Journal of Mechanical Science and Technology 24 (2010) 81~84
www.springerlink.com/content/1738-494x
DOI 10.1007/s12206-009-1116-7

Investigation of temperature and thermal stress in ventilated disc brake based on 3D thermo-mechanical coupling model †

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(Manuscript Received May 2, 2009; Revised September 30, 2009; Accepted October 12, 2009)

Abstract

Ventilated disc brakes are widely used for reducing velocity due to their braking stability, controllability and ability to prove a wide-ranging brake torque. During braking, the kinetic energy and potential energies of a moving vehicle are converted into thermal energy through friction heating between the brake disc and the pads. The object of the present study is to investigate the temperature and thermal stress in the ventilated disc-pad brake during single brake. The brake disc is decelerated at the initial speed with constant acceleration, until the disc comes to a stop. The ventilated pad-disc brake assembly is built by a 3D model with a thermo-mechanical coupling boundary condition and multi-body model technique. To verify the simulation results, an experimental investigation is carried out.

Keywords: Ventilated disc brake; Temperature field; Thermal stress; Thermo-mechanical couple; Multi-body model

1. Introduction

As a part of the automobile safety system, the brake system plays an important role in protecting the driver and passengers. Ventilated disc-pad brakes (Fig. 1) are widely used for reducing velocity due to their braking stability, controllability, and ability to provide a wide-ranging brake torque. The brake disc with vanes rotates through the caliper. A ventilated disc with straight vanes is most popular, easy and straightforward to make. The pressure on the pistons pushes the pads with three-dimensional geometry against the brake disc and produces brake torque.

Most of the mechanical energy of a moving vehicle is converted into heat through the friction between the brake disc and pads in the braking process, and 99% heat energy is dissipated through the brake disc and pad. The braking processes in the friction units of a brake are very complicated. In the course of braking, all parameters of the processes (velocity, load, temperature, and the conditions of contact) vary with time.

Many studies about the brake disc thermo-mechanical coupling analysis have been done. Choi and Lee developed an axisymmetric finite element model for the thermoelastic contact problem of brake disk and investigated the thermoelastic instability phenomenon of disc brake during the drag-braking process and repeated braking process [1, 2]. Gao and Lin et al. analyzed the transient temperature field and thermal fatigue fracture of the solid brake disc by a three-dimensional thermal-mechanical coupling model [3, 4], In 2007, the authors investigated the temperature field and thermal distortion of the ventilated brake disc by axisymmetric model and partial 3D model [5]. In 2008, the authors identified the temperature field of ventilated brake disc in the repeated braking based on the thermo-mechanical coupling and multi-body model [6].

The object of the present study is to investigate the temperature and thermal stress in the ventilated disc-pad brake during single brake based on multi-body technique and 3D thermo-mechanical coupling model.

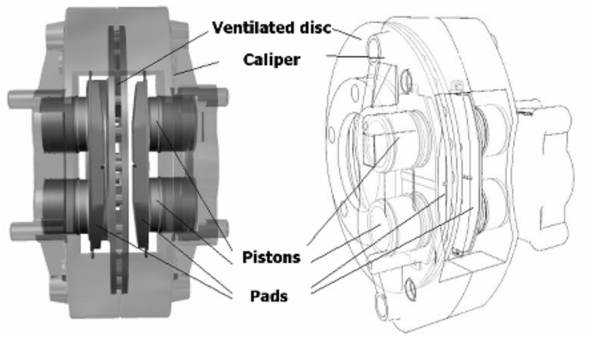

Fig. 1. Ventilated disc-pad brake.

† This paper was presented at the ICMDT 2009, Jeju, Korea, June 2009. This paper was recommended for publication in revised form by Guest Editors Sung-Lim Ko, Keiichi Watanuki.

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2. Formulation

2.1 Heat flux

During braking, the kinetic and potential energies of a moving vehicle are converted into thermal energy through friction heating between the brake disc and the pads. Frictional heat is generated on the surface of the brake disc and brake pads. In the present work, considering the amount of heat generation by wear is very small relative to the heat generated by friction, so the effect of material wear is neglected. The friction heat flux generated in the interface of disc and pad can be expressed as

$$q(x,y,t) = \mu P(x,y)v(t) = \mu P(x,y)\omega(t)\cdot r \tag{1}$$

where μ is friction coefficient, P is contact pressure, v is sliding velocity, which is defined by angular velocity of the disc ω and radius of brake disc r.

The total heat generated on the frictional contact interface q equals the heat flux into the disc q_D and the heat flux into the pad q_P. The relative braking energy γ absorbed by brake disc is

$$\gamma = \frac{q_D}{q} = \frac{q_D}{q_D + q_P} = \frac{1}{1+\left(\frac{\rho_P c_P k_P}{\rho_D c_D k_D}\right)^{\frac{1}{2}}} \tag{2}$$

In the above equation, ρ is density, c is specific heat, k is thermal conductivity, and subscripts D and P identify the disc and pad, respectively. This value is in terms of the material properties of the brake disc and pad.

Besides the relative brake energy, the thermal contact behavior between the disc and pad was considered: thermal contact conductance. According to Lee's research [7], the thermal contact conductance is 30000 W/mK in the present paper.

2.2 Convection heat transfer coefficient

In braking, the major portion of the generated heat flows out to the air, and some goes out into the air by radiation, conducted into the hub and pad, and the rest of the heat is stored in the disc rotor. The convective heat transfer coefficients of the ventilated disc brake are quoted from the experiential formulas by Limpert [8].

For the solid part of the ventilated disc, the convection heat transfer coefficient associated with laminar flow can be approximated by

$$h_R = 0.70(k_a/D)\mathrm{Re}^{0.55} \tag{3}$$

where, D is the outer diameter of disc, Re is the Reynolds number, and k_a is the thermal conductivity of air.

The cooling effectiveness associated with the internal vanes tends to decrease somewhat for higher speeds due to the increased stagnation pressure of the air. For straight vane of disc in the laminar flow condition $\mathrm{Re}<10^4$, the heat transfer coefficient inside the vanes of the brake disc is

$$h_R = 1.861(\mathrm{Re\,Pr})^{1/3}(d_h/l)^{0.33}\times(k_a/d_h) \tag{4}$$

where l is length of cooling vane, Pr is the Prandtl number, and Re equals $(\rho_a d_h/\mu_a)\mathrm{V}_{\mathrm{average}}$. The hydraulic diameter d_h is defined as the ratio of four times the cross-sectional flow area (wetted area), divided by the wetted perimeter as illustrated in Fig. 2.

Table 1. Material property and dimension of brake disc and pad.

	Disc	Pad
Inner radius (mm)	81.5	85
Outer radius (mm)	128	125
Thickness (mm)	16	10
Density (kg/m^3)	7031	2595
Specific heat (Nm/kgK)	495	1465
Thermal conductivity (Nm/s℃m)	56.72	1.212
Young's Modulus (GPa)	125	1.5
Poisson's Ratio	0.29	0.25
Thermal Expansion Coefficient (㎛/mK)	10	60

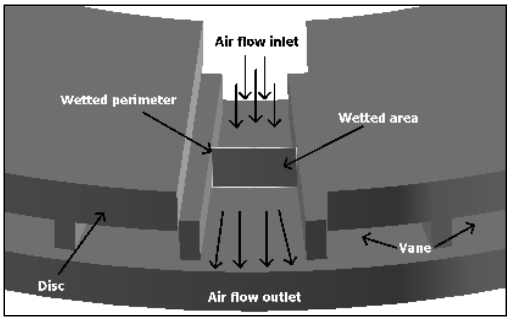

Fig. 2. Air flow in the vane of ventilated brake disc.

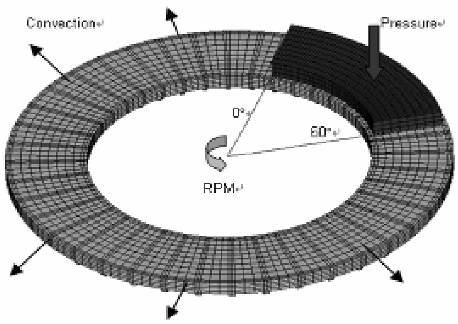

Fig. 3. 3D thermo-mechanical coupling model of ventilated brake disc.

2.3 Thermal strain and thermal stress

Thermal strain is an elastic strain that results from expansion with increasing temperature, or contraction with decreasing temperature. The thermal strain at a given temperature T can be assumed to be proportional to the temperature change ΔT over a limited range of temperatures and is described as

$$\varepsilon = \alpha(T - T_0) = \alpha(\Delta T) \tag{5}$$

where T_0 is a reference temperature and α is the thermal expansion coefficient.

The elastic deformation is satisfied by Hooke's law; hence the thermal stress σ is described as

$$\sigma = E\alpha(\Delta T) = E\varepsilon \tag{6}$$

3. Simulation and experiment

In the present simulation, the vehicle weight is 1900 kg, friction coefficient of contact pair is 0.38 and initial temperature is 40℃. The disc material is gray cast iron GC250. The dimensions and material properties of the brake disc and pad are listed in Table 1. The vehicle is decelerated at an initial velocity of 100 kph (87.6 rad/s), to the ending velocity 0 kph with 0.6g in 4.72 seconds; 95% mechanical energy is converted into the thermal energy.

The model is symmetric about the central crossing plane of ventilated disc. The cover angle of the pad is 60°, from θ=0° (the exit of friction region) to θ=60° (the entrance of friction region). On the work surface of the brake disc and pad, the frictional contact pair is defined to simulate the friction heat process. The brake pressure 2.38MPa is applied on the pad to generate the brake force. Based on multi-body technique, a revolute joint is controlled by the angular velocity time function at the center of the brake disc. The convection heat transfer coefficient is applied to the surfaces of brake disc.

In the present research, the coupling process is as follows: the first step, calculate the contact pressure distribution in contact pair, then calculate the heat flux, obtain the temperature field in the disc and pad, and in the next step calculate thermal stress and thermal distortion. The contact pressure is changed due to the thermal distortion, and a new coupling loop is carried out again until the velocity is equal to 0.

To verify the simulation, an experimental investigation is carried out to determine the temperature distribution in the brake disc. The disc temperature is burnished to the initial temperature, and accelerated to 100 kph, then the brake is applied with constant deceleration 0.6g, until the disc comes to a stop. The brake is operated three times.

4. Result

4.1 Contact pressure

The Fig. 4 plots the non-axisymmetric contact pressure distribution in the friction interface of pad. From the distribution at t=1.18s, it is found that the maximum contact pressure is occurred at the entrance and descends towards the exit of friction region. The contact pressure distribution at the mid stage of braking shows that the maximum contact pressure is at the center of contact region due to the thermo-mechanical coupling behavior.

4.2 Temperature field

Friction heating, thermal distortion, and elastic contact, affects the contact pressure and temperature on the contact surface. This thermo-mechanical coupling behavior due to a relative high sliding speed that exceeds the critical sliding speed can be unstable, leading to localized high-temperature contact regions called "hot spots" on the sliding interface [9]. The Fig. 5 shows the temperature field with the distortional hot spot in the ventilated brake disc in the different stages of braking operation. At the t=2.36 seconds, the temperature distribution

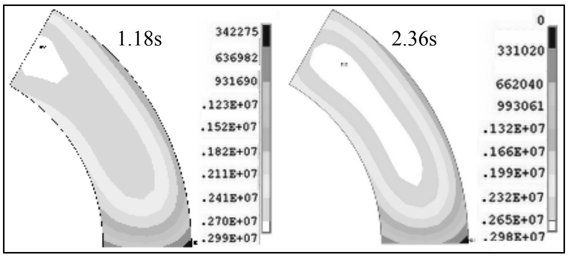

Fig. 4. Contact pressure in the in the friction interface of pad.

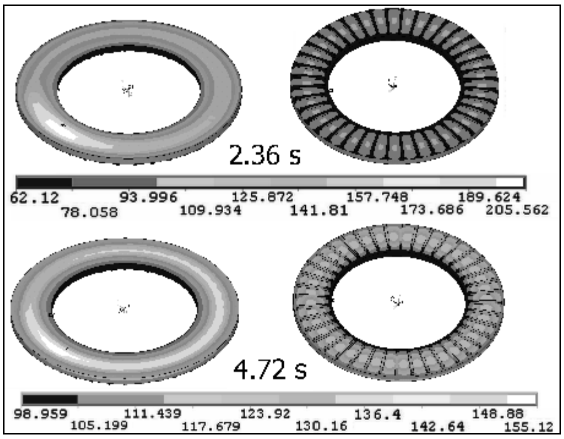

Fig. 5. Temperature field in the ventilated brake disc.

is not uniform field, since the frictional heat generated in the contact region and the rest region dissipates the heat by the convection and heat conduction of the disc. The maximum temperature (205.56℃) is occurred occurs around the middle circle of the contact surface at the exit of contact region, since the middle circle region is where the maximum contact pressure region is generated. The maximum temperature is 155.12℃ at t=4.72 second, decreased 50℃. Compared with the contour at t=2.36s, the temperature field at the end stage of braking presents approximately axi-symmetric due to the decreasing of heat generated ratio on the frictional surface and convection in the disc. Temperature in the surfaces of vanes is lower than the temperature in the work surface, because of the conductive behavior in the disc and higher convection in the vanes. The temperature field in the pad plotted in the Fig. 6 presents non-uniformity characteristics, and presents approximately axi-symmetric at the end of braking, similar with to the temperature field in the disc.

The comparison of experimental result and simulation results is shown in Fig. 7, where the curve DiscR105 gives temperature of the contact node in disc at the radius r=105mm, θ=60°. The disc node temperature curves present fluctuation in the braking operation, except DiscR81.5. The temperature fluctuation is due to heat generated in the contact region of the disc work surface and the cooling effect in the rest of the region. The temperature change at non-contact node r=81.5mm is relative slow, because there is no heat generated, only conductivity effect. During braking, there is no cooling on the contact surface of the pad; thus PadR105 is a smooth curve.

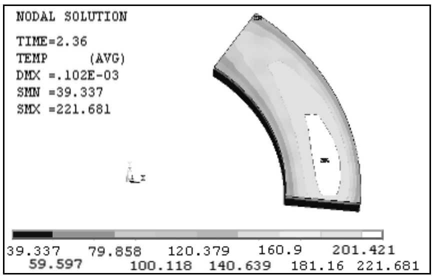

Fig. 6. Temperature field in the pad.

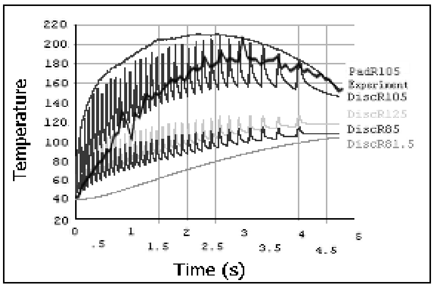

Fig. 7. Comparison of simulation result and experimental result.

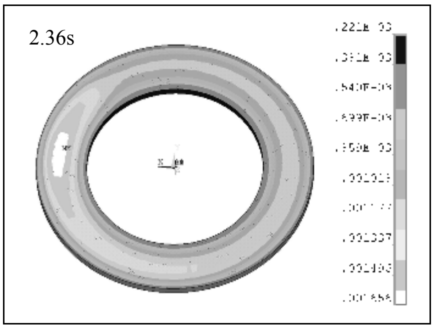

Fig. 8. Thermal strain in the ventilated brake disc.

The maximum temperature of disc measured in the experiment is 204.56℃. The simulation results are in good agreement with the experimental values as shown in Fig. 7.

4.3 Thermal strain and stress

The thermal strain contour of the brake disc is plotted in Fig. 9. According to Eqs. (6) and (7), the material nonlinear behavior is not considered, the thermal strain and thermal stress due to the thermal expansion is only related to the temperature change, because Poisson's ratio, Young's modulus and the thermal expansion coefficient are assumed as constant for isotropic. The thermal strain distribution is in accordance with the temperature distribution through a comparison of Fig. 5 and Fig. 8.

5. Conclusions

The present study investigated the thermo-mechanical behavior in the ventilated pad-disc brake during single braking.

Different from the axisymmetric model, in the 3D thermo-mechanical coupling simulation, non-axisymmetric contact pressure leads to the non-axisymmetric temperature field. The temperature field affects the thermal expansion and leads to variation of contact pressure distribution. Due to the heat generation ratio decrease, conduction in the disc and convection on the surface of brake disc, temperature field presents approximately axi-symmetric in the end stage of the braking operation. Lower temperature in the vanes is due to the effect of disc conductivity and higher convection in the vanes. The node temperature on the work surface in the disc is presenting fluctuation. The thermal strain and thermal stress distributions are in accordance with the temperature distribution

Acknowledgment

This research was supported by 2007 Yeungnam University Research Support Program.

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