Thermal Analysis of Double Mass Flywheel Secondary Flywheel

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Abstract—To solve the problem of thermal overload failure in dual-mass flywheel (DMF), we performed temperature field calculation and thermal analysis of the secondary flywheel in DMF. Based on the results of this analysis, we performed thermal stress analysis on the secondary flywheel. The heat transform model was built for the secondary flywheel using the thermodynamics theory. Finite element model (FEM) was built for the secondary flywheel using ABAQUS 6.12-3 software. This model was solved to obtain the distributions of the temperature field and thermal stress in the secondary flywheel after it completed five engagement-disengagement cycles. Finally, thermal shock test was conducted for the secondary flywheel on the thermal shock test bench. We performed a targeted test to verify the accuracy of thermal analysis of secondary flywheel. The analysis method can be used to optimize product parameters such as the product thickness.

1. Introduction
Dynamic transmission system is an important component located between the engine and drive wheels of an automobile [1]. Dual mass flywheel (DMF) is one of the best torsional vibration dampers in the transmission system, and it is widely used in automobiles [2]. Secondary flywheel is a key component of DMF, which engages with the clutch friction disc and transmits power to the speed transmission system via friction [3]. The transmission system may get shocked or overloaded during the process of power transfer, especially in bad working conditions. In such situations, tremendous heat is generated due to the relative sliding friction between the clutch pressure plate and secondary flywheel of DMF. This heat cannot be dissipated in time because of the closed shell, leading to a higher thermal stress. The pressure plate and the secondary flywheel ablate and break under the effect of this high thermal stress. When the secondary flywheel breaks or ablates, the damping system of DMF fails.

In this paper, the secondary flywheel in DMF was used as the research object. Referring to the thermo-mechanical coupling model of the clutch, the temperature field model and the thermal conduction and convection formulae were built according to the engaging characteristics of the dual mass flywheel and the clutch. Based on the heat conduction model of the dual mass flywheel and the heat-structure coupling model of the clutch pressure plate, we developed the thermal-stress finite element model of the secondary flywheel to study the distributions of the temperature field and...
thermal stress in the secondary flywheel of DMF during five engagement-disengagement cycles. Finally the accuracy of thermal analysis of secondary flywheel was verified by a test.

2. Temperature field calculation of secondary flywheel
The heat transfer process is a very complex phenomenon, it is usually divided into the following basic heat transfer methods: heat conduction, heat convection and thermal radiation. Heat is transmitted from the friction pair to the secondary flywheel by thermal conduction. The heat of secondary flywheel is dissipated into air via thermal convection and thermal radiation. In this paper, heat radiation and the heat transferred by the secondary flywheel to other components are not considered. Therefore, we developed a separate model for the heat transfer of the secondary flywheel. We can use formula (1) to calculate heat conduction:

\[ Q = \lambda A' \frac{\Delta t}{\delta} . \]  

(1)

Here, \( A' \): cross-sectional area perpendicular to the direction of heat transfer; \( \delta \): Thickness of the conductor; \( \Delta t \): Temperature difference between the two sides of the conductor; \( \lambda \): Coefficient of thermal conductivity.

2.1. Determination of heat flux density of the secondary flywheel
The friction pair generated heat, which was then transmitted in the form of heat flux density into the secondary flywheel.

Figure 1 is a schematic flowchart for calculating the heat flux density on the working surface of secondary flywheel.

The secondary flywheel tightly engages with the friction disc during clutch slipping. The work of slipping friction is calculated using formula (2):

\[ L = T_{\text{emax}} \left( \frac{J_a \omega_0}{3} - \frac{T_{\text{remax}}}{(36)^{4/3}} \right) \left( \frac{7}{3} \frac{h+1}{6} - \frac{15}{8} T_{\nu} - \frac{9}{2} T_{\nu} \right). \]  

(2)

Where, \( T_{\text{emax}} \) is the maximum output torque (N \cdot m); \( J_a \) is the equivalent moment of inertia converted from the gross mass of vehicle (kg \cdot m^2); \( h \) is the coefficient constant, which are 2.2 for gasoline engine; \( \omega_0 \) is the angular velocity of engine at the beginning of clutch slipping (r/min); \( T_{\nu} \) is the moment of road resistance (N \cdot m).

The temperatures of friction disc and secondary flywheel will increase after absorption of heat. The larger the sliding friction radius, the higher the sliding friction speed. The greater the sliding friction torque, the more heat is generated. As shown in Figure 2, the friction surface can be simplified into a concentric circle with an inner diameter \( r_1 \) and outer diameter \( r_2 \).
Fig. 2 Concentric circle representing the friction surface.

The heat flux density of the friction surface with radius $r$ is given by the following equation:

$$q = \frac{dN_r}{dt} = \frac{3N_{fr}}{2\pi (r^2 - r_i^2)} \left(1 - \frac{r}{t_0}\right). \tag{3}$$

The thermal conductivity of secondary flywheel and friction disc varies according to the material used to manufacture them \cite{7}. Slipping friction generates heat flux density, which is then partitioned between the secondary flywheel and the friction disc; the partition coefficient is calculated by the formula \cite{8}:

$$k = \frac{q_{sf}}{q_{fd}} = \frac{\lambda_{sf} \cdot C_{sf} \cdot \rho_{sf}}{\lambda_{fd} \cdot C_{fd} \cdot \rho_{fd}}. \tag{4}$$

Here, $\lambda_{sf}$ is the coefficient of thermal conductivity ($W/(m \cdot K)$); $C_{sf}$ is specific heat capacity of the secondary flywheel ($J/(kg \cdot K)$); $\rho_{sf}$ is density of secondary flywheel ($kg/m^3$); $\lambda_{fd}$ is the coefficient of thermal conductivity of friction disc ($W/(m \cdot K)$); $C_{fd}$ is the specific heat capacity of friction disc ($J/(kg \cdot K)$); $\rho_{fd}$ is density of friction disc ($kg/m^3$).

The sum of heat flux density transmitted to the secondary flywheel and friction disc is the total heat flux density generated from slipping friction. The heat flux density of secondary flywheel is given by formula (5):

$$q_{sf} = q \cdot \frac{k}{1+k}. \tag{5}$$

The simultaneous equation is built by combining formulae (3) and (5). At a given time $t$, the heat flux density of the secondary flywheel with radius $r$ is as follows:

$$q_{sf} = \frac{3N_{fr}}{2\pi (r^2 - r_i^2)} \left(1 - \frac{r}{t_0}\right) \cdot \frac{k}{1+k}. \tag{6}$$

2.2. Determination of heat transfer coefficient of convection

Surfaces other than the friction surface of the secondary flywheel will dissipate heat when they come into contact with air during engagement. When the secondary flywheel separates from the friction disc, all surfaces will dissipate heat as they come into contact with air. Convection is the main mode of heat transfer in this system. Therefore, the heat transfer coefficient of convection is estimated by the following formula:

$$h = 0.023 \frac{K_a}{d} R_e^{0.8} P_r^{0.4}. \tag{7}$$

Here, $K_a$ is the coefficient of thermal conductivity of air ($W/(m \cdot K)$); $R_e$ is Reynolds number; $d$ is the diameter of secondary flywheel (m); $P_r$ is Prandtl number.

There is a very small difference between the convection coefficients at consecutive temperatures. In order to reduce the amount of data input during simulation, this paper takes $25^\circ C$ as the starting
temperature and takes 80°C, 130°C, and 180°C as the intermediate interpolation temperature. According to the relevant parameters of “The Table of Physical Properties of Dry Air”, the convective heat transfer coefficient of each temperature node is calculated and used as the input data of the FEM simulation calculation formulae.

Table 1. Heat transfer coefficients of convection at different temperatures.

| Temperature (°C) | Heat transfer coefficients (W/(m²°C)) |
|-----------------|-------------------------------------|
| 25              | 72.07                               |
| 80              | 65.72                               |
| 130             | 60.63                               |
| 180             | 573.3                               |

3. Finite modeling of secondary flywheel
In this experiment, we performed thermal analysis of secondary flywheel after it completed five engagement-disengagement cycles; the analysis was conducted by referring to the engineering experiences encountered while performing relevant experiments on clutches. The flow chart of thermal analysis of secondary flywheel is shown in Figure 3.

3.1. Establishment of 3D model and grid generation
The 3D physical model of the secondary flywheel was built using Pro/E software and meshed using ABAQUS software. Figure 4 illustrates the grid model of secondary flywheel.

3.2. Temperature field boundary condition
Before starting, we placed the DMF in a room and ensured that the temperature of DMF was equivalent to the room temperature. The temperature-field boundary condition of the secondary flywheel was set to the room temperature (25°C). When the clutch is engaged and slipped, the surface temperature of secondary flywheel surface rises continually. On the one hand, the secondary flywheel absorbs heat flux and transfer to the other parts, on the other hand, the secondary flywheel also dissipates heat into the surrounding air by convection.
3.3. Analysis step configuration of FEM model
We determined the distributions of the temperature field and thermal stress in the secondary flywheel after it had undergone five continuous cycles of engagement and disengagement. We defined 10 analysis steps corresponding with the five disengagement/engagement cycles. The analysis steps were used for carrying out transient heat transfer analysis.

The engagement and disengagement between secondary flywheel and the friction pair were considered to be equivalent to the engagement and disengagement between secondary flywheel and clutch friction disc. The engagement time was 2.25s, whereas the disengagement time was 1s.

3.4. Heat density flux boundary condition
During engagement, slipping friction occurs between the friction disc and the working surface of secondary flywheel. The slipping friction generates heat, which is then absorbed by the friction disc and secondary flywheel in the form of heat flux density.

The heat flux density boundary condition was constrained for the working surface of secondary flywheel in the load module of ABAQUS software; the expression of heat flux density has been presented in formula (6).

3.5. Displacement constraints condition
On the condition of the displacement constraints of the auxiliary flywheel, this paper is determined through the installation of dual mass flywheel. In general, secondary flywheel is installed on the first flywheel through bearing, connected with damping flange through rivets. The clutch assembly is fixed on the secondary flywheel. Only the peripheral orifice of the auxiliary flywheel is a fixed connection. Therefore, the displacement constraint is restricted and the displacement freedom is restricted.

3.6. Convective heat-loss boundary condition
Except for the working surface of the secondary flywheel, surfaces dissipate heat when they come into contact with air during engagement. When the secondary flywheel separates from the friction disc, all the surfaces dissipate heat as they come into contact with air.

Convective heat-loss boundary condition was configured using the parameters presented in Table 1. The input ambient temperature was the room temperature of 25°C. Five interactions were configured in ABAQUS software and convective heat transfer was configured for each step.

3.7. Configuration of material properties
With reference to the standard ISO1083-2004, we presented the material parameters of QT450 in Table 2. The material parameters were input in the property module of ABAQUS software to establish the material mode.

| Material parameter                  | Value            |
|------------------------------------|------------------|
| Density                            | $7.06 \times 10^3$ kg/m$^3$ |
| Coefficient of thermal conductivity| $4.7 \times 10^{-3}$ W/(m·K) |
| Young’s modulus                    | $1.69 \times 10^{11}$ N/m$^2$ |
| Specific heat capacity             | $5.1 \times 10^3$ J/(kg·K) |
| Coefficient of thermal expansion   | $1.01 \times 10^{-5}$ K |
| Poisson’s ratio                    | 0.257            |

4. Distributions of temperature field and thermal stress in secondary flywheel

4.1. Temperature field distribution of secondary flywheel
To obtain the temperature field distribution of secondary flywheel, we solved the FEM model using ABAQUS software. The above finite element model is presented to solve the temperature field distribution of the vice flywheel, and the results are presented in the form of temperature field distribution nephogram. Figure 5 is the secondary flywheel temperature fields after the first to the fifth engagement-disengagement cycles, respectively.

Fig. 5 Temperature field distribution of the secondary flywheel in the five engagement/disengagement.

The temperature field of secondary flywheel had a radial distribution pattern. The temperature of the secondary flywheel increased significantly when it was engaged with the friction disc. Moreover, the temperature of the secondary flywheel did not drop apparently when the secondary flywheel separated from the friction disc. After the fifth engagement, we observed that the temperature of the secondary flywheel was the highest at 158.7°C. Figure 6 shows the variation trend of temperature from the first to the fifth cycles of engagement and disengagement. For the secondary flywheel, the average temperature rise was 32.4°C.

Fig. 6 Secondary flywheel temperature curve after consecutive 5- engagement and disengagement.

4.2. Thermal stress distribution of secondary flywheel
Due to the action of temperature field and external displacement constraints, thermal expansion occurs in the secondary flywheel. This thermal expansion induces loads and causes thermal stress in secondary flywheel. This belongs to a heat-mechanics coupling issue [9].

Thermal analysis model was built for the secondary flywheel using ABAQUS software. The results of temperature field analysis were the load inputs of the model. Sequentially heat-mechanics analytical method was used for performing thermal stress analysis of secondary flywheel [10]. Figure 7 shows the results of thermal stress analysis of secondary flywheel.
Fig. 7 Thermal stress distribution of secondary flywheel.

The results indicate that the thermal stress distribution of secondary flywheel agreed well with the temperature field distribution of secondary flywheel. There was apparent stress concentration at the inner ring of the secondary flywheel. The maximum thermal stress was 414 MPa, so it was less than the tensile strength (450 MPa) of the material. Therefore, the secondary flywheel did not show fracture failure.

5. Thermal shock test for secondary flywheel
Thermal shock test was performed on secondary flywheel to verify the results of thermal stress analysis. The equipment, procedures, and results of thermal shock test are as follows:

5.1. Thermal shock test bench
The bench primarily consisted of the following components: power system, hydraulic servo system, and loading system. Figure 8 illustrates a schematic diagram of the thermal shock test bench.

Fig. 8 Schematic diagram of the thermal shock test bench.

The bench was used to perform thermal shock test on DMF. The working conditions of the test were as follows:

1. Ambient temperature: Room temperature (25°C).
2. Rotational speed: 3000 ± 100 rpm; torque: 380 Nm.
3. Ventilation was maintained during the test.
4. Loading energy was determined using the above-mentioned formulae. Its calculated value was 117 KJ in this experiment.

5.2. Thermal shock test
According to the requirements of the thermal shock test for DMF, we performed the thermal shock test for DMF.

1. First, turn on the variable frequency motor and accelerate to 3000 RPM. Then the hydraulic servo system was shut down, and the clutch was engaged with the secondary flywheel for 2.25 s.
2. Hydraulic servo system was turned on to separate the clutch from the secondary flywheel for 1 s. The brake was applied to keep the specimen motionless.
(3) The above steps were repeated five times.

5.3. Results and analysis
 Thermal shock test was performed on DMF using the above-mentioned procedure. As shown in Figure 9, the DMF was removed from the bench and detected.

![Fig. 9 The secondary flywheel after thermal shock test](image)

According to the internal standard requirements of the relevant enterprise, secondary flywheel should neither break nor develop cracks after completion of the thermal shock test. Figure 9 clearly illustrates that the surface of the secondary flywheel is smooth without any cracks or breaks. The results agree well with the simulation analysis results of section 5.2, which is presented earlier in this paper.

According to the test standard provided by the manufacturer, an intuitive method is adopted to judge the material strength limit. When the thermal stress is greater than the material limit, the specimen will crack and damage. When the thermal stress is less than the material limit, the specimen will not crack. In the second chapter, the maximum thermal stress of the secondary flywheel under the above temperature field is 414Mpa less than its tensile strength of 450 Mpa. After the test, the test piece was disassembled, and the second flywheel found that the surface integrity of the secondary flywheel was not produced, and its thermal stress was less than the material strength limit. The feasibility of the reliability analysis of the flywheel is verified by the above comparison.

6. Conclusion
 We used ABAQUS software to analyze the distributions of temperature field and thermal stress in the secondary flywheel of DMF. We developed the FEM model of secondary flywheel using the thermodynamic theory. We analyzed the distributions of temperature field and thermal stress after five cycles of engagement and disengagements between the secondary flywheel and friction disc were solved. Thermal shock test was performed to verify the thermal analysis. We arrived at the following conclusions:

1) The temperature of the secondary flywheel increased gradually from the first to the fifth cycle of engagement. The temperature of the secondary flywheel apparently decreased following each disengagement. After the fifth engagement, the temperature of the secondary flywheel was the highest at 157.8 °C. For the five cycles of engagement and disengagement, the average temperature rise was 32.4 °C.

2) Thermal stress distribution of secondary flywheel basically corresponded with the temperature field distribution of secondary flywheel. Thermal stress concentration was observed at the inner ring of the secondary flywheel during engagement. The maximum thermal stress was 414Mpa, so it was less than the material’s tensile strength (450Mpa). Therefore, the secondary flywheel did not show any signs of fracture failure, which was verified by thermal analysis. Our findings can serve as theoretical and experimental basis for the design of DMF.
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