Research on temperature estimation of smart clutch for vehicle

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Abstract. Clutch is an important component used to transmit or cut off power from engine to transmission. When the clutch is not completely engaged, slipping will happen, yielding generation of friction heat. If slipping frequently happens, the clutch will generate a large amount of friction heat, immediately resulting in increasing of clutch temperatures. Too much high temperature deteriorates the quality and lifespan of the clutch disk. Therefore, a new type clutch, called as smart clutch, is necessary, which can real-time estimate temperature of the clutch in order to prevent overheating and to improve working lifespan. In this paper, a thermal model of the clutch based on lumped parameter is established by analyzing heat conduction and heat convection of all main components of the clutch. Two different test conditions are applied to the clutch in experiment and temperature of the pressure plate is measured. The estimated temperature of the pressure plate in simulation is compared with that of experiment. It is found that the estimated temperature based on the lumped parameter model can fit the experimental results very well, which shows feasibility of the temperature estimation model.

1. Introduction
Clutch is one component in automotive transmission system, consisting of pressure plate, flywheel, disk and other parts. If the clutch works under extreme condition, such as frequent shifting, slipping on the facing surface of the disk causes a large amount of heat, which makes the disk temperature increase highly. Overheating of the disk deteriorates material properties (such as friction coefficient), reduces the clutch lifespan, and even causes accidents. Therefore, it is necessary to estimate temperature of the clutch in real time during practical driving. Based on this objective a so-called smart clutch is developed in our research and the construction of the temperature estimation model is proposed in the paper.

Many researchers have conducted temperature model for the clutch over these years [1]. Newcomb conducted one of the earliest analytical and experimental [2]. Pacey took a simple numerical method with the use of convective heat transfer correlations for the wet clutches [3]. Hebbale, K. developed a thermal model by employing a lumped parameter approach for dry dual clutch transmissions to predict operating temperature of both pressure and center plates during all maneuvers [4]. Fang researched the
heat transfer model for DCT based on finite element method and verified by experiment [5]. Paulraj constructed the thermal model of the clutch facing and the clutch housing to estimate clutch life [6]. Mahmud estimates to the output torque characteristics and temperature rise in simulation for a wet clutch [7]. There existed some temperature error between simulation results and experimental results for DCT in our early research [8]. Now we improve the developed model and apply it to the smart clutch in the paper.

Considering temperature estimation in real time, a lumped parameter model with less calculation is adopted to develop the thermal model in the paper. The remaining parts are arranged as below. Some reasonable assumptions are given in section 2 in order to build the temperature estimation model. In section 3 heat flow paths inside the clutch are analyzed in detail. And then theoretical thermal models, based on lumped parameter, of the pressure plate, the flywheel, the disk, the clutch housing, and air inside the clutch are established. The validation of the proposed model via experiment is implemented in section 4 and discussion about error is given too. The last section is conclusion.

2. Analysis of heat transfer inside clutch

Heat transfer process, including heat conduction, convection, and radiation among all components inside the clutch, is rather complicated. For example, friction heat due to slipping between the disk and the pressure plate conducts to each other, heat of the rotary pressure plate and flywheel convects outwards, and high temperature components radiate heat towards low temperature ones (including air inside the clutch). In order to simplify analysis of model the following assumptions are made.

1. The temperature estimation model is based on lumped parameter considering that heat conduction inside the flywheel, the pressure plate, and the clutch housing [4]. Thus, temperature distributions of all components of the clutch are neglected in the research.
2. All radiation among all components of the clutch is neglected because the amount of them is very small, comparing with that of heat conduction.
3. All slipping energy is used to generate friction heat.
4. During engagement, the amount of friction power during slipping is evenly divided into two parts, which are evenly converted heat imposed on the left and right sides of the disk. But how much friction heat absorbed by the flywheel, the pressure plate, and the disk depends on their material properties.
5. Shapes of the flywheel, the disk, and the pressure plate are considered as cyclic annular.

The overall heat transfer paths are illustrated in figure 1.

![Heat transfer paths inside the clutch](image)

**Figure 1.** Heat transfer paths inside the clutch.
3. Mathematic model of temperature estimation

Based on the above analysis of heat transfer path inside the clutch, the mathematic thermal models of the flywheel, the pressure plate, the disk, the clutch housing, and the air inside the clutch can be constructed as following sections.

3.1. Thermal models of the flywheel and the pressure plate

Figure 2 shows the detail heat transfer paths among the flywheel, the disk, and the pressure plate. Friction heat ($\phi_{sp}$) generated by slipping enters the pressure plate across the friction facing surface. Next, the heat flow diffuses inside the pressure plate by heat conduction. At the same time, part of heat ($\phi_{vp}$) of the pressure plate also convects into air inside the clutch through the outer surface of the pressure plate. Internal energy ($\phi_{ip}$) of the pressure plate goes high with increasing of its temperature. The heat transfer process of the flywheel can be analysis by the same way.

![Diagram](image)

**Figure 2.** Heat transfer paths of the flywheel, the disk, and the pressure plate.

During slipping process, namely not complete engagement of the clutch, power used to generate heat is determined by the following equation.

$$P = M_f \cdot \Delta \omega$$

(1)

According the assumption, half of friction heat ($P$) is absorbed by the flywheel and the disk and other half done by the disk and the pressure. However, how much heat distribution of the disk or the pressure plate depends of material properties of themselves, including specific heat, density, and thermal conductivity, which can be express by the distribution coefficient defined as [9][10].

$$\alpha_1 = \frac{c_p \rho_p \sqrt{R_p}}{c_p \rho_p \sqrt{R_p} + c_d \rho_d \sqrt{R_d}}$$

(2)

Therefore friction power absorbed by the pressure plate and the flywheel are as below, respectively

$$\phi_{sp} = \frac{\alpha_1 p}{2}$$

(3)

$$\phi_{sf} = \frac{\alpha_1 p}{2}$$

(4)

The area of which the pressure plate exposes to air varies with clutch engagement or disengagement. A coefficient ($\beta_1$) defined by equation (5) is employed to indicate variation of the exposed area between the pressure plate and air.

$$\beta_1 = \begin{cases} 1 & \text{engaged} \\ 2 & \text{disengaged} \end{cases}$$

(5)

It is well-known that amount of convection heat flow is proportional to convective coefficient, the convective area between the object and fluid, and temperature difference between them. The convective area between the pressure plate and air inside clutch is both side surface, outer and inner.
surface of circumference, namely \( \pi (\beta_1 (R_{p2}^2 - R_{p1}^2) + 2(R_{p2} + R_{p1})l_p) \). Thus convection heat from the pressure plate to air is calculated as below [11].

\[
\phi_{vp} = \pi a_p (\beta_1 (R_{p2}^2 - R_{p1}^2) + 2(R_{p2} + R_{p1})l_p) (T_p - T_a)
\]  

(6)

Similarly, convection heat from the flywheel to air is obtained by equation (7)

\[
\phi_{vf} = \pi a_f (\beta_1 (R_{f2}^2 - R_{f1}^2) + 2(R_{f2} + R_{f1})l_f) (T_f - T_a)
\]  

(7)

The masses of the pressure plate and the flywheel are \( \rho_p \pi (R_{p2}^2 - R_{p1}^2)l_p \) and \( \rho_f \pi (R_{f2}^2 - R_{f1}^2)l_f \), respectively. According to the relationship between internal energy and temperature, the internal energy change rate [11] of the pressure plate and the flywheel are expressed by equation (8) and (9), respectively.

\[
\phi_{ip} = \rho_p \pi (R_{p2}^2 - R_{p1}^2)l_p C_p \frac{dT_p}{dt}
\]  

(8)

\[
\phi_{if} = \rho_f \pi (R_{f2}^2 - R_{f1}^2)l_f C_f \frac{dT_f}{dt}
\]  

(9)

According to the law of conservation of energy, the heat equilibrium equations of the pressure plate and the flywheel are obtained as below, respectively.

\[
\phi_{sp} = \phi_{vp} + \phi_{ip}
\]  

(10)

\[
\phi_{sf} = \phi_{vf} + \phi_{if}
\]  

(11)

3.2. Thermal models of the disk

Based on the heat flow of the disk shown in figure 2, friction heat \( \phi_{sd} \) due to slipping, convection heat \( \phi_{vd} \) from the disk to air during clutch disengagement, and internal energy change rate \( \phi_{id} \) can be calculated in the same way.

The heat distribution coefficient of the disk is obtained by equation (12) considering that friction heat is 100 percent distributed to the disk and the flywheel or the pressure plate.

\[
\alpha_2 = 1 - \alpha_1
\]  

(12)

As a result, sliding friction power absorbed by the disk is as below

\[
\phi_{sd} = (1 - \alpha_1)P
\]  

(13)

The coefficient \( \beta_2 \) is introduced to show variation of side contact area between the disk and air when the clutch is engaged or disengaged.

\[
\beta_2 = \begin{cases} 
0 & \text{engaged} \\
1 & \text{disengaged}
\end{cases}
\]  

(14)

The both sides of contact area are covered by the flywheel and the pressure plate as engagement and heat of the disk is convected only via its outer and inner circumference surfaces. Therefore, the convection area between the disk and air inside clutch can be described as \( \pi (\beta_2 (R_{d2}^2 - R_{d1}^2) + 2(R_{d2} + R_{d1})l_d) \). Convection heat from the disk to air, accordingly, is obtained.

\[
\phi_{vd} = \pi a_d (\beta_2 (R_{d2}^2 - R_{d1}^2) + 2(R_{d2} + R_{d1})l_d) (T_d - T_a)
\]  

(15)

The mass of the disk is \( \rho_d \pi (R_{d2}^2 - R_{d1}^2)l_d \) and the internal energy change rate of the disk is expressed by equation (16).

\[
\phi_{id} = \rho_d \pi (R_{d2}^2 - R_{d1}^2)l_d C_d \frac{dT_d}{dt}
\]  

(16)

If the law of conservation of energy is applied the heat equilibrium equation of the disk is obtained.
3.3. Thermal models of the clutch housing and air inside the clutch

Convection heat from the pressure plate, the flywheel, and the disk makes air temperature high. Because the pressure plate and the flywheel are in high speed rotation and clutch housing material is generally aluminium alloy with irregular shape and thin thickness, as assumed in section 2 there is no temperature distribution in air inside the clutch and the clutch housing. Internal energy change rate of air inside the clutch is as below.

\[ \phi_{ad} = \phi_{vd} + \phi_{id} \]  

(17)

The outer surface of the clutch housing is estimated to be \( A_h \). Thus, convection heat from the clutch housing to the atmosphere is expressed in equation (19).

\[ \phi_{vh} = \alpha_h A_h (T_h - T_{ao}) \]  

(19)

The average thickness of the clutch housing is \( \delta_h \) and then internal energy change rate of the clutch housing is calculated as below.

\[ \phi_{ih} = \rho_h C_h A_h \delta_h \frac{dT_h}{dt} \]  

(20)

Finally, the law of conservation of energy is applied to the clutch housing and air and then the following energy equilibrium equation can be obtained.

\[ \phi_{vp} + \phi_{vf} + \phi_{vd} = \phi_{ia} + \phi_{vh} + \phi_{ih} \]  

(21)

The temperature estimation model has been established based on the above equations.

4. Validation of temperature estimation model by experiment

Based on the mathematically established thermal model, the corresponding Simulink simulation model is constructed. Parameters of the real commercial clutch are measured and then set in the simulation model. On the other hand, the experiment is designed in order to validate the proposed temperature estimation model. The layout of the clutch test bench is illustrated in figure 3, in which the motor drives the flywheel and the disk of the clutch is braked. The transmitted power (the multiplication of the flywheel rotary speed and transmitted torque) by the clutch and the driving speed can be specified in the experiment. Two types are considered, intermittent power transmission (the normal clutch operation mode) and continuous one (the extreme operation mode). The transmitted power is set as 20KJ in the former, repeating 20 times with interval of five seconds while power is continuously transmitted up to 800KJ in the latter. The rotary speed of the flywheel is 700RPM and the disk is fixed. The transmitted power, which is converted into heat, is calculated by equation (1). A thermocouple is implanted inside the pressure plate. All measured data in the experiment including rotary speeds of the flywheel, the transmitted torque, and temperature of the pressure plate are recorded.

The measured rotary speed of the flywheel and the transmitted torque are used as inputs in the simulation model. After simulation is implemented temperature of the pressure plate is accordingly estimated as output. It can be seen from figure 4(a) that the estimated temperature of the pressure plate is very close to that of experiment in the first test mode. The maximal error between them is less than 5 °C which shows a excellent performance of the temperature estimation model. In figure 4(b) the estimated temperature of the pressure plate is consistent with that measured in experiment. However, there exists difference in the middle period to some extend and the maximal error reaches to 25 °C. It is supposed that the convection in simulation model reduce the temperature. However, the continuous slipping of the clutch rarely happens in real driving but this is an effective test item for model validation and evaluation of the clutch.
5. Conclusions

The temperature estimation model of the clutch based on lumped parameters is established firstly in the paper and then the corresponding simulation model is constructed. Experiment is done for validation of the proposed temperature estimation model. The simulation results show that the developed thermal model can properly estimate temperature although there exists temperature error compared with experimental results in continuous slipping mode. Results of temperature estimation are better than what we had done before.

One of advantages of is that the lumped parameter thermal model is simple, convenient, and less calculation compared with the distributed parameter model, which is significant to temperature estimation in real time. Based on the developed thermal model, the algorithm for temperature estimation can be used to rapidly on-line estimate temperature of the clutch, which is benefit to overheat protection and temperature control of the clutch.

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Nomenclature

\( P \) friction heat
\( \phi \) heat flow
\( \alpha_1 \) distribution coefficient of pressure plate and flywheel
\( \alpha_2 \) distribution coefficient of disk
\( R_1 \) inner radius
\( R_2 \) outer radius
\( \rho \) density
\( \lambda \) thermal conductivity
\( c \) specific heat capacity
\( \alpha \) convective coefficient
\( V_a \) volume of the air inside clutch
\( l \) thickness
\( A_h \) clutch surface area
\( T \) temperature
\( T_{ao} \) atmospheric temperature
\( \delta_h \) average thickness of clutch housing

subscript

\( v \) convection,
\( i \) internal energy
\( p \) pressure plate
\( f \) flywheel
\( s \) slipping friction
\( d \) disk
\( a \) air inside clutch
\( h \) clutch housing