Finite Element Analysis of Thermal Load Characteristics of Dry Dual Clutch

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Abstract. The dry dual clutch of a passenger car is taken as the research object, and its three-dimensional model is established by software CATIA. On this basis, the thermal load characteristics of the dry dual clutch under the condition of multiple starting on the ramp are analyzed by software ABAQUS. The analysis results show that the temperature rise and thermal stress of the parts caused by the friction heat are within the normal range, which indicates that the designs of the pressure plate and the intermediate driving plate meet the requirements. This study obtains the distributions of the transient temperature field of the clutch friction pair under the conditions of the multiple starting, which has an important reference value for the design and optimization of the dry dual clutch.

1. Introduction

In the case of the frequent separation and combination of the dry dual clutch, the active part and the driven part of the friction pair will interact to generate a lot of friction heat, and then cause the temperature of the friction plate, intermediate drive plate and pressure plate to rise. If the heat cannot be dissipated in time, it will cause the temperature of the friction pair parts to be too high, or even burn the friction plate, thus causing a series of safety problems. The non-uniform temperature distribution will produce the uneven thermal deformation of the intermediate drive plate and pressure plate, which will affect the travel of the clutch pressure plate travel and result in the reduction of clutch torque transmission ability and incomplete separation \cite{1}. These problems will affect the service performance and life of the dry dual clutch, therefore it is necessary to analyze its thermal load characteristics in the design process. In this paper, combined with the specific working conditions, the temperature fields of the friction plate, intermediate drive plate and pressure plate of the designed dry dual clutch are analyzed.

2. Working state analysis of dry dual clutch

The material of the friction plate used in the dry dual clutch is the powder metallurgy material. Since the service life of the friction plate is related to the temperature \cite{2}, the wear of the friction plate will increase rapidly when its working surface temperature exceeds a certain range. At present, the critical temperature of the friction plate material is about 400 °C. If the critical temperature is exceeded, the clutch will be permanently damaged due to the burning of the friction plate. In order to ensure the longer service life of the friction plate, the critical working temperature of the clutch is selected as 180
When the car is started continuously on a ramp, the clutch will be engaged several times. Compared with other working conditions, the clutch is more likely to exceed the set critical operating temperature in this case [3]. Therefore, it is of great significance to analyze and study the thermal load characteristics of the dry dual clutch in this specific working state. The specific working conditions are as follows: finishing 8 times of starting on the ramp with a slope of 30 ° in 70s, and the ambient temperature and the initial test temperature of the clutch are both 25 °C.

3. Modelling and meshing of clutch

At the very beginning, the three-dimensional (3D) solid model of the dry dual clutch is established by software CATIA (shown in Fig. 1). Due to the focus on the temperature field distribution of the friction plate, pressure plate and intermediate drive plate, these parts need to be separated. In the case of not affecting the accuracy of the finite element analysis, the structure should be appropriately simplified or ignored, such as the small fillet on the part and the groove of the friction plate. When studying the thermal load characteristics of the dry dual clutch, the analysis type of the software ABAQUS is set as General: Heat transfer. After simplification, the 3D model is imported into ABAQUS in *STP format, and the finite element mesh is done. The selected element type is DC3D8, and the number of grids is 30401, as shown in Fig. 2.

4. Determination of material properties and boundary conditions

4.1. Determination of Material properties

In the study the copper based powder metallurgy material, whose material property is anisotropic, is used for the friction plate. It is very difficult to accurately measure the physical properties of the anisotropic material. Therefore, we can regard it as the isotropic material, and the physical property parameters of the material do not change with temperature and time. The cast iron material is used for the pressure plate and intermediate driving plate, which is also regarded as isotropic. The material properties of powder metallurgy and cast iron are shown in Table 1.

| Name                      | Elastic modulus (GPa) | Poisson's ratio |
|---------------------------|-----------------------|-----------------|
| Cast iron                 | 120                   | 0.3             |
| Copper based powder metallurgy | 110                  | 0.275           |

4.2. Determination of boundary conditions

According to the actual situation, the boundary conditions of the finite element model are simplified: ignoring the wear of materials; assuming that the heat generated by the clutch friction is completely absorbed by the friction pair; assuming that the friction coefficient is constant, and the value is the average dynamic friction factor.
4.2.1. Establishment of contact pairs
The contact method of surface to surface (standard) is selected, and the following contact pairs are established: pressure plate-friction plate and intermediate drive plate-friction plate.

4.2.2. Determination of convective heat transfer coefficient
The convective heat transfer coefficient of dry clutch is related to its structure and cooling mode. If the error between the predicted temperature and the actual measured value is less than 30%, the value of convective heat transfer coefficient is considered to be reasonable \[4\]. The pressure plate and intermediate driving plate are rotating disks, and the convective heat transfer with air is forced convection heat transfer. The convective heat transfer coefficient can be calculated by the following empirical formula.

\[
h = \begin{cases} 
0.04(\lambda_a / d)R_e^{0.8}, & R_e > 2.4 \times 10^4 \\
0.7(\lambda_a / d)R_e^{0.55}, & R_e \leq 2.4 \times 10^4 
\end{cases}
\]

\[R_e = \frac{\omega r^2}{\nu}\]

\[(1)\]

where \(\lambda_a\) is the air thermal conductivity, and \(\lambda_a = 0.772 \, \text{w} \cdot \text{m}^{-1} \cdot \text{°C}^{-1}\); \(d\) is the diameter of rotating disc (mm); \(R_e\) is the Reynolds number; \(\omega\) is the angular velocity of rotating disk (\(\text{rad/s}\)); \(r\) is the radius of rotating disk (mm); \(\nu\) is the air kinematic viscosity, and \(\nu = 14.8 \times 10^{-6} \, \text{m}^2/\text{s}\).

The convective heat transfer coefficient of clutch pressure plate and intermediate drive plate can be obtained by (1).

4.2.3. Determination and distribution of heat flux
Heat flux density refers to the heat flux per unit area per unit time. The heat flux density generated between the friction pairs of the dry dual clutch can be calculated by the following mathematical expression\[5\]:

\[q = \mu p r = \mu p r \Delta \omega\]

\[(3)\]

where \(q\) is the heat flux, \(\mu\) is the dynamic friction coefficient, \(p\) is the surface pressure, and \(\Delta \omega\) is the speed difference.

Through the dynamic analysis of the starting process of the car, the relationship between the relative rotational speed of the active part and the driven part of the clutch, \(\Delta n\) can be obtained as follows:

\[
\Delta n = \begin{cases} 
1000, & t \leq 0.5 \\
-80r + 1400, & 0.5 < t \leq 1.75 \\
0, & t > 1.75 
\end{cases}
\]

\[(4)\]

Finally, the heat flux loading expression is obtained:

\[
q(r,t) = \begin{cases} 
9043200 \times r, & t \leq 0.5 \\
9043200 - 7324560 \times r \times t, & 0.5 < t \leq 1.75 \\
0, & t > 1.75 
\end{cases}
\]

\[(5)\]

In this paper, it has been assumed that all the heat generated by sliding friction is absorbed by the friction pair, and the absorbed heat is distributed on the contact surface of the intermediate drive plate, pressure plate and friction plate, and the distribution coefficient, \(K_q\), is as follows\[6\]:

\[K_q = \frac{q_1(r,t)}{q_2(r,t)} = \left(\frac{\lambda_a \rho c_v}{\lambda_a \rho c_v} \right)^{0.5}
\]

\[(6)\]

where \(q_1\) is the heat flux input to the middle drive plate and pressure plate, \(q_2\) is the heat flux of friction plate, and their calculation formula are:
It is calculated that the heat distribution ratio of the former and the latter is 61.45% and 38.55%, respectively, in the middle drive disc friction plate contact pair and the pressure plate friction plate contact pair. Figs. 3 and 4 show the convective heat transfer coefficient and heat flux loading diagram, respectively.

\[
q_1 = \frac{K_q}{K_q + 1}, \quad q_2 = \frac{1}{K_q + 1} \tag{7}
\]

5. Analysis of thermal load simulation results during continuous starting

Fig. 5 shows the temperature field distribution nephogram of the clutch after 8 starts in 70s on a 30% slope. It can be seen from Fig. 5 that the maximum temperature of the clutch is about 87.01 \( ^\circ \text{C} \).

In order to further analyze the thermal load characteristics of clutch in the continuous starting process, half of the clutch is analyzed. Fig. 6 shows the temperature field nephogram of clutch at the time of maximum temperature at each combination. The following conclusions can be drawn from Fig. 6: (1) With the increase of starting times, the maximum temperature of the clutch gradually increases; (2) In this process, the maximum temperature is 112.1 \( ^\circ \text{C} \) and occurs at the outermost end of the middle drive plate, which appears in the eighth starting process of the car at \( t = 63.9 \text{s} \); (3) The maximum temperature of friction plate is 104.3 \( ^\circ \text{C} \), which appears at the moment \( t = 64.1\text{s} \).
At the fourth combination

At the fifth combination

At the sixth combination

At the seventh combination

At the eighth combination

Therefore, it can be seen that the maximum temperature of the clutch designed in this paper is lower than the required critical temperature of 240 °C ~ 260 °C, which shows that the thermal load characteristics of the friction pair can meet the design requirements.

6. Conclusion
In this paper, the 3D geometric model of dry dual clutch is established, and based on this model, the thermal load characteristics of the dry dual clutch are analyzed by the finite element method. The results show that the temperature rise and thermal stress caused by friction heat are within the normal range, which indicates that the design of pressure plate and intermediate driving plate meets the requirements. Through the research, it provides some theoretical guidance and reference for the application of dry dual clutch in practical engineering.

Acknowledgement
This work was supported by Practical Innovation Training Program for College Students in Jiangsu (SPITP) (No.201710298063Z).

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