Investigation on the thermal problems of wet clutches for the tracked DCT vehicle during the launching process

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Abstract. Tracked vehicles usually travel on unstructured roads, and the transmission system bears heavy impact load. As one of the most vulnerable parts, wet clutches slip to transfer the engine power. To investigate the thermal condition of these wearing components, a dynamic system is built firstly to describe the launching process of the tracked vehicle. After that, a thermal resistance model is established for the gearbox cooling system based on the lumped parameter method. Combing these two models, a coupled model is proposed for the tracked DCT vehicle. From the simulation results, under the condition of launching with one clutch, the temperature rise is 42°C; while, when two clutches involved, the temperature is 21°C and 14°C respectively, which means the mitigation of the thermal problem. In addition, compared with the condition of launching with one clutch, the shock load decreased from 2250 N·m to 1745 N·m and the launching time is reduced by 1.0s by launching with two clutches.

1. Introduction
To increase ride comfort and fuel efficiency, the automotive industry and the researchers have been focusing on improving the performance of the transmissions. In recent years, Dual Clutch Transmission (DCT) is gaining more attention among the designers. Compared with the automatic mechanical transmission, DCT can achieve the shift process without power interruption by the cooperation between two clutches [1]. In addition, DCT poses a same level of vehicle performances regarding ride comfort and drivability, compared with conventional manual transmissions [2]. However, for this kind of transmission, further studies are still required to enhance its performance, particularly during the shifting and launching process.

Goetz [3-5] propose a number of examples of simulation models for the DCT vehicles to study the shift process. The proposed gearshift controller consists of various control loops, and the simulation results indicate that the engine and the clutch can achieve smooth synchronization without excessive vehicle jerk, clutch wear and loss in traction. However, Goetz mainly focuses on the timing to apply clutches, the relationship between two clutches is ignored. In [6], Liu propose a control strategy based on the analysis of the power flow to achieve a smooth shift without power interruption and circulation. As for the launching process, if DCT launches with one clutch, it is much similar to the launching process in AMT vehicles. However, only one clutch engaged during the launching process is not sufficient to utilize the advantages of DCT. Zhao [7] proposed an optimal torque coordinated control strategy to solve the problem of launching with twin clutches simultaneously based on the minimum value principle. From the simulation results, the proposed launching control strategy not only can effectively reflect the driver’s intention and extend the life span of twin clutches, but also obtain an excellent launching quality.
Thanks to these considerable researches, the performance of the DCT vehicle has been improved significantly. However, in practice, DCT always encounters various problems, especially for the tracked vehicle. This kind of vehicle always operates on the unstructured roads and, compared with the automobile, its transmission system suffers higher load. Finally, the clutches bear huge friction work, leading to reduction of service life and decrease of carrying capacity. Zagrodzki [8] studies transient thermomechanical phenomena occurring in a pack of friction discs of a multidisc wet clutch or a brake. As for the dry clutch, Pisaturo [9] and Ye [10] focus on the heat transfer process through finite element method. When it comes to the mechanism of temperature rise, Jen [11] analysis a wet-disk clutch subjected to a constant energy engagement. However, the engagement process they focused on are limited, as the input energy are uniform and not in accordance with the actual condition. To investigate the thermal conditions in clutches during the operating process, this paper proposes and coupled model by integrating the dynamic model and the temperature prediction model, taking the launching process as an example.

2. Dynamic model of the tracked DCT vehicle

In order to obtain the real-time dynamic load of the wet clutches during the launching process, it is necessary to build a dynamic model of the heavy-duty tracked vehicle equipped with the DCT. The simplified diagram of the DCT vehicle is shown in Figure 1. The transmission system consists of the engine, input shaft, DCT, final drive and the vehicle body.

![Figure 1. DCT powertrain system.](image)

According to the simplified diagram of vehicle transmission system and Newton's second theorem, the dynamic equation of DCT vehicle during the launching process can be obtained:

\[ I_c \dot{\omega}_c = M_c \eta_c - M_{in} \]  
\[ (I_m + \frac{I_{m1}}{i_1^2} \omega_m) \dot{\omega}_m = M_m - \frac{M_1}{i_1 \eta_1} \]  
\[ I_1 \dot{\omega}_1 = M_1 - M_{cl} \]  
\[ (I_2 + \frac{I_{m2}}{i_2^2} \omega_2) \dot{\omega}_2 = M_{cl} - \frac{M_2}{i_2 \eta_2} \]  
\[ (I_3 + \frac{I_{m3}}{i_3^2} \omega_3) \dot{\omega}_3 = M_{cl} - M_3 \]  
\[ (I_4 + \frac{I_{m4}}{i_4^2} \omega_4) \dot{\omega}_4 = M_3 - \frac{M_R}{i_4 \eta_4} \]  
\[ (I_5 + \frac{I_{m5}}{i_5^2} \omega_5) \dot{\omega}_5 = M_3 - \frac{M_R}{i_5 \eta_5} \]
where $\eta_X$ is the efficiency of the track, and its value can be calculated by the following empirical equation.

$$\eta_X = 0.95 - 0.0017v_e$$

(12)

$$v_e = \frac{\alpha_e r_w}{i_o}$$

(13)

As mentioned above, if only one clutch engages during the launching process, the advantage of the DCT is not utilized. To launch quickly, two clutches can slip together to transfer the engine power. Then, the equation (4) should be changed into the equation (14).

$$\left( I_2 + \frac{I_{40}}{i_L^2} \right) \dot{\theta}_2 = M_{CL} + M_{CH} \frac{M_2}{i_L^2 \eta_L}$$

(14)

If the vehicle starts with the 1st gear as the target gear, after the active and passive sides of the clutch CL synchronized, according to the above equations, the inertia torque transmitted by the clutch CL can be described by the equation (15). While, under the slipping condition, the friction torque generated by the clutch can be calculated by the equation (16).

$$M_{CL} = \frac{M_1(I_2 + \frac{I_{40}}{i_L^2}) + M_2I_1}{I_1 + I_2 + \frac{I_{40}}{i_L^2}}$$

(15)

$$M_{Clutch} = \mu NPA \left( R_1^3 - R_2^3 \right)$$

(16)

In the previous research work, the friction coefficient is described by the slipping speed between the input and output side of the clutch. However, in actual, the temperature of the clutch always varies drastically. In this paper, an improved friction coefficient is adopted.

$$\mu = 23e^{\left[ \frac{-0.028574n}{\ln T - 3.2} \right]^{\left[ (28.3)^{0.55} - 0.87 \right]^{5.16}}} + 0.011\ln(0.04398n + 1)$$

$$+ 0.08(e^{-0.067T} - 1)(e^{-0.022n} - 1) - 0.005\ln(28.3p) + 0.035$$

(17)
For this tracked DCT vehicle, a diesel engine is adopted. The relationship between the engine speed and the output torque is described by a look-up table and is shown in Figure 2. In addition, the PID method is adopted to engage the clutches [8].

3. Heat transfer model

During the launching process, the wet clutch slips and will definitely generate a large amount of frictional heat. Currently, Finite Element Method (FEM) is the most widely used numerical calculation method for engineering analysis of thermal problems. However, there are shortcomings such as overly complicated theory, huge calculation workload and long calculation time. In this paper, from the perspective of heat transfer balance of the lubrication system, a thermal resistance network model is constructed to predict the average temperature rise of the wet clutches based on the lumped parameter method [8].

As shown in Figure 3, the entire lubrication 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).

![Figure 3. Lumped heat transfer model.](image)
The heat conduction equation for each element is shown as follow [8].

\[ \rho c V \frac{dT}{dt} = A h \Delta T \]  

(18)

Considering the heat transfer between elements, the heat transfer of lubricating oil between elements, the heat exchange between components, and the heat generation of components etc., the temperature variations of the entire system can be deduced.

\[
\begin{align*}
C_i \frac{dT_i}{dt} &= \dot{Q}_{io,i} - K_{i,j,o}(T_i - T_{j,o}) - K_{i,e}(T_i - T_e) \\
C_{i,o} \frac{dT_{i,o}}{dt} &= \dot{Q}_{io,o} - K_{i,j,o}(T_{i,o} - T_{j,o}) - \sum_{j} K_{j,o,j}(T_{i,o} - T_{j,o})
\end{align*}
\]  

(19) with 
\[ C_{i,o} = c_{i,o} \rho_{i,o} V_{i,o} \]. \[ K_{i,j,o} \] is the equivalent heat transfer coefficient and the methods to calculate its value can be found in reference [12].

Taking the clutch CL as an example, the temperature can be deduced based on the equation (18),

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

(20)

where \[ \dot{Q}_{CL} = M_{CL} \Delta \alpha_{CL} \].

As mentioned in equation (20), the generated friction work can be deduced by the friction torque and the relative speed. Then, this model and the established dynamic model is integrated.

4. Simulation results discussion
With the developed dynamic model to simulate the generated friction work during the launching process and the established heat transfer model to predict the temperature of the wet clutches, a coupled model is obtained and the thermal condition of the clutches can be calculated. The simulation platform is Matlab/Simulink and the simulation results are shown in Figure 4-7 and the subscript S means launching with single clutch CL, while D means launching with two clutches together. Main parameters for simulation are listed in Table 1.
Figure 6. Speed.

Figure 7. Temperature.

Table 1. Main parameters for simulation.

| Symbol          | Value  | Symbol          | Value  |
|-----------------|--------|-----------------|--------|
| $A$ (m²)        | 3.6    | $I_1$ (kg·m²)  | 0.2523 |
| $A_f$ (m²)      | 0.0427 | $K_{a1}$ (N·m/rad) | 89389  |
| $C_D$           | 0.62   | $K_{a2}$ (N·m/rad) | 110088 |
| $C_d$ (N·m/s/rad) | 2      | $K_{a3}$ (N·m/rad) | 65114  |
| $C_{a1}$ (N·m/s/rad) | 0.5  | $K_{in}$ (N·m/rad) | 283429 |
| $C_{a2}$ (N·m/s/rad) | 0.5 | $R_i$(m)     | 0.086  |
| $C_{a3}$ (N·m/s/rad) | 0.5 | $R_o$(m)     | 0.125  |
| $I_R$ (kg·m²)   | 18.9021| $f$            | 0.123  |
| $I_e$ (kg·m²)   | 3.65   | $i_a$          | 5.3585 |
| $I_{o1}$ (kg·m²) | 0.01637| $i_b$          | 3.818  |
| $I_{o2}$ (kg·m²) | 0.01538| $i_c$          | 1.59   |
| $I_1$ (kg·m²)   | 0.2102 | $i_{H}$        | 0.92   |
| $I_2$ (kg·m²)   | 0.002373| $i_1$        | 0.925  |
| $I_3$ (kg·m²)   | 0.002373| $m$ (kg)      | 7840   |
| $I_4$ (kg·m²)   | 0.2034 | $\beta$(rad)  | 0.174  |
| $I_{in1}$ (kg·m²) | 0.01931| $\delta$      | 1.0554 |
| $I_{in}$ (kg·m²) | 0.001535| $r_c$ (m)    | 0.1935 |

The pressure applied on clutches is shown in Figure 4. Due to the large load, when the speed difference between the active and passive parts of clutch CL decreases to zero, the pressure reaches up to 1.3 MPa when launching with one clutch. While, for the other condition, the launching process is completed when the pressure applied on CL reaches 1.1 MPa, and then it is increased to the system pressure. As for the pressure applied on CH, it drops after 0.55 MPa due to the decrease in the speed difference between the active and passive ends. During different launching processes, the torque changing trend is shown in Figure 5. At the end of the launching process, compared with the shock load generated by the single clutch (2250 N·m), the condition under double clutch launching process is only 1745 N·m.

The speed increase process is shown in Figure 6. As the target gear is the 1st gear, the final speed under two conditions are equal to each other, which is 14.0 Km/h. However, for the condition launching with only one clutch it is 3.6s; as for the dual clutch condition, it takes only 2.6s to complete...
the launching action. Figure 7 shows the thermal conditions of clutches under different launching process. For the launching process with one clutch, the temperature varies from 100°C to 142°C. However, when two clutches are involved in the launching process, the friction work is shared by two clutches. For the clutch CL, its temperature increase is 21°C, and for the clutch CH, it is 14°C. The wear of the clutches is decreased and the service life can be increased significantly.

5. Conclusions
In the transmission system of the tracked vehicle, the wet clutches are the vulnerable part, due to the heavy load. To investigate the thermal condition of these friction components, the dynamic model of the DCT vehicle is established and a heat transfer model is proposed. With these two models, an integrated model is obtained to simulate the launching process and predict the temperature variations of the clutches.

Compared with the condition of launching with one clutch, the shock load is decreased from 2250N·m to 1745N·m by launching with two clutches. In addition, the launching time is also shortened from 3.6s to 2.6s, which means a better dynamic performance. As for the thermal condition, under the condition of launching with one clutch, the temperature has a sharp increase from 100°C to 142°C. While for the condition of launching with two clutches, the temperature rise is 21°C and 14°C respectively, which will definitely decrease the wear of clutches and increase the service life.

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