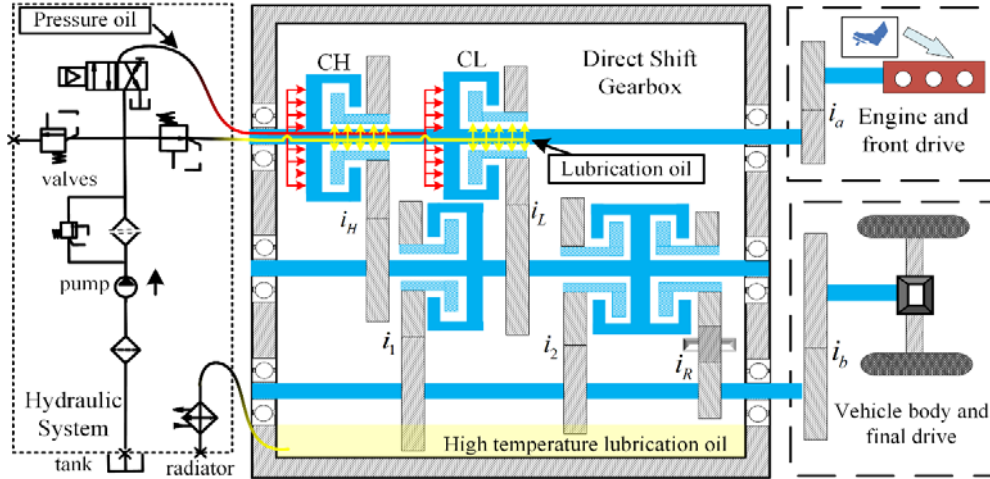

Fig. 1 Schematic diagram of the DSG vehicle

III. Friction Model and Temperature Variation of the Clutch System

The premises of proposing a creeping control strategy is to predict the temperature variation of the clutches precisely during the slipping process. Moreover, as the thermal power is generated from the slipping plates of the clutch, an accurate description of the transferred friction torque will be beneficial to reduce calculation errors effectively. To achieve the goal of predicting the clutch temperature, a thermal resistance model for the DSG cooling system is established by introducing the lumped parameter method.

A. Lumped Thermal Resistance Heat Transfer Model

As shown in Fig. 2, the entire DSG cooling system is divided into a number of lumped elements geometrically, including the operating environment (E), clutches (CH and CL), the radiator (R), the hydraulic fluid reservoir (F), the relief safety valve (RV), the pump (P) and the pressure regulating valve (V).
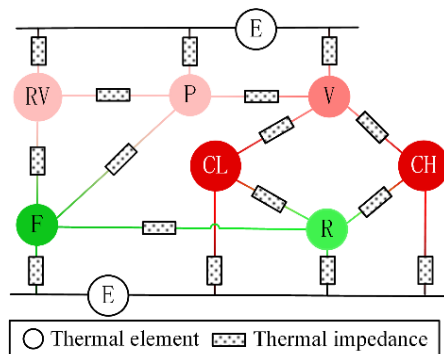

Fig. 2 Lumped heat transfer model

In each element, there is a component to store the thermal

power, and the inner lubrication oil has heat generation and interconnections to neighboring elements through a linear mesh of thermal impedances. For the component (subscript i) and the lubrication oil (subscript i_o) in each element, the heat status can be obtained according to the traditional heat conduction equations, expressed as:

$$\begin{cases} c_i \rho_i \dfrac{\partial T_i}{\partial t} = \lambda_i \nabla^2 T_i + \dot{\Phi}_i \\ c_{io} \rho_{io} \left(\dfrac{\partial T_{io}}{\partial t} + q_{io} \nabla T_{io} \right) = \lambda_{io} \nabla^2 T_{io} + \dot{\Phi}_{io} \end{cases} \quad (1)$$

Then, the temperature variations of the component and the lubrication oil can be deduced and described by (2).

$$\begin{cases} C_i \dfrac{dT_i}{dt} = \dot{Q}_i - K_{i,io}(T_i - T_{io}) - K_{i,e}(T_i - T_E) \\ C_{io} \dfrac{dT_{io}}{dt} = \dot{Q}_{io} - K_{io,i}(T_{io} - T_i) - \sum_j K_{io,jo}(T_{io} - T_{jo}) \end{cases} \quad (2)$$

with $C_{i(o)} = c_{i(o)} \rho_{i(o)} V_{i(o)}$. $K_{i(o),j(o)}$ is the equivalent heat transfer coefficient and the methods to calculate its value can be found in [7].

As the paper focuses on the temperature variations of clutches, the clutch CL is investigated as an example. As the friction plate and the separator plate rotate with different speed during the creeping process, friction heat is generated. Meanwhile, the lubrication oil flows through the grooves to cool the plates. Naturally, the generated heat transfers between the plates and the oil. Based on (2), the heat state equation for the element CL can be expressed as.

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$$\begin{cases} C_{CL}\dfrac{dT_{CL}}{dt} = \dot{Q}_{CL} - K_{CL,E}(T_{CL}-T_E) - K_{CL,CLo}(T_{CL}-T_{CLo}) \\ C_{CLo}\dfrac{dT_{CLo}}{dt} = \dot{Q}_{CLo} - K_{CLo,Vo}(T_{CLo}-T_{Vo}) - K_{CLo,Ro}(T_{CLo}-T_{Ro}) \\ \qquad\qquad - K_{CLo,CL}(T_{CLo}-T_{CL}) - K_{CLo,E}(T_{CLo}-T_E) \end{cases} \quad (3)$$

For simplicity, it is assumed that all the generated friction power transfers into heat without any loss during the creeping process and the heat flux is described by (4). Also the throttle loss resulted from the differential pressure between the inner and outer sides of the clutch is illustrated below:

$$\begin{cases} \dot{Q}_{CL} = M_{CL}\Delta\omega_{CL} \\ \dot{Q}_{CLo} = q_r\Delta p_{CL} \end{cases} \quad (4)$$

As for the other elements, heat transfer equations can be deduced in a similar way.

B. Improved Friction Torque Model

In DSG, the primary heat producing elements are the clutches, determining the temperature rise of the entire hydraulic system. From (4), the factor, which has a vital important influence on the generated heat flux, is the friction torque transferred by the clutch. To simulate the temperature variation precisely, an effective the friction torque model, capable of characterizing the transferred friction torque by the wet multi-plate clutch under the slipping condition, should be guaranteed, as described in (5).

$$M_{CL} = \mu N P_{CL} A_f \frac{2}{3}\left(\frac{R_o^3 - R_i^3}{R_o^2 - R_i^2}\right) \quad (5)$$

Currently, in most developed friction models, the friction coefficient changes depending on the clutch slipping speed or is just considered as a constant value, without considering the influence of the temperature. However, under the creeping condition, the clutch temperature always varies drastically and the simplified friction coefficient is not sufficient to describe the transmitted friction torque and the generated heat flux precisely. In this paper, an improved friction coefficient model, fitted based on a large number of experimental data, is adopted.

$$\mu = 23e^{\left(\frac{-0.028574n}{(\ln T - 3.2)\left((28.3P)^{0.4}-0.87\right)}-5.16\right)} + \frac{0.01\ln(0.04398n+1)}{e^{0.005T}}$$
$$+0.08\left(e^{-0.005T}-1\right)\left(e^{-0.0022n}-1\right) - 0.005\ln(28.3P) + 0.035 \quad (6)$$

C. Model Validation

From (5), after the structure of clutch is determined, the parameters, which have an obvious influence on the thermal status of the clutch, are the pressure and the relative speed. Considering the actual working load of the clutches and the limitation of the measuring range and accuracy of sensors, four different conditions are conducted in experiments, including 0.2 MPa, 150 r/min; 0.2 MPa, 300 r/min; 0.3 MPa, 150 r/min and 0.3 MPa, 300 r/min. The slipping time is set as 100 s to simulate the thermal status during the long-time creeping process. After that, the simulation and test results are presented in Figs. 3-5 and the subscripts 1-4 represent the above four conditions respectively.

As shown in Fig. 3, the temperature profiles obtained from the simulations (solid lines) and test (dotted lines) show a similar increasing trend during the entire slipping period. Under all four conditions, the calculated temperature rise are 21 °C, 47 °C, 33 °C and 74 °C respectively after 100 s slipping, which are 1~4 °C higher than those obtained from the experiments. These differences resulted due to the fact that, in simulations, the generated friction work is fully converted into heat flux to increase the clutch temperature, which is not the real case.
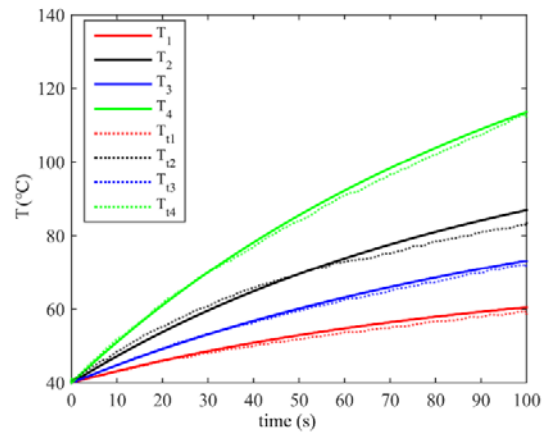

Fig. 3 Temperature variations

As for the torque profiles shown in Fig. 4, although the pressure and the relative speed are constant, the value of the transferred torque increases continuously. Taking the condition of 0.3 MPa, 300 r/min as an example, the friction torque varies from 149.5 Nm to 164.8 Nm, increased by 10.2%. This increase is caused by the change of the friction coefficient, as shown in Fig. 5. With the rise of the temperature from 40 °C to 114 °C, the friction coefficient varies from 0.054 to 0.060, demonstrating the great influence of the temperature.
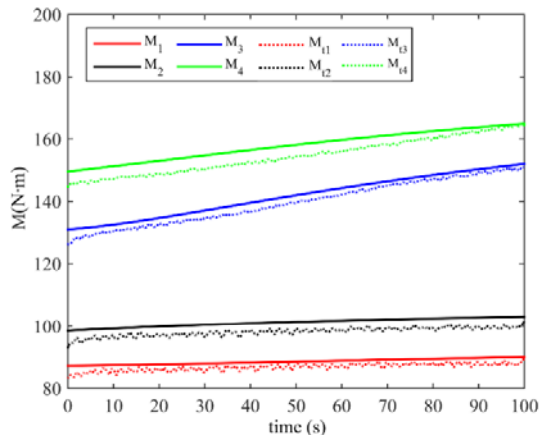

Fig. 4 Torque variations

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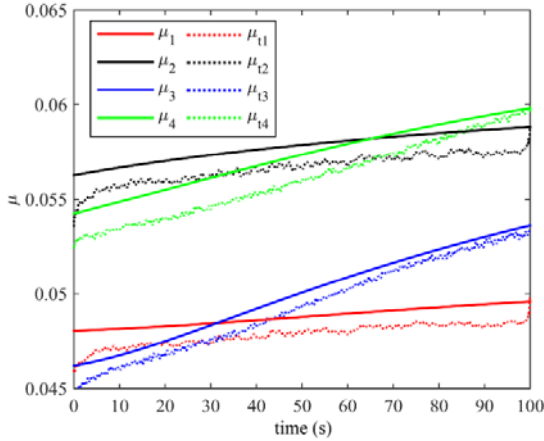Vol:13, No:4, 2019

Fig. 5 Friction coefficient variations

Comparing the simulation results and the test data in Figs. 3 and 4, even with the optimization of the friction coefficient, there still exists a certain difference, resulted from the drag torque, resistance of roller bearing and other factors. However,

on the whole, these errors are allowable and within the engineering permissible error range. Thus, it can be concluded that, the heat transfer model and the friction torque model are capable of predicting the temperature variation and simulating the transferred friction torque during the clutch slipping process.

IV. POWERTRAIN MODEL OF THE DSG VEHICLE

As the DSG creeping process is not a constant dynamic process, it is not sufficient to simulate the temperature variations with only the heat transfer model. To calculate the transferred friction torque real-timely, a powertrain model of the DSG vehicle is further developed. Following the traditional methods, the powertrain of the heavy duty vehicle equipped with DSG is modeled as lumped inertia elements connected with corresponding stiffness and damping components, shown in Fig. 6. When DSG creeps with slipping CL, the powertrain equations of motion are deduced as follows.
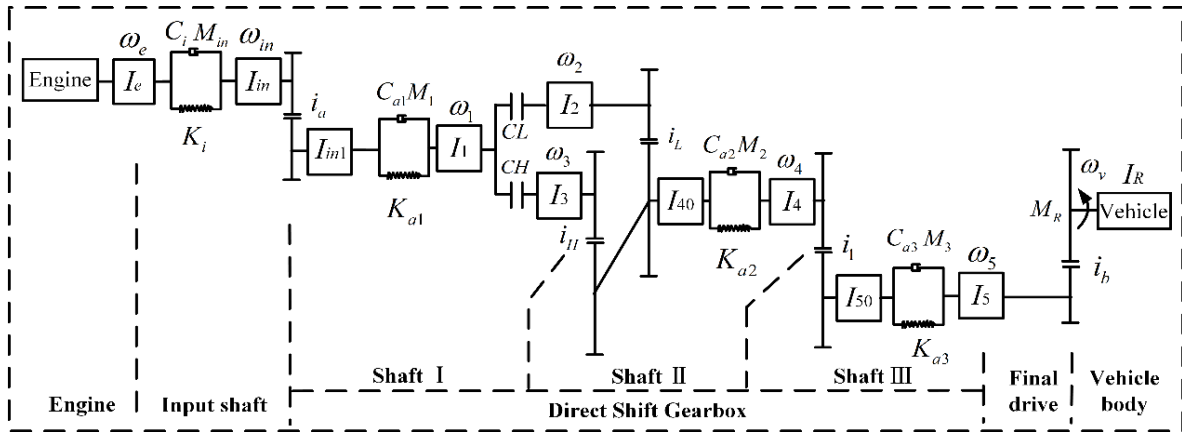

Fig. 6 DSG powertrain system

$$I_e \dot{\omega}_e = M_e \eta_e - M_{\text{in}} \tag{7}$$

$$(I_{\text{in}} + \frac{I_{\text{in}1}}{i_1^2})\dot{\omega}_{\text{in}} = M_{\text{in}} - \frac{M_1}{i_a \eta_a} \tag{8}$$

$$I_1 \dot{\omega}_1 = M_1 - M_{\text{CL}} \tag{9}$$

$$(I_2 + \frac{I_{40}}{i_L^2})\dot{\omega}_2 = M_{\text{CL}} - \frac{M_2}{i_L \eta_L} \tag{10}$$

$$(I_4 + \frac{I_{50}}{i_1^2})\dot{\omega}_4 = M_2 - \frac{M_3}{i_1 \eta_1} \tag{11}$$

$$(I_5 + \frac{I_R}{i_b^2})\dot{\omega}_5 = M_3 - \frac{M_R}{i_b \eta_b} \tag{12}$$

$$M_R = \left(\underbrace{\frac{1}{2}\rho_{air} C_D A v_e^2}_{\text{air resistance}} + \underbrace{mgf \cos\beta}_{\text{wheel resistance}} + \underbrace{mg\sin\beta}_{\text{gradient resistance}} + \underbrace{\delta m \frac{dv_e}{dt}}_{\text{acceleration resistance}} \right) r_w \tag{13}$$

$$M_{\text{in}} = K_{\text{in}}(\theta_e - \theta_{\text{in}}) + C_{\text{in}}(\omega_e - \omega_{\text{in}}) \tag{14}$$

$$M_1 = K_{a1}(\frac{\theta_{\text{in}}}{i_a} - \theta_1) + C_{a1}(\frac{\omega_{\text{in}}}{i_a} - \omega_1) \tag{15}$$

$$M_2 = K_{a2}(\frac{\theta_2}{i_L} - \theta_4) + C_{a2}(\frac{\omega_2}{i_L} - \omega_4) \tag{16}$$

$$M_3 = K_{a3}(\frac{\theta_4}{i_1} - \theta_5) + C_{a3}(\frac{\omega_4}{i_1} - \omega_5) \tag{17}$$

V. CREEPING CONTROL STRATEGY AND SIMULATION RESULTS DISCUSSION

A. DSG Creeping Control Strategy

From the identification of the driver's intention; engagement of the on-coming clutch; the temperature prediction of the slipping clutch and then the shift process between two clutches, the creeping process can be considered as a logical combination of a series of independent events. As the paper focuses on the temperature variation of the clutch system during the creeping

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process, it is assumed that the DSG starts immediately to track the desired vehicle speed after the transmission control unit makes the creeping decision. The architecture of the proposed control strategy is described in Fig. 7. It is rather remarkable that the safety temperature of clutch is set as 250 °C.
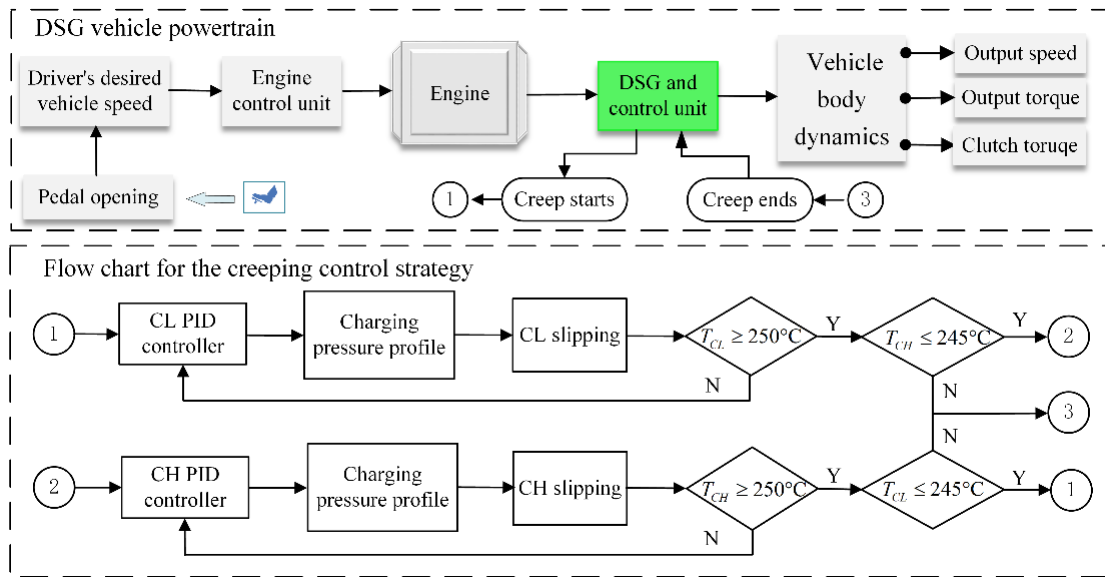

Fig. 7 The architecture of the DSG creeping control strategy
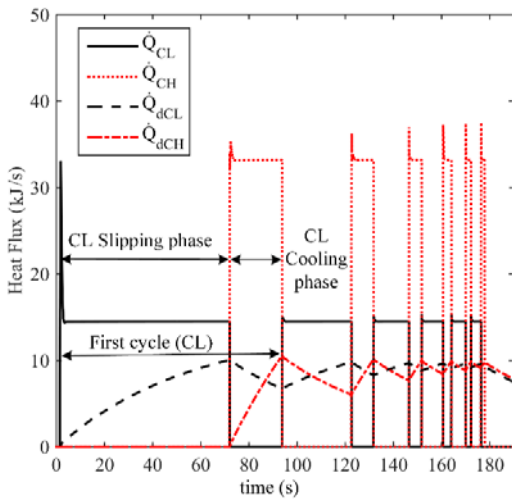
B. Simulation Results Discussion

Fig. 8 Heat flux conditions in two clutches

During the creeping process, the heat flux conditions and the temperature variations of two clutches are illustrated in Figs. 8 and 9. From Fig. 8, it can be seen that, the convective losses cannot dissipate the thermal energy generated during the slipping phase completely for both two clutches. Taking CL as an example, the dissipated heat flux is always smaller than the generated during the slipping phase. Thus the CL temperature increases from 100 °C to 250 °C gradually (Fig. 9). Then, according to the control strategy, CL is disengaged and CH is engaged. During the CH slipping phase, the lubrication oil continues cooling CL. However, as shown in Fig. 9, when T_{CH} increases to the safety temperature, T_{CL} only decreases to 205.5 °C, instead of the initial temperature (100 °C). After that, CL is

engaged again and CH is released. Due to the accumulated thermal power in clutches, the DSG vehicle cannot creep uninterruptedly and, after 174 s creeping, the temperature of the on-coming clutch (CL) reaches to 246 °C. Based on the proposed control strategy, the DSG vehicle stops to protect the clutches from overheat and the creeping process ends.
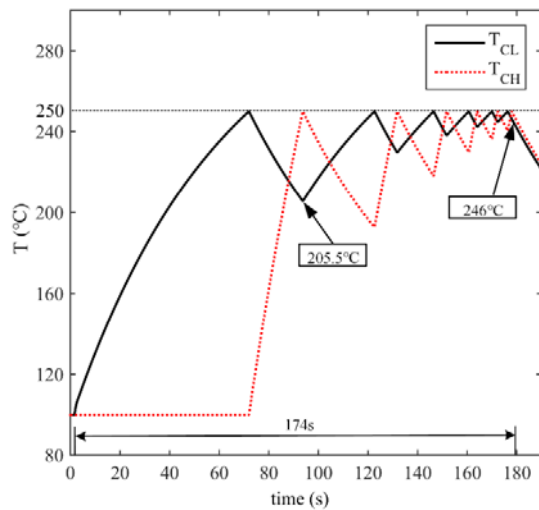

Fig. 9 Temperature variations of two clutches

As shown in Fig. 10, the changing trend of the friction coefficient of two clutches is presented. It is rather remarkable that, during each slipping period, the friction coefficient indicates an increasing trend. Taking μ_{CH} as an example, its value varies from 0.063 to 0.075 (19%) gradually during the first slipping period. This dramatic increase is mainly caused by the rise of the temperature, because the relative speed of CH

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remains unchanged during the slipping phase and the pressure has little influence on the friction coefficient. Under this condition, if P_{CH} remains unchanged, the output torque will definitely increase to accelerate the vehicle. Thus, to avoid the effect of the temperature and obtain a constant output torque, the designed PID controller plays its important role to decrease the pressure applied on clutches. As shown in Fig. 24, P_{CH} changes from 0.75 MPa to 0.65 MPa, decreased by 13.3%. All these changes in the pressure profiles and the friction coefficient indicate the great influence of the temperature on the dynamic process powerfully, when DSG operates in extreme conditions.
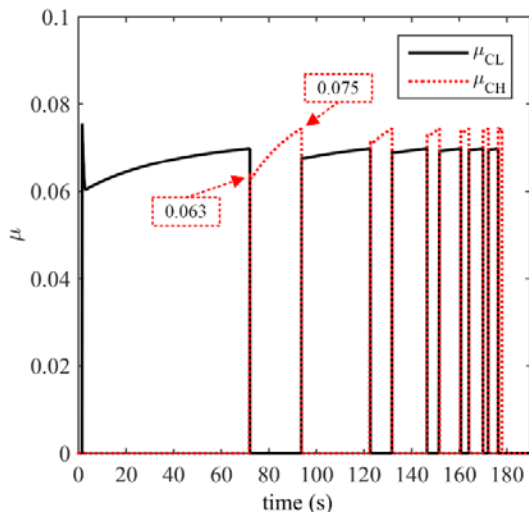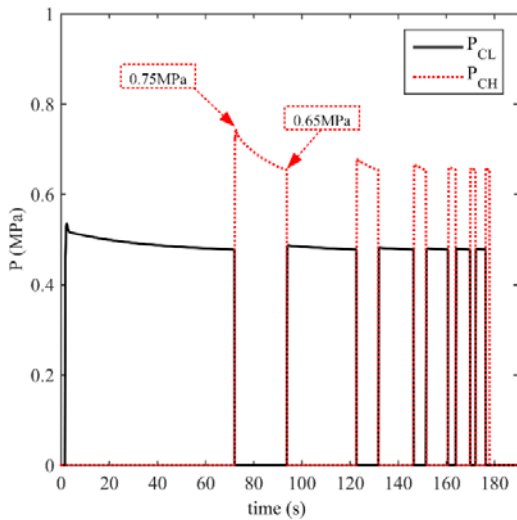

Fig. 10 Variations of friction coefficient

Fig. 11 Pressure profiles

Throughout the obtained simulation results during the entire creeping process, the DSG vehicle is under the control of the proposed strategy to slip two clutches by turns to extend the creeping time as long as possible. The temperature of two clutches is always under the safety value. Although the friction coefficient varies significantly due to the temperature variations of clutches, the designed controllers are sufficient to

adjust the pressure profiles to track the desired vehicle speed ideally, shown in Fig. 12. Thus, in summary, the proposed control strategy is validated to be capable of achieving a high-quality DSG creeping process.
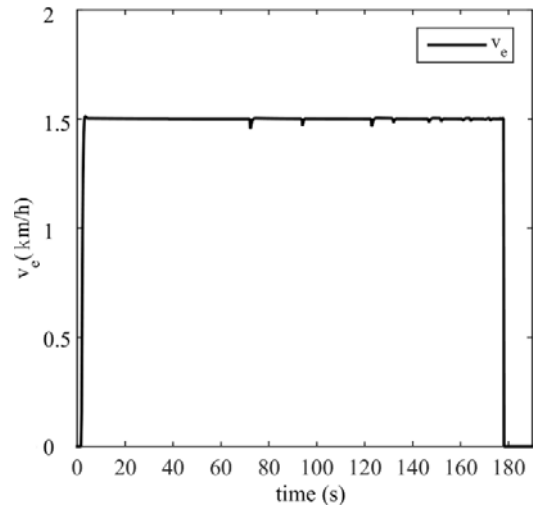

Fig. 12 Vehicle speed

VI. CONCLUSION

The purpose of this paper is to propose a specific control strategy for the DSG creeping process to avoid the thermal failures in clutches. To achieve this outcome, a thermal resistance model is developed for the DSG cooling system to predict the temperature variation of the wet multi-plate clutch under the slipping condition. Furthermore, a friction torque model, optimized with an improved friction coefficient, is developed to ensure the precision of this model. After the development of the dynamic powertrain model the DSG vehicle, an integrated co-simulation model, is presented to investigate the dynamic responses and predict the temperature variations of two clutches during the creeping process.

With the application of the strategy on the evolved co-simulation model, the simulation results indicate that the temperature of two clutches is always under control, which means that the overheat condition and thermal failures can be avoided effectively. With two clutches slipping alternatively, the DSG vehicle can be under the creeping status for 174 s uninterruptly, with CH slipping for 130.5 s and CH slipping for 43.5 s. In summary, results achieved from both the experiments and simulations demonstrate the potential of the developed models in addressing the thermal problem of clutches and, with the application of the proposed control strategy on actual cars, including but not limited to the creeping condition, the extension of the service life of friction components can be anticipated.

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APPENDIX

TABLE I
MAIN PARAMETERS FOR SIMULATION

Symbol	Quantity	Value
A (m^2)	front area of vehicle	3.6
A_f(m^2)	surface area of the piston	0.0427
C_D	drag coefficient	0.62
C_{a1}(N·m·s/rad)	damping coefficient of shaft I	2
C_{a2}(N·m·s/rad)	damping coefficient of shaft II	0.5
C_{a3}(N·m·s/rad)	damping coefficient of shaft III	0.5
C_{in}(N·m·s/rad)	damping coefficient of input shaft	0.5
I_R(kg·m^2)	moment of inertia of vehicle body	18.9021
I_e(kg·m^2)	moment of inertia of the engine	3.65
I_{in}(kg·m^2)	moment of inertia of the engine output shaft	0.01637
I_{in1}(kg·m^2)	moment of inertia of the input shaft	0.01538
I_1(kg·m^2)	moment of inertia of the shaft I	0.2102
I_2(kg·m^2)	moment of inertia of the CL	0.002373
I_3(kg·m^2)	moment of inertia of the CH	0.002373
I_4(kg·m^2)	moment of inertia of the shaft II	0.2034
I_{40}(kg·m^2)	moment of inertia of the gears on the shaft I	0.01931
I_{50}(kg·m^2)	moment of inertia of the gears on the shaft III	0.001535
I_5(kg·m^2)	moment of inertia of the shaft III	0.2523
K_{a1}(N·m/rad)	stiffness coefficient of the shaft I	89389
K_{a2}(N·m/rad)	stiffness coefficient of the shaft II	110088
K_{a3}(N·m/rad)	stiffness coefficient of the shaft III	65114
K_{in}(N·m/rad)	stiffness coefficient of the input shaft	283429
R_i(m)	inside radius of friction disc	0.086
R_o(m)	outside radius of friction disc	0.125
f	friction coefficient of road	0.123
i_a	gear ration (front drive)	5.3585
i_b	gear ration (final drive)	3.818
i_L	gear ration (clutch CL)	1.59
i_H	gear ration (clutch CH)	0.92
i_1	gear ratio (shaft III)	0.925
m (kg)	vehicle weight	7840
β(rad)	road grade angle	0.174
δ	increasing coefficient of mass	1.0554
r_w(m)	radius of wheel	0.1935

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