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Thermomechanical modelling of dry contacts in automotive disc brake

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ABSTRACT

The objective of this study is to analyze the thermal behavior of the full and ventilated brake discs of the vehicles using computing code ANSYS. The modeling of the temperature distribution in the disc brake is used to identify all the factors and the entering parameters concerned at the time of the braking operation such as the type of braking, the geometric design of the disc and the used material. The numerical simulation for the coupled transient thermal field and stress field is carried out by sequentially thermal-structural coupled method based on ANSYS to evaluate the stress fields and deformations which are established in the disc with the pressure on the pads. The results obtained by the simulation are satisfactory when compared with those of the specialized literature.

1. Introduction

The modeling of the problems involved in the phenomena of transfers in general and of thermals in particular is of primary importance, on the one hand, for the phase study or design of a product, and on the other hand, for the follow-up of the product in phase of operation. Parallel to technological progress, of the significant projections were born in the field of the transfers of heat and of mass, and sciences related to thermals in particular and this discipline have developed for a few decades at intervals raised in many sectors :nuclear power, space, aeronautical, automobile, petro chemistry, etc [1].

In 2002, Nakatsuji et al. [2] did a study on the initiation of hair-like cracks which formed around small holes in the flange of one-piece discs during overloading conditions. The study showed that thermally induced cyclic stress strongly affects the crack initiation in the brake discs. In order to show the crack initiation mechanism, the temperature distribution at the flange had to be measured. Using the finite element method, the temperature distribution under overloading was analyzed. 3D unsteady heat transfer analyses were conducted using ANSYS. A 1/8 of the one piece disc was divided into finite elements, and the model had a half thickness due to symmetry in the thickness direction. In 2000, Valvano and Lee [3] did a study on the technique to determine the thermal distortion of a brake rotor. The severe thermal distortion of a brake rotor can affect important brake system characteristics such as the system response and brake judder propensity. As such, the accurate prediction of thermal distortions can help in the designing of a brake disc. In 1997, Hudson and Ruhl [4] did a study on the air flow through the passage of a Chrysler LH platform ventilated brake rotor. Modifications to the production rotor's vent inlet geometry are prototyped and measured in addition to the production rotor. Vent passage air flow is compared to existing correlations. With the aid of Chrysler Corporation, investigation of ventilated brake rotor vane air flow is undertaken. The goal was to measure current vane air flow and to improve this vane flow to increase brake disc cooling. Rise resulting from the temperature can strongly influence the properties of surface of materials in slip, support physicochemical and microstructural transformations and modify the rheology of the interfacial elements present in the contact [5]. Recent numerical models, presented to deal with rolling processes [6,7] have shown that the thermal gradients can attain important levels which depend on the heat dissipated by friction, the rolling speed and the heat convection coefficient .Many other works [8,9] dealt with the evaluation of temperature in solids subjected to frictional heating. The temperature distribution due to friction process necessitates a good knowledge of the contact parameters. In fact, the interface is always imperfect – due to the roughness – from a mechanical and thermal point of view. Recent theoretical and experimental works [10,11] have been developed to characterize the thermal parameters which govern the heat transfer at the vicinity of a sliding interface. In certain industrial applications, the solids are provided with surface coating. A recent study has been carried out to analyze the effect of surface coating on the thermal behavior of a solid

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Nomenclature

a	deceleration of the vehicle, ms^{-2}
A_d	disc surface swept by a brake pad, m^2
$[C]$	thermal capacity matrix, JK^{-1}
C_p	specific heat, $Jkg^{-1}K^{-1}$
E	Young modulus, GPa
g	gravitational acceleration (9.81 m/s^2)
h	convective heat transfer coefficient, $Wm^{-2}K^{-1}$
k	thermal conductivity, $Wm^{-1}K^{-1}$
$[K]$	thermal conductivity matrix, WK^{-1}
$\{L\}$	vector operator
m	mass of the vehicle, kg
$\vec{n}$	unit normal
q_0	heat flux entering the disc, W
Q	heat quantity generated during the friction, J
Q^*	heat flux specified on a surface, W
S_T	surface temperature, m^2
S_Q	surface in heat flux, m^2
S_c	surface in convection, m^2
t	time, s
T	temperature, °C
T^*	temperature specified on a surface, °C
T_f	fluid temperature, °C
T_P	temperature imposed, °C
v	initial speed of the vehicle, ms^{-1}
$\{\upsilon\}$	vector speed of mass transport
z	braking effectiveness

Greek symbols

α	thermal expansion coefficient, 1/°C
ε_p	factor load distribution on the disc surface
ν	Poisson coefficient
ρ	mass density, kgm^{-3}
υ	kinematic viscosity, m^2/s
ϕ	rate distribution of the braking forces between the front and rear axle

subjected to friction process [12]. In the braking phase, temperatures and thermal gradients are very high. This generates stresses and deformations whose consequences are manifested by the appearance and the accentuation of cracks [13,14]. This solution is applied to the problem of determining the transient temperatures reached at the friction surfaces of a disk brake when a constant deceleration is produced during braking [15].

2. Numerical modeling of the thermal problem

2.1. Equation of the problem

The first law of thermodynamics indicating the thermal conservation of energy gives:

$$C_p\left(\frac{\partial T}{\partial t}+\{\upsilon\}^T\{L\}T\right)+\{L\}^T\{Q\} = p \tag{1}$$

In our case there is not an internal source ($p = 0$), thus Eq. (1) is written:

$$pC_p\left(\frac{\partial T}{\partial t}+\{\upsilon\}^T\{L\}T\right)+\{L\}^T\{Q\} = 0 \tag{2}$$

With:

$$\{L\} = \begin{Bmatrix} \frac{\partial}{\partial x} \\ \frac{\partial}{\partial y} \\ \frac{\partial}{\partial z} \end{Bmatrix} \tag{3}$$

$$\{\upsilon\} = \begin{Bmatrix} \upsilon_x \\ \upsilon_y \\ \upsilon_z \end{Bmatrix} \tag{4}$$

The law of Fourier (2) can be written in the following matrix form:

$$\{Q\} = -[K]\{L\}T \tag{5}$$

With:

$$[K] = \begin{bmatrix} k_{xx} & 0 & 0 \\ 0 & k_{yy} & 0 \\ 0 & 0 & k_{zz} \end{bmatrix} \tag{6}$$

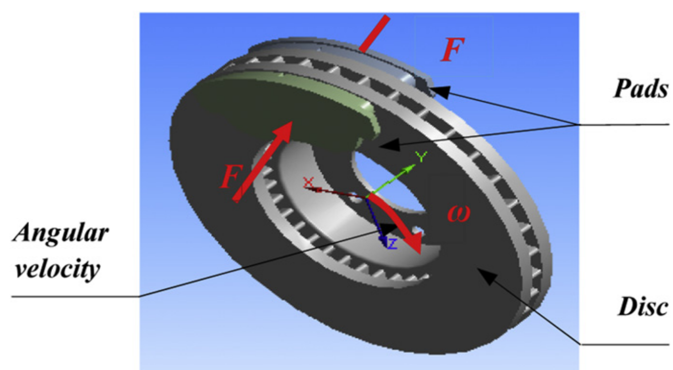

Fig. 1. Disc-pads assembly with forces applied to the disc.

k_x, k_y and k_z Represent the conditions along axes x, y, z respectively. In our case the material is isotropic thus $k_{xx} = k_{yy} = k_{zz}$

- $\{L\}$ Vector operator.
- $\{v\}$ Vector speed of mass transport.
- $[K]$ Matrix conductivity.

By combining two Eqs. (2) and (5), we obtain:

$$pC_p\left(\frac{\partial T}{\partial t}+\{\upsilon\}^T\{L\}T\right)+\{L\}^T[K]\{L\}T = 0 \tag{7}$$

Table 1
Geometrical dimensions and application parameters of automotive braking.

Item	Values
Inner disc diameter, mm	66
Outer disc diameter, mm	262
Disc thickness (TH), mm	29
Disc height (H), mm	51
Vehicle mass m, kg	1385
Initial speed v_0, km/h	28
Deceleration a, m/s^2	8
Effective rotor radius R_{rotor}, mm	100.5
Rate distribution of the braking forces Φ, %	20
Factor of charge distribution on the disc ε_p	0.5
Surface disc swept by the pad A_d, mm^2	35993