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Numerical and experimental analysis of transient temperature field of ventilated disc brake under the condition of hard braking

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ABSTRACT

The ventilated disc brake of vehicles was selected as the objective of this study, which was built on 3D modeling technology. Through establishing thermo-structural coupling model, this study analyzed the transient temperature field of automobile brake under the condition of hard braking. Meanwhile, the test of hard braking was carried out on professional platform of vehicle test bench in this paper, and temperature curves of brake disc in radial and circumferential directions were obtained. By comparison, the experimental values and simulation values were basically equal. The rationality of the selected finite element analysis (FEA) was attributed, which provided a better theoretical basis for further experimental analysis.

1. Introduction

Hard braking is a working condition in which the speed of a vehicle is reduced to zero within split seconds under the condition of a accident. In this process, the kinetic energy converts into mechanical energy which is accompanied by the dissipation of amount frictional heat [1,2]. The frictional heat generated at the contact interface disc-pads is almost absorbed by the brake, which is approximately 90% [3]. Belhocine and Bouchetara [4] pointed that it lead to a high temperature gradient of the brake disc and friction plate. Meanwhile, the thermal stresses are also changed correspondingly with the temperature. Thevenet et al. [5] pointed that the temperature improvement rapid of the brake causes a negative effect on the brake inevitable during the braking process. Such as heat fade, oxidation, and even heat cracks caused by thermal stress, which is dangerous in the case of hard braking.

To better study the temperature of brake disc in hard braking phase, not only analytical methods but also experimental were employed in this paper [6]. Compared with previous researches, this study, by means of the professional platform of the vehicle test bench, arranged a series of thermocouples as dynamic temperature measuring points near the surface of the disc, and the automatic data acquisition instrument was used to collect the dynamic data of the whole braking process. In addition, in order to verify the accuracy of numerical calculation, this paper compared the experimental results with the numerical results.

2. Theory analysis

2.1. The FEA of transient heat

Transient heat conduction in 3D heat-transfer problem is governed by the following differential equation:

$$-\left(\frac{\partial q_x}{\partial x}+\frac{\partial q_y}{\partial y}+\frac{\partial q_z}{\partial z}\right)+Q=\rho c\frac{\partial T}{\partial t} \tag{1}$$

Where q_x, q_y, q_z are the conduction heat fluxes in directions of X, Y and Z, respectively; Q is the internal heat generation rate per unit volume.

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T: Temperature,K	α: Thermal expansion coefficient, (10^{-6} K^{-1})
t: Time, s	a: Deceleration of the vehicle, m s^{-2}
ρ: Mass density, kgm^{-3}	q: Frictional heat flux density, J m^2 s^{-1}
m: Mass of the vehicle, kg	u: Initial speed of the vehicle, m s^{-1}
c: Specific heat, J kg^{-1} K^{-1}	k_{cf}: Correction factor of rotational mass
ν: Poisson ratio	k: Thermal conductivity, W m^{-1} K^{-1}
E: Young modulus,GPa	A: Area of a single friction disc, m^2
η: Conversion rate, $\eta = 0.9$	$\triangle T$: Temperature difference, K
n: Number of friction pads	Q: Unit volume heat flux, J m^{-3}
v: Speed, km/h	ω: Angular velocity, rad/s

2.2. Friction heat theory

Under the condition of high rotation speed, heat generated by the friction pair can be considered as a heat source in the actual contact point. The heat flux density of friction disc brake can usually be used to calculate energy reduction method.

Assuming that the car begins to brake at the speed of v, it will not stop until the speed is lowered to zero. The actual braking process affected by many environmental factors, kinetic energy of the vehicle cannot be converted into friction heat absolutely, and this article assumes that 90% of the total kinetic energy is absorbed by brake disc and friction pads [3,7]. Energy conversion efficiency must be considered. If the conversion rate is η, the energy equation is as following:

$$\mathrm{Q}(\mathrm{t}) = \eta \cdot \frac{1}{2} m k_{cf} u^2 \tag{2}$$

Supposing that the vehicle is braking at a constant deceleration and the friction heat generated in the friction region is evenly distributed. According to the definition of the heat flux, the calculation formula of heat flux and braking time can be expressed as following:

$$q(t) = \frac{dQ(t)}{n \cdot A \cdot dt} = \frac{d\left(\eta \frac{1}{2} m k_{cf}\left(u^2 - (u - at)^2\right)\right)}{n \cdot A \cdot dt} = \frac{\eta k_{cf} m a (u - at)}{n \cdot A} \tag{3}$$

2.3. Transient thermo-structural coupling theory

2.3.1. FEA theory of transient thermal

The definition of transient heat conduction is that the temperature of the object is not persistent over time, until it reaches a new equilibrium. This unsteady process is generally caused by a sudden step change of the boundary condition, a transient variation of the inner heat source and so on [8]. In this study, the contact friction heating of the brake is equal to a transient variation of moving heat source. So we can solve the temperature field through the method of transient thermal analysis.

The formula of transient thermal:

$$\rho c\left(\frac{\partial T}{\partial t} + \{v\}^T\{L\}T\right) = \{L\}^T[D]\{L\}T + \dddot{q} \tag{4}$$

$\{L\}^T = \left[\frac{\partial}{\partial x}, \frac{\partial}{\partial y}, \frac{\partial}{\partial z}\right]$ is vector operator; $\{v\}^T = [v_x, v_y, v_z]$ is heat transfer velocity vector; $[D] = \begin{bmatrix} K_{xx} & 0 & 0 \\ 0 & K_{yy} & 0 \\ 0 & 0 & K_{zz} \end{bmatrix}$; K_{xx}, K_{yy}, K_{zz} represent heat conduction coefficient in direction x, y, z; $\{q\}$ is heat flux vector; $\dddot{q}$ is the heat generation rate per unit volume;

2.3.2. FE method of thermo-structural coupling analysis

During braking phase, they are mutual influence with each other between the temperature field and the stresses field of disc. The variation in temperature can cause the structural deformation of the object, including thermal expansion and contraction, also the deformation of the structure has an influence on the thermal boundary conditions when solve the temperature field.

This study selected direct coupling method to solve thermo-structural coupling problem, which is the iterative calculation of the temperature field and the stresses field.

Energy equation of coupled thermo-structural analysis:

$$\int_V \left\{\rho\left(Q - \frac{\partial U}{\partial t}\right) + \sigma_{ij}\frac{\partial V_i}{x_j}\right\}\mathrm{d}V = \int_S H\mathrm{d}S \tag{5}$$

H is boundary heat flux density; V is volume of continuous medium; U is internal energy; σ_{ij} is Cauchy stress component.

According to the principle of virtual work, the formula of structural displacement is established:

$$\int_V \rho_{ij}\frac{\partial \delta u_i}{\partial x_i}\mathrm{d}V = \int_V \rho b_i \delta u_i \mathrm{d}V - \int_V \rho\frac{\partial V_i}{\partial t}\delta u_i \mathrm{d}V \tag{6}$$

The finite element equation of the structural transient temperature field and thermal stress and strain field analysis is as following:

$$\dot{u}^T\left(K_u \cdot \dot{u}(t) + M_T \cdot \dot{T}(t) - F(t)\right) = 0 \tag{7}$$

$$T^T\left(C_u \cdot \dot{T}(t) + M_u \cdot \dot{u}(t) - D - Q - K_T \cdot T(t)\right) = 0 \tag{8}$$

C_u is heat capacity matrix; K_T is heat conduction matrix; M_u is thermos-mechanical coupling matrix; D is dissipative vector; Q is thermal load vector.

3. FE model

3.1. Basic assumptions

In actual hard braking, the transient temperature field and thermal stress distribution of the disc brake are influenced by a lot of environmental factors. In order to achieve the simulation results to close to the actual results as far as possible, as well as more convenient for numerical calculation, it is necessary to make the following basic assumptions:

(1) The wheel is in a state of pure roll, the slip rate is zero;
(2) The friction between the brake disc and the friction pads are in accordance with the Coulomb friction law;
(3) The pressure of friction surface is constant, and the distribution is uniform [9];
(4) Material properties are isotropic and independent of the temperature [10];
(5) The abrasion of the disc and the plate are neglected, and the friction heat flux is completely allocated to the disc and the pads [11];
(6) Ignore thermal radiation.

As shown in Fig. 1, the 3D product is designed with the Siemens NX software, and the FE model is based on computer code ANSYS Workbench platform in Fig. 2 [12].