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FRICTION THERMODYNAMICS RESEARCH OF AUTOMOTIVE VENTILATED DISC BRAKE

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Abstract

In order to realize the friction thermodynamic calculation of automotive ventilated disc brake that meets the conditions of heat production and heat transfer, the analytic method, experimental method and numerical simulation method are comprehensively used in the paper. The elastic-plastic state transition condition is analyzed by using micro convex theory, and the entropy balance equation is deduced based on an extensive amount of phenomenological equation. The flow control equation and the definite solution in two-dimensional space are concluded according to the boundary layer theory, and the multiplication method is come up for the three-dimensional heat transfer calculation. A new simulation method based on fluid-solid-thermal coupling calculation of ventilated disc brake is come up, and the synchronous iterative calculation based on MPCCI platform is finished by using ABAQUS for thermal-solid coupling and FLUENT for fluid-solid coupling. As a result, the transient temperature field, stress field and convective heat transfer coefficient are acquired with accurate boundary conditions. During the Link3900 NVH friction experiment result, it shows that the maximum temperature deviation between numerical simulation and experimental value is 5.6% in the braking cycle, which has obvious consistency, so the numerical simulation method is accurate and feasible.

Keywords: thermodynamic, numerical simulation, thermal-solid coupling, fluid-solid coupling, brake disc

INTRODUCTION

Ventilated disc, also known as air-cooling type brake disc, has stronger thermal decay resistance than the solid disc due to the fin structure (shown in Fig.1), and is widely used in automotive braking. At present, the friction thermodynamic studies of brake disc are mostly based on the simplified numerical simulation method, which has big difference with the actual boundary conditions, such as structure-thermal coupling (ignoring air flow factors) or fluid-structure coupling (ignoring friction heating conditions). Among the typical research achievements, Zhao [1] and Dong [2] obtain the temperature field and stress field of the disc brake under different working conditions with the structure-thermal coupling method that assume the convective heat transfer coefficient is constant. Cho [3] and Robert [4] choose the indirect coupling method to improve the computational convergence during the transient thermo elastic behavior research of disk brake, of which temperature field is set as boundary condition for stress field calculation, but the error is bigger than the direct coupling algorithm; Meng[5] and Chen[6] adopt the fluid-structure interaction method to calculate the heat transfer coefficient and the temperature field distribution with CFD (Computational Fluid Dynamics) analysis and assumes the disk body has stable heat input.

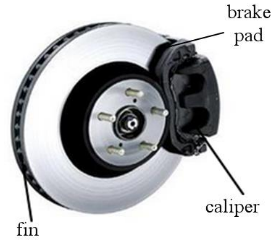

Figure.1 Structure diagram of ventilated dis

As for the thermodynamic calculation of ventilated disc brake, only the boundary conditions that both satisfied the heat transfer and heat generation can obtain accurate iterative calculation results, otherwise, the result error is large. According to the research background, multiple theories are used for braking process analysis and a new advanced method of fluid-structure-thermal direct coupling simulation for ventilated disc brake is proposed in the paper, which can effectively improve the thermodynamic calculation accuracy. In order to verify its feasibility, the friction brake test is finished by Link3900 NVH.

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THEORY RESEARCH IN BRAKING PROCESS

Contact analysis

The contact surface between the brake pad and ventilated disc is discrete. Therefore, the actual contact area consists of all the contact points and determines the size of the friction heat source. According to up pal and Probert's research, it can be known that the contact points will remain or even decrease along with the increase of the actual contact area if the brake pressure increases to the critical load. Thus it can be seen that the braking torque will not always increase when the brake pressure is increasing.

In the braking process, the contact of friction pair has three statuses, including elastic, plastic and middle (elastic-plastic) forms, and the increase rate of the actual contact area in the elastic state is smaller than that of the plastic state. According to the contact point of the spherical micro convex body theory [7], when P_0=2.76τ_y (where P_0 is actual contact area center pressure, τ_y is ultimate tangential stress), plastic deformation will occur at the contact surface. The condition of the elastic-plastic deformation on the surface of the brake disc can be expressed by Tabor's depth of press calculation method as follows:

$$\frac{a}{r} = k_n \frac{\sigma_T (1-\mu^2)}{E} \tag{1}$$

Where a is the contact spot radius, r is spherical micro convex body radius, σ_T denotes the average shear stress, k_n is a scaling factor (determined by ratio of the average shear stress and the effective yield stress). The influence of surface profile on the elastic-plastic state can be expressed by Williamson and Greenwood index method as shown in Eq. (2).

$$\phi=\frac{E'}{H}\left(\frac{\sigma}{r}\right)^{0.5}=\frac{1}{H}\left(\frac{1-\mu_1^2}{E_1}+\frac{1-\mu_2^2}{E_2}\right)\left(\frac{\sigma}{r}\right)^{0.5} \tag{2}$$

Where σ is the standard deviation of the profile, E' is the average elastic modulus of brake pad and ventilated disc, H is hardness of the ventilated disc, $\mu_1$and E_1、$\mu_2$and E_2 are respectively Poisson's ratio and elastic modulus of ventilated disc and brake pad. In unit load, when $\phi<0.6$, elastic state will remain, and when $\phi>1$, the plastic state will occur.

Analysis of friction heat

The thermal stress of ventilated disc caused by friction is one kind of non- equilibrium thermodynamics problem. The phenomenological equation [8] can be expressed in space V as follows:

$$\frac{\partial Z}{\partial \tau}=\int_V \frac{\partial(\rho z)}{\partial \tau}dV=\int_V(-\nabla\cdot J_z+\sigma_Z)dV \tag{3}$$

Where ρz is density of extensive quantity Z, J_z is density of for extended flow, σ_Z is generation rate of extensive quantity, τ is time. Assume that there are i kinds of forces in the friction system, according to Newton's second law and the symmetry of the stress tensor, the momentum equation of the ventilated disc can be deduced as follows:

$$\rho\frac{d\boldsymbol{U}}{d\tau}=-\nabla\cdot\boldsymbol{P}+\rho\sum_i \boldsymbol{F}_i \tag{4}$$

Where $\boldsymbol{U}$ is the velocity vector of center of mass of ventilated disc, ρ is density of the disc, $\boldsymbol{P}$ is stress tensor, $\boldsymbol{F}_i$ is the volume force of each function component. Ignoring chemical reaction factor between the brake disc and brake pad, the entropy balance equation [9] of ventilated disc can be deduced by the momentum equation, mass and energy conservation laws as Eq. (5).

$$\frac{ds}{d\tau}=-\nabla\cdot\left[sU+\frac{J_q}{T}-\frac{\mu_k}{T}J_k\right]+J_q\cdot\nabla\left(\frac{1}{T}\right)+J_k\cdot\left[-\nabla\left(\frac{\mu_k}{T}\right)+\frac{M\sum_i F_i}{T}\right]-\frac{1}{T}\Pi:\nabla\boldsymbol{U} \tag{5}$$

Where s is the entropy density, J_q is density of heat flow rate, J_k and μ_k are respectively diffusion flow and chemical potential of the material, T is temperature, M is the molar mass, Π is interfacial stress tensor. The entropy balance equation can fully express the fluid-solid-thermal coupling state of the ventilated disc. During the Eq.(5), $\Pi:\nabla\boldsymbol{U}$ can be expressed in the $Oxyz$ coordinate system as Eq. (6).

$$\Pi : \nabla \boldsymbol{U} = \begin{bmatrix} P_{11} & P_{12} & P_{13} \\ P_{21} & P_{22} & P_{23} \\ P_{31} & P_{32} & P_{33} \end{bmatrix} \begin{bmatrix} \frac{\partial u_1}{\partial y_1} & \frac{\partial u_2}{\partial y_1} & \frac{\partial u_3}{\partial y_1} \\ \frac{\partial u_1}{\partial y_2} & \frac{\partial u_2}{\partial y_2} & \frac{\partial u_3}{\partial y_2} \\ \frac{\partial u_1}{\partial y_1} & \frac{\partial u_2}{\partial y_2} & \frac{\partial u_3}{\partial y_3} \end{bmatrix} \tag{6}$$

Heat transfer analysis of fin structure

According to the conservation of energy law and Fourier's law, the heat flux vector of any point in the space can be decomposed in *Oxyz* coordinates. In infinitesimal parallelepiped, 3-D unsteady heat conduction differential equation can be expressed as Eq. (7).

$$\rho c \frac{\partial t}{\partial \tau} = \frac{\partial}{\partial x}(\lambda \frac{\partial t}{\partial x}) + \frac{\partial}{\partial y}(\lambda \frac{\partial t}{\partial y}) + \frac{\partial}{\partial z}(\lambda \frac{\partial t}{\partial z}) + \dot{\Phi} \tag{7}$$

Where ρ, c, τ, $\dot{\Phi}$ are respectively density, specific heat, time, and heat that generated in unit time and volume. The ventilated disc has a great number of fins, which can increase the heat radiation area of the convection and radiation, improve heat transfer capability, and ensure the strength.

The fin structure is generally uniform, so the thermal conductivity performance in height direction can be expressed by typical one dimensional steady state heat conduction theory. The 1/2 model of fin heat conduction is shown in Fig.2. During the model, the internal heat source in unit volume can be calculated as follows:

$$\dot{\Phi} = -\frac{\Phi_s}{A_c \mathrm{d}x} = \frac{hP(t - t_\infty)}{A_c} \tag{8}$$

Where $A_c\mathrm{d}x$ is elemental volume, P and A_c are respectively the perimeter and area of heat transfer cross section, h is convective heat transfer coefficient, t is fin temperature, t_∞ is air temperature. According to the heat conduction differential Eq. (8) can be deduced as follows:

$$\frac{\mathrm{d}^2 t}{\mathrm{d}x^2} + \frac{\dot{\Phi}}{\lambda} = \frac{\mathrm{d}^2 t}{\mathrm{d}x^2} - \frac{hP}{\lambda A_c}(t - t_\infty) = 0 \tag{9}$$

Assumed the height of ventilated disc fin is H, excess temperature is θ ($\theta = t - t_\infty$). The boundary conditions for Eq. (9) can be expressed as follows:

$$\begin{cases} x = 0, & \theta = \theta_0 = t_0 - t_\infty \\ x = H, & \frac{\mathrm{d}\theta}{\mathrm{d}x} = 0 \end{cases} \tag{10}$$

Thus, the general solution for Eq. (10) is

$$\theta = c_1 e^{mx} + c_2 e^{-mx} \tag{11}$$

In the formula,

$$\begin{cases} m = \sqrt{\dfrac{hP}{\lambda A_c}} = \text{const} \\ c_1 = \theta_0 \dfrac{e^{-mH}}{e^{mH} + e^{-mH}} \\ c_2 = \theta_0 \dfrac{e^{mH}}{e^{mH} + e^{-mH}} \end{cases} \tag{12}$$

Heat transfer effect of the fin structure can be judged by ventilation efficiency η_f, which can be calculated as Eq.(13).

$$\eta_f = \frac{\dfrac{hP}{m}\theta_0 \text{th}(mH)}{hPH\theta_0} = \frac{\text{th}(mH)}{mH} \tag{13}$$

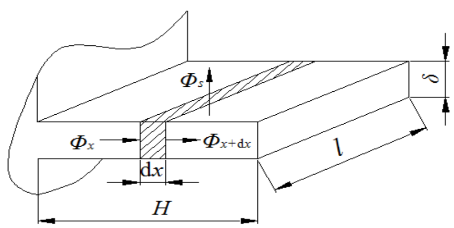

Figure. 2 1/2 model of fin heat conduction

Two dimensional coupling control equation

During the heat exchange calculation between the ventilated disc and air, the convection heat transfer coefficient (also called surface heat transfer coefficient) h is dynamic [10]. Generally, the key physical quantities that affect h include air physical properties, flow rate, flow state, coupling surface shape, etc. In the paper, assume air velocity at the point (x,y) is (u,v), temperature is t. Ignoring internal energy that produced by viscous dissipation, according to the conservation of energy law, the energy equation of convective heat transfer in two-dimensional space can be deduced as follows:

$$\rho c_p \left(\frac{\partial t}{\partial \tau} + u\frac{\partial t}{\partial x} + v\frac{\partial t}{\partial y} \right) = \lambda \left(\frac{\partial^2 t}{\partial x^2} + \frac{\partial^2 t}{\partial y^2} \right) \tag{14}$$

The mass conservation equation can be expressed as:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{15}$$

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The momentum conservation equation can be expressed as:

$$\rho\left(\frac{\partial u}{\partial \tau}+u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}\right)=F_x-\frac{\partial p}{\partial x}+\eta\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}\right) \tag{16.1}$$

$$\rho\left(\frac{\partial v}{\partial \tau}+u\frac{\partial v}{\partial x}+v\frac{\partial u}{\partial y}\right)=F_y-\frac{\partial p}{\partial y}+\eta\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}\right) \tag{16.2}$$

Where ρ, c_p, τ, λ, t are respectively density, specific heat, time, heat conductivity coefficient and temperature of ventilated disc, p stands for air pressure, F_x and F_y are respectively the volume force in x direction and y direction.

It can be seen from Eq.(14) ~ Eq.(16) that the equations are closed and there are only four unknown boundary parameters. If these unknown values are given, the coupling equation can be solved in principle. But due to the complexity and nonlinear characteristics of the momentum equation, the mathematical computation cannot be completely obtained at present [11]. In order to solve the problem, boundary layer conditions are introduced, so that equation (16) can be simplified. It is known from the Prandtl theory that the viscosity of the fluid is confined in the thin layer (boundary layer) which is close to the coupling surface and the viscous effect can be ignored because the air velocity gradient is quite small. According to the hydrodynamic theory, ignoring the velocity v and two order derivative in the main flow direction, the momentum equation can be simplified as follows:

$$u\frac{\partial u}{\partial x}+v\frac{\partial u}{\partial y}=-\frac{1}{\rho}\frac{\mathrm{d}p}{\mathrm{d}x}+v\frac{\partial^2 u}{\partial y^2} \tag{17}$$

The multiplication method of multi-dimensional heat conduction calculation

The heat transfer problem of ventilated disc in a multi-dimensional space can be solved by multiplication method because its structure is regular. According to the fin structure, the analytical solution of one dimensional (x direction) heat transfer can be obtained by separating the variable method as follows:

$$\frac{\theta(\eta,\tau)}{\theta_0}=\sum_{n=1}^{\infty}C_n\exp(-\mu_n^2F_0)\cos(\mu_n\eta) \tag{18.1}$$

Where μ_n is root of the transcendental equation, C_n is characteristic value, $F_0=\frac{a\tau}{\delta^2},\eta=\frac{x}{\delta}$. When τ=0, the initial condition of C_n can be obtained by Fourier series theory as follows:

$$C_n=\frac{2\sin\mu_n}{\mu_n+\cos\mu_n+\sin\mu_n} \tag{18.2}$$

Similarly, assume r is disc diameter, t_0 is initial disc temperature, t_∞ is air temperature. The analytical solution of temperature field can be obtained as follows:

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$$\frac{\theta(\eta,\tau)}{\theta_0}=\sum_{n=1}^{\infty}C_n\exp(-\mu_n^2F_0)J_0(\mu_n\eta) \tag{19.1}$$

$$C_n=\frac{2}{\mu_n}\frac{J_1\mu_n}{J_o^2(\mu_n)+J_1^2(\mu_n)} \tag{19.2}$$

Where J_0 and J_1 is Bessel function.

Through the multiplication method, temperature field solution of the fin can be calculated as follows:

$$\Theta=\frac{\theta(x,y,\tau)}{\theta_0}=\Theta_{p1}(x,\tau)\cdot\Theta_{p2}(y,\tau) \tag{20}$$

The cylindrical surface temperature field can be calculated as follows:

$$\Theta=\frac{\theta(x,r,\tau)}{\theta_0}=\Theta_p(x,\tau)\cdot\Theta_c(r,\tau) \tag{21}$$

In Eq.(20) and Eq.(21), Θ_p and Θ_c respectively stand for one-dimensional temperature field analytical solutions of transverse and cylinder surface under the third boundary conditions.

FLUID-SOLID-THERMAL COUPLING NUMERICAL SIMULATION

Coupling implementation scheme

In order to realize the fluid-solid-thermal coupling calculation of the ventilated disc brake, a numerical simulation method based on multi physics field coupling standard MPCCI, nonlinear structural analysis software ABAQUS and fluid analysis software FLUENT. The principle of coupling calculation is that ABAQUS and FLUENT will calculate solid-thermal model and fluid-solid model respectively and simultaneously according to the FEA (Finite Element Analysis) method [12].

During the fluid-solid-thermal coupling calculation process, ABAQUS performs the function of coupling data receiving and provides parameters such as node displacement and wall temperature. FLUENT performs the function of coupling data exchanging, and provides parameters such as the convection temperature and the convective heat transfer coefficient. Meanwhile, MPCCI will make sure the coupling data has synchronization, which means that the whole interpolation calculation step is the same, and the boundary conditions of the coupling region are continuously updated.

During the interpolation calculation, MPCCI will use the bucket pre-contact search algorithm [13] to determine the grid matching attribute between the fluid model and the solid model. According to the physical quantity, the coupling data can be divided into non-conservation data (such as displacement, velocity, temperature, etc.) and conservation data (such as quality, flow, etc.).

Model establishment and pre process

The FEA models of solid and fluid (air flow direction is parallel to the disc transverse plane) should be established in the same coordinate system. In order to ensure the computational efficiency and convergence, the element partition and local optimization method are used in hexahedral mesh division of both model as shown in Fig.3. The mesh type of C3D8HT in ABAQUS model is chosen, of which cells number is 17370 and nodes number is 25821. The sliding mesh technology is applied in the FLUENT model, of which cells number is 174587, and nodes number is 146158.

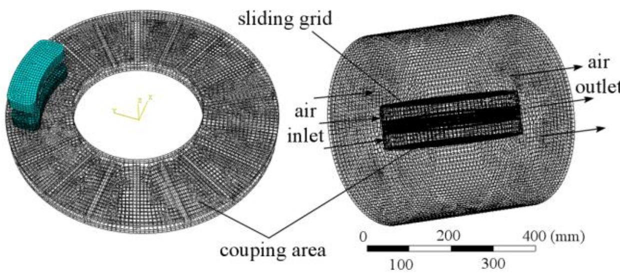

Figure.3 Mesh division diagram

Before the calculation of fluid-solid-thermal coupling, the constraint, load and boundary conditions of the solving model are defined as follows:

(1) In the pre process of ABAQUS, the material properties of the ventilated disc (marked as *D*) and the brake pad (marked as *P*) should be defined according to the physical properties as shown in Tab.1 (where ρ, E, λ, c, l_n are respectively density, elasticity modulus, heat conductivity coefficient, specific heat and thermal expansion coefficient). The analysis step is established with temperature-displacement coupling method, and the time step is set as 0.001s, while the load step number is set as 1000. The movement of the ventilated disc is restricted by the method of surface coupling reference point. The contact friction coefficient and braking pressure are defined by the experimental data that obtained from Link3900 NVH device. The contact between the friction pairs is set as “hard contact”, and the influence of the thermal radiation in the braking process is defined. The initial temperature of the model is set as 100℃, which is consistent with experimental condition.

Table.1 Material property

symbol	ρ	E	λ	c	l_n
unit	$Kg\cdot m^{-3}$	Pa	W/(m·K)	kJ/(kg·K)	
P	7100	1.4e11	48	452	1.0e-5
D	2000	4.1e8	1.4	1200	0.18e-5

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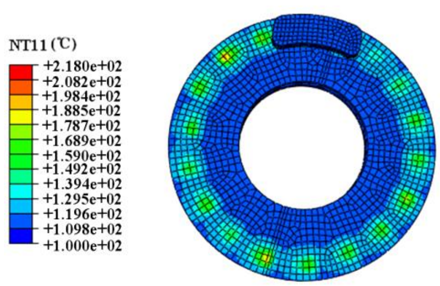
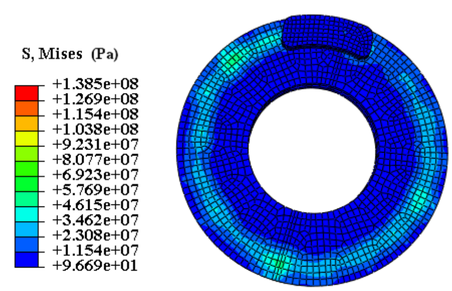
(2) In the pre process of FLUENT, the dynamic fluid region is realized by sliding grid method, which can ensure the continuity and consistency of coupling nodes movement. The angular velocity of the moving area is consistent with the speed of the ventilated disc. The k-epsilon model is used to define the parameters of air flow rate and initial temperature, which are also consistent with the experimental conditions. The iterative step size is set as 0.001s.

(3) In the pre process of MPCCI, the coupling area and calculation time step will be defined. In order to ensure the data is shared properly, the fluid model is set as “data exchange”, and the solid model is set as “data receive”.

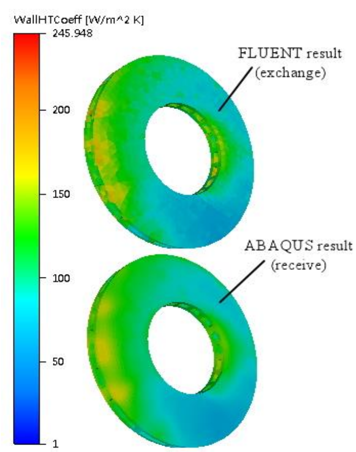
Numerical simulation results

From the calculation process, it can be known that the direct coupling iteration calculation work quantity is quite large, which need lots of computer memory. As a result, the workstation Wise team SP260 is used for improving the calculation efficiency. Through the results, the temperature field, stress field and the heat transfer coefficient of ventilated disc at the moment of 1.0s are shown in Fig.4.





(a) Transient temperature field (b) Transient stress field



(c) Surface heat transfer coefficient

Figure. 4 Numerical simulation results

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According to the numerical simulation results of fluid-solid-thermal coupling, the conclusions can be concluded as follows:

(1) From Fig.4 (a) and Fig.4 (b), it can be seen that the “hot spot” phenomenon in the temperature field is quite more obvious than the stress field, especially in initial braking stage. The “hot spot” phenomenon is mainly caused by the ventilation disc fin structure, which will result in unbalancing brake pressure on the contact surface. The brake pressure on the overlapping position of the brake pad and fin is larger than both sides, so the friction heat is higher, and also is the temperature, thermal deformation and stress. Because the elastic deformation is recovered very soon, the “hot spot” phenomenon in stress field does not last long.

(2) From Fig.4(c), it can be seen that the greater air speed and pressure will come out greater heat transfer coefficient. The convective heat transfer coefficient in the coupling surface between FLUENT model and ABAQUS model shows the same distribution although the mesh is different. As a result, the transient temperature and stress calculation should be more accurate.

EXPERIMENTAL VERIFICATION AND ANALYSIS

Experimental scheme

At present, there are mainly three methods for disc brake performance testing in tribology research, such as small sample experimental method, bench test and road test method. According to the research target in the paper, the bench test method based on Link3900 NVH device is applied for verifying the fluid-solid-thermal coupling calculation results. The Link3900 NVH device is one of the most advanced tribology experimental machine currently, of which detected parameters includes temperature, pressure, friction coefficient, rotate speed, humidity, torque, moment of inertia, noise, etc. During the experimental scheme, the ventilated disc and related sensor are installed as shown in Fig.5.

In order to improve the reliability of the experimental results, the PC control program is designed for each repeated experiment cycle in the paper. The principle of experiment cycle process is as follows: When the last experiment cycle n is finished, the brake pressure will be canceled, and the pressure will not start until the disc temperature is falling to environment temperature, then the new experiment cycle $n+1$ will begin. Meanwhile, the time step sampling frequency is set as 0.01s, and the experimental cycle number is set as 100.

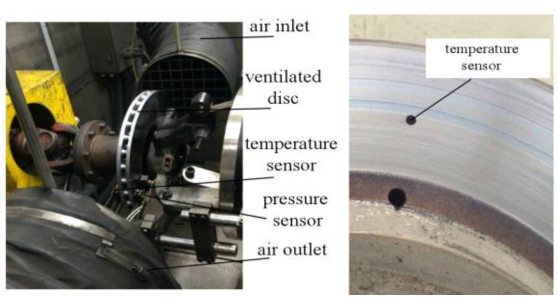

Figure.5 The ventilated disc and related sensor installation diagram

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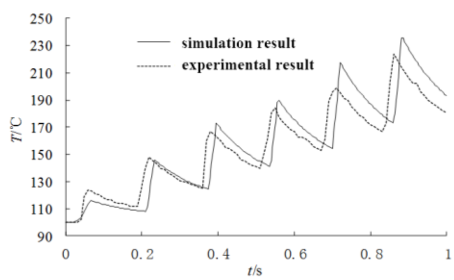
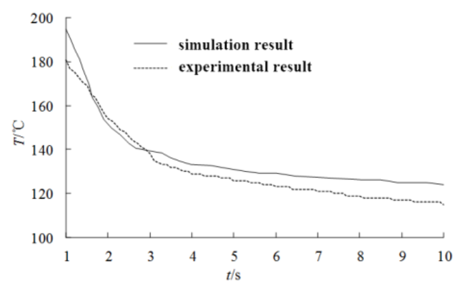
Results analysis

The temperature contrast between experiment data and simulation data in the same position can effectively verify the accuracy of simulation results. The contrast curve of the experimental and numerical values in a single cycle that shown in Fig.6. It can be concluded as follows:

(1) From the experimental and simulation result, it can be known that the experimental value is basically consistent with the simulation result. During the brake process, the temperature is increasing intermittently with the disc rotation. When the test point is separated from brake pad, the friction work will stop and the temperature will reduce slightly due to the effect of thermal conductivity, heat convection and heat radiation.

(2) The maximum deviation between the numerical simulation and the experimental value is 5.6%。Considering with the experimental error and the structural deviation between the numerical analysis model and the physical prototype, it can be concluded that the calculation method of fluid-thermal-structural coupling based on MPCCI is accurate and feasible.





(a) Comparison of 0~1s temperature changes (b) Comparison of 1~10s temperature changes

Figure.6 Contrast of experiment data and simulation data

CONCLUSIONS

The theoretical analysis, numerical simulation and experimental verification are comprehensively used for the fluid-thermal-structural coupling study during the braking process in the paper, and the conclusions can be obtained as follows:

(1) In the aspect of theory research, the elastic plastic transition condition of the ventilated disc is analyzed according to the theory of friction micro convex body. The heat convection, heat conduction and the relationship between the temperature and the stress are expressed by the entropy balance equation. The coupling control equation in two-dimensional space is derived based on the heat transfer theory, and the solution of heat transfer model is extended to three-dimensional space by multiplication solution method under the third boundary conditions.

(2) A numerical simulation based on multi physics field coupling standard MPCCI, nonlinear structural analysis software ABAQUS and fluid analysis software FLUENT is presented. In order to verify the accuracy and feasibility of the numerical simulation, the friction experiment

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based on NVH Link 3900 test rig for ventilated disc is carried out. With the same boundary conditions and loads between the numerical simulation method and the experimental method, the results show that the test value from the temperature sensor is quite consistent with the numerical value from the corresponding location, which means the numerical simulation scheme has a good accuracy. Therefore, the multi physical field coupling analysis of ventilated disc in this paper has important significance for the improvement of thermodynamic calculation validity and structure optimization.

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