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A Model of Heat Dissipation for the Interface of Braking Systems using the Discrete Element Method

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Abstract

The objective of this study focuses on the energy dissipation by friction on the interface of a braking system. Facing the difficulty of defining velocity accommodation and thermal partition between the two bodies in contact (disc and pad), it is proposed to model the third body layer using circular particles. The heat is generated by contacts with friction between the grains and is dissipated in the first bodies. This coupling micro-macro model aims to determine the thermal resistance of pad-disc interface. From a numerical point of view, we propose a strategy of short-circuit of mechanical calculations to accelerate thermal processing in the DEM code. The results are compared with classical models using finite element method.

Keywords: heat dissipation, friction, discrete element method, contact interface, braking system.

1 Introduction

In frictional braking systems, the thermal problem is of an eminent importance. The heat generated during braking involves temperature rise in the brake components that affect brake performances and modify friction properties and wear of materials. Moreover, two major difficulties remain in studying the thermal problem of frictional problems: the coupling phenomena between tribological, mechanical and thermal behaviour with material degradation; and the multiscale aspect, from the local heat generation to the volumic thermal dissipation [1-2]. On a tribological point of view, size of real contact surface is highly reduced compared to apparent contact area. On this real contact surface, a relative speed accommodation takes place by first body surfaces degradation (disc and pads) and third body shearing [3].

In the thermal problem, it is commonly assumed that the kinetic energy is mainly converted to heat. The difficulty remains in the effective location of heat generation and to the thermal gradient between the first bodies in contact due to non-continuous contact (in the case of a disc-pad for example), to the interface layer (third body), etc. [4-6]. A way of modelling this non continuous thermal problem across the contact interface is introducing a thermal contact resistance or volumic third body layer [4, 7-10]. Nevertheless parameters of these models are difficult to identify while including complex interface layer effects. The idea of this paper is to base the determination of the thermal layer properties by studying microscopic phenomena in the contact interface which can lead to non-continuous contact. It is proposed to consider the granular nature of the interface by using the Discrete Element Method (DEM) which could help understanding physical phenomena at fine-scale and to approach contact variations over time.

This paper presents an evolution of two-dimensional (2D) contact interface, based on the tribological circuit proposed by Desplanques and Degallaix [4] for the application in automotive and rail braking. In this study, we investigate the recirculation flow on the contact interface by relative translation of the first body. The numerical tool is a software based on DEM whose originality consists in using as solver bipotential contact [12]. The present configuration is limited to spherical, non-deformable and non-penetrable particles. The advantage of this approach is that it not only investigates on the important number of modelled particles, but also on the extension of three-dimensional model which is easy to implement. The calculation cycle is a "step by step" algorithm requiring the repetition of a solving scheme. The equation of motion is discretized via a time integration scheme. At the beginning of each computation cycle, interactions between particles are determined from the position of particles. Indeed, these interactions are formed and disappear during the simulation based on the relative particles locations and the interaction impacts. Once identified interactions, a contact law is applied between the particles. For particulate media, force-displacement law is applied to each contact between two particles to calculate the intensities of contact forces resulting from relative motion and the constitutive model. When all the contact forces have been evaluated and the resultant forces and moments of each particle have been calculated, the equation of motion is solved to update the new positions and velocities.

For the thermal problem, the heat is generated by friction between the particles. The heating flow between particles occurs through a contact conductance, calculated from the Hertz theory [13]. Diffusion is also calculated through the first bodies on which convective transfers with the environment are introduced.

2 Model of contact interface

The system studied consists of a disc and a pad represented by two solid with an intermediate granular layer as in [14]. In this geometry of shearing plane (Figure 1), the granular material is sheared between two parallel rough walls. *Ox* is the flow direction and *Oy* the orthogonal one. The first lower body is kept moving at the velocity *V*. The system is compressed by imposing a vertical load *F* on the first upper body. Geometric periodicity conditions are imposed on flow boundaries. A particle that leaves the border is immediately replaced by another one on the opposite side with the same physical and dynamics properties (Figure 1). This technique allows us modelling an infinite space from a limited number of particles.

L is the length of the simulated particle flow, the size is chosen so that the edge effects are negligible. In this part, the wall roughness is modelled by assembling particles having the same characteristics with which one is in motion. According to [5], thermal modelling of braking system can be treated as horizontal symmetrical model.

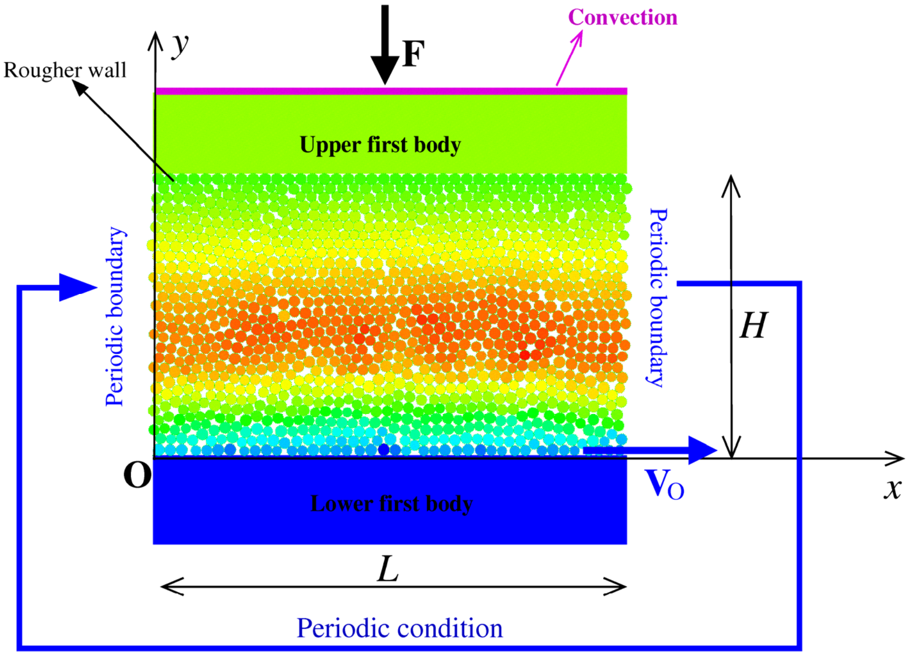

Figure 1. Pad/disc interface mode

We are therefore interested in a two-dimensional granular media subjected to a shearing by imposed velocity. These particles are characterized by a local friction coefficient μ and a restitution coefficient e; other parameters are given in Table 1. Material properties correspond to the pad material ones, with low thermal conductivity, compared to the disc one.

Particle number		1200	
Average diameter		1	μm
Third bodies thickness	H	12	μm
Length	L	20	μm
Pad thickness		9	mm
Disc thickness		18	mm
Density	ρ	2500	kg/m^3
Poisson ratio	ν	0.29	
Young modulus	E	196	GPa
Friction coefficient	μ	0.3	
Restitution coefficient	e	0.8	
Thermal conductivity	λ	5	W/(mK)
Heat capacity	C_P	900	J/(kg K)

Table 1. Parameter of model, third bodies and of upper first body (pad) used in simulations.

Main properties of disc are detailed in Table 2.

Density	ρ	7800	kg/m^3
Thermal conductivity	λ	43.5	W/(mK)
Heat capacity	C_P	440	J/(kg K)
Convection coefficient	h	20	W/ (m^2K)

Table 2. Parameter of lower first body (disc) used in simulations.

3 Mechanical energy dissipation and heat transfers

3.1 Energy balance

Model of heat transfer by conduction, convection and heat generation by friction have been detailed and validated in previous papers [15, 16].

In agreement with these studies, the variation of temperature of a particle Ω_i is related to the dissipation of mechanical energy, to the heat transfer by contact with neighbouring bodies and also to the convective transfer with the environment. So, the dynamic of heat transfer of a particle Ω_i contacting with neighbouring particles α, is given by the expression:

$$m_i C_{P_i} \frac{dT_i}{dt} = \sum_1^{\alpha} \left(H_C^{ij}(T_j - T_i) + \frac{1}{2}\Phi_\mu^{ij} \right) + \Phi_v^i \tag{1}$$

where m_i and C_p are respectively the mass and the heat capacity for Ω_i.

In shear cells, the role of heat transport is dominant. The experimental study of Wang and Campbell [17] showed that, at low shear rates, the contact stress chain is broken, thus affecting the conduction paths of heat transfer. However, when the shear rate increases, the self-diffusion coefficient increases and affects on the thermal conductance. More recently, this behavior was observed by Morelus [18] on a granular bed in motion. Noting that the local shear rate varies around the global value [19], by accepting the experimental correlation in [17], one suppose that the thermal conductance H_C increases linearly with global shear rate in accordance with:

$$H_C = H_0 + b_C \frac{V}{H} \tag{2}$$

where b_C is a constant which depends on the solid fraction and H_0 is the thermal conductance calculated from the Hertz theory [13]. The coefficient H_0, which is a function of the compression force, refers to the ability for two materials in contact to transfer heat through their mutual interface. In two-dimensional case, contact conductance between particles Ω_i and Ω_j is modelled by:

$$H_0^{ij} = 2\lambda\sqrt{2aL_C} = 2\lambda \left(\frac{8r_n^{ij}a^*}{\pi E^*} \right)^{\frac{1}{4}} \tag{3}$$

with λ the thermal conductivity, L_C is the length of the cylinder. a represents the radius of contact area, calculated from the Hertz theory. a^* is the effective radius, r_n the normal contact force and E^* the effective Young's modulus.

Φ_μ given by (4) represents the energy flow dissipated by friction and is proportional to the sliding velocity $\dot{u}_t^{ij}$ and to the tangential contact force:

$$\Phi_{\mu}^{ij} = \dot{u}_t^{ij} r_t^{ij} \tag{4}$$

Φ_V representing the convective flow is expressed by:

$$\Phi_v^{i} = hS_V(T_i - T_a) \tag{5}$$

where h denotes the convection coefficient, S_v the convective surface and T_a the room temperature.

Equation (1) assumes that the temperature of each particle is homogeneous. It is therefore necessary to verify that the heat transfer's resistance (conduction) through a particle is significantly lower than the contact resistance between two particles. This condition is represented by the Biot number (*Bi*) [13]. The heat equation is solved with a small time step (1.10^{-8}s) to assume that the temperature of each particle changes slowly so that thermal perturbations do not propagate further than its immediate neighbours during one time step. This second condition can be expressed by:

$$\frac{\Delta t H_C}{mC_P} << 1 \tag{6}$$

3.2 Heat transfer by conduction in first bodies

From experimental investigations for railway brakes, Dufrénoy [20] showed the appearance of stable hot bands, whose sizes are less than 20 mm, on the surface of the disc during a braking time more than 0.5s. Majcherczak [21] has found that heat propagation by conduction through the first bodies is dominant. Taking into account the size of our model, we consider that the heat flow in the direction Ox is negligible compared to the outflow of the surface (Oy direction). Note that this one is similar in spirit to the Newcomb and Limpert models in [5]. The equation of temperature evolution is:

$$\lambda \frac{\partial^2 T}{\partial^2 y} = \rho C_P \frac{\partial T}{\partial t} \tag{7}$$

We establish the longitudinal bands of length L and uniform height Δy, which pass through the first body. The finite differences allow us to write:

$$\frac{T_{j+1}^t - 2T_j^t + T_{j-1}^t}{\Delta y^2} = \frac{\rho C_P}{\lambda} \frac{T_j^{\Delta t + t} - T_j^t}{\Delta t} \tag{8}$$

where $T_{j-1}^t, T_j^t, T_{j+1}^t$ represent respectively the temperature at node *j-1, j* and *j+1* (Figure 2).

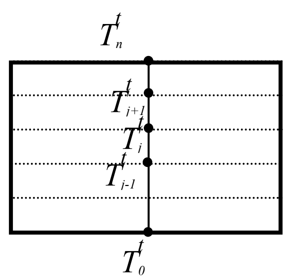

Figure 2. 1D model by centered finite differences.

Note that T_0^t is the average temperature of roughness wall. A symmetry condition at the lower first body is assumed. The thermal convection is taken into account on the contact surface with the environment of the first upper body (Figure 1). The Neumann boundary condition is written as:

$$-\lambda \frac{\partial T_n^t}{\partial y} = h(T_n^t - T_a) \tag{9}$$

This equation is discretized by this expression:

$$-\lambda \frac{T_n^{t+\Delta t} - T_n^t}{\Delta y} = h(T_n^t - T_a) \tag{10}$$

4 Numerical simulation and discussions

4.1 Braking system modelled by DEM: Results and discussions

The DEM simulation takes classically long computing time. However, we note that after a short start-up phase, the shearing stabilizes. We propose to store a cycle of stabilized mechanical results and to use it for successive cycles, over tile, where only the thermal calculations are performed. It allows reducing computational cost.

Now back to the representative model (Figure 1), we impose a normal force F of 40N (equivalent pressure 2 MPa), a shearing velocity of 1 m/s. We then present the results obtained for a mechanical cycle (2.10^{-5}s). In Figure 3, the variation of the reaction component Σr_{yy} normalized by the force F is presented. It indicates that, as expected, this ratio is near 1; the imposed force F is therefore transmitted by the granular medium.

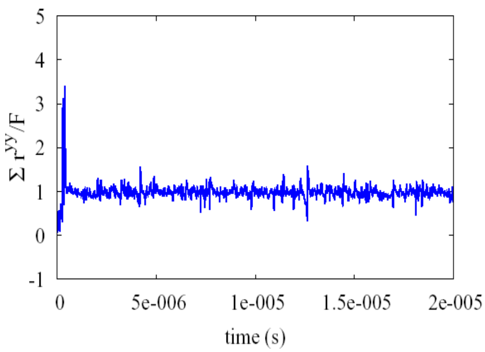

Figure 3. Variation of reactions sum along Oy normalized by force F

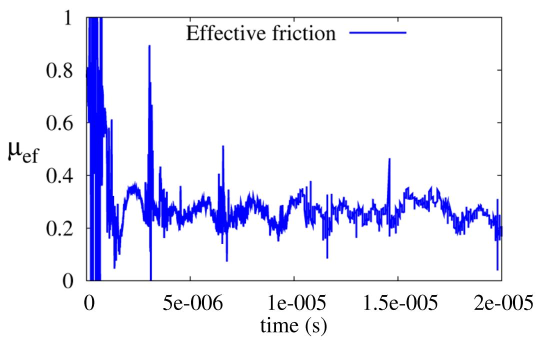

Figure 4. Evolution of effective friction during the mechanical cycle.

We then the effective friction coefficient μ_{ef}, which is defined by:

$$\mu_{ef} = \frac{\sum r_{xy}}{F}\,. \tag{11}$$

with Σr_{xy} the sum of the contact reaction's projections along Ox between the bottom wall and the particles of granular flow.

Figure 4 shows the variation of friction coefficient with time. After the start-up phase, it stabilizes.

In order to verify the energy balance of the system, we show, in the figure 5, the evolution of the mechanical energy compared to the variation of thermal energy generated by friction. The agreement is generally observed.

By using the strategy of recuperation of mechanical results in a cycle (equivalent time of 2.10^{-5}s), the temperature in the first bodies and the third body may be determined. Simulations were performed during 1s.

Figure 6 shows a thermal map of the temperature obtained at the end of the simulation. The maximal temperature is around 25°C in the third body layer. The temperature distribution is heterogeneous, with most heating particles near the pad surface. It could be explained by a higher diffusivity of the disc, with much more heat absorption.

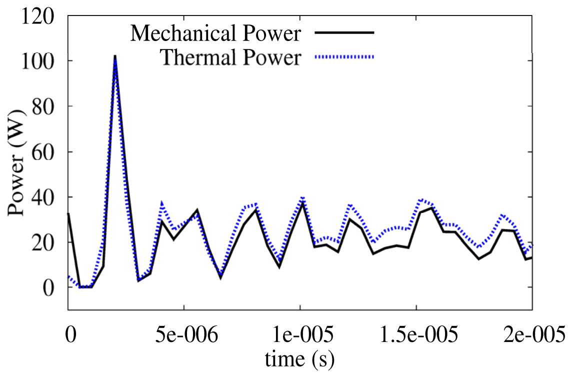

Figure 5. Verification of energy balance of the system.

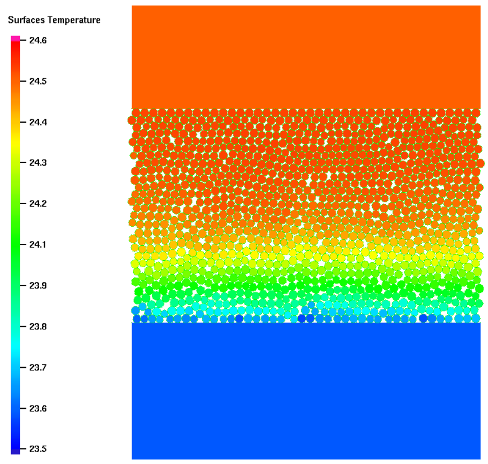

Figure 6. Thermal map after 1.0s heating.

In order to “validate” the DEM results, a simulation using a Finite Element Model (FEM) has been used as described in [5]. In this case, the interface layer is not taken into account in the FEM analysis (assumption of equal temperature of contact surfaces). Comparison of the results of FEM and DEM analyzes is presented in figures 7 and 8. The temperature distribution after 1s (figure 7) shows an equal temperature at the disc/pad interface with the FEM analysis while the DEM analysis gives a maximal temperature in the third body with a jump around 2°C between the disc and pad surfaces. In the first bodies, the temperature distribution shows a good agreement between the two analyses.

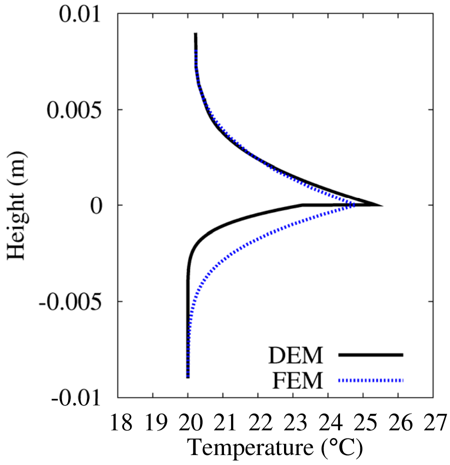

Figure 7. Temperature profile at 1.0s heating.

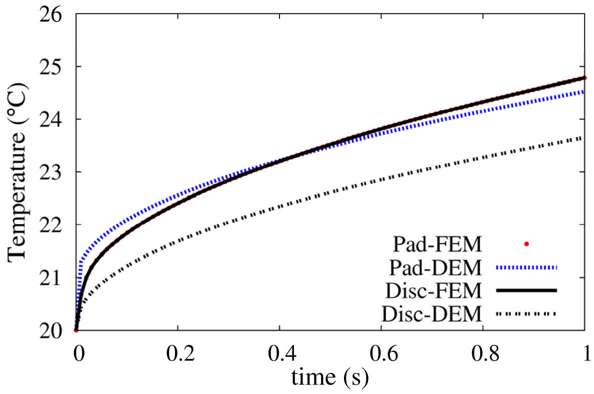

Figure 8. Thermal results of DEM compared to FEM assuming equal surface temperature.

According to DEM results, the third body interface layer acts as a thermal barrier and creates a thermal resistance. This tendency is in agreement with experimental observations found in literature [22]. We see that the equivalent thermal contact conductance (inverse of thermal resistance), which is equal $5x10^5$ $W/m^2.K$, is relatively high compared to those usually assumed [5,8,10]. That can be explained by a high density of the particles allowing good heat flow conduction. Equivalent properties remain closed to the pad material ones, which are probably physically higher than the ones of the third body.

4.2 Application to automotive braking

From the results previously obtained in terms of thermal contact conductance, a simulation of an automotive braking has been performed using the FEM analysis. Data of brake geometry and braking configuration are given in table 3 and are the same as those used in [5].

Inner disc diameter	132	mm
Outer disc diameter	227	mm
Disc thickness	11	mm
Pad thickness	10	mm
Disc thickness	18	mm
Initial speed	100	km/h
Braking time	3.96	s
Deceleration	7	m/s^2
Total energy	165	kJ
Peak power time	0.5	s
Sliding length	37	mm
Mean sliding radius	94.5	mm
Convection coefficient	60	$W/(m^2K)$

Table 3. Parameters of automotive brake application

The disc and pad surfaces temperatures evolutions are presented in figure 9. Two cases are compared: evolution of disc/pad surface temperatures with the assumption of equal temperature and taking into a third body layer with parameters identified from the DEM model (equivalent thermal conductance of 5 x10^5 W/m^2.K).

As the disc absorbs main part of the heat flux, the disc surface temperature evolution is closed for the two assumptions. Introduction of thermal conductance leads to a difference of temperature between the 2 surfaces of almost 30°C. That is consistent with previous numerical and experimental results [5,8,10], even if it remains lower, probably due to the same reasons as previously discussed (relative good density of particles and properties closed to the pad material ones). Nevertheless this result shows that the DEM model is relevant for improving the interface modelling.

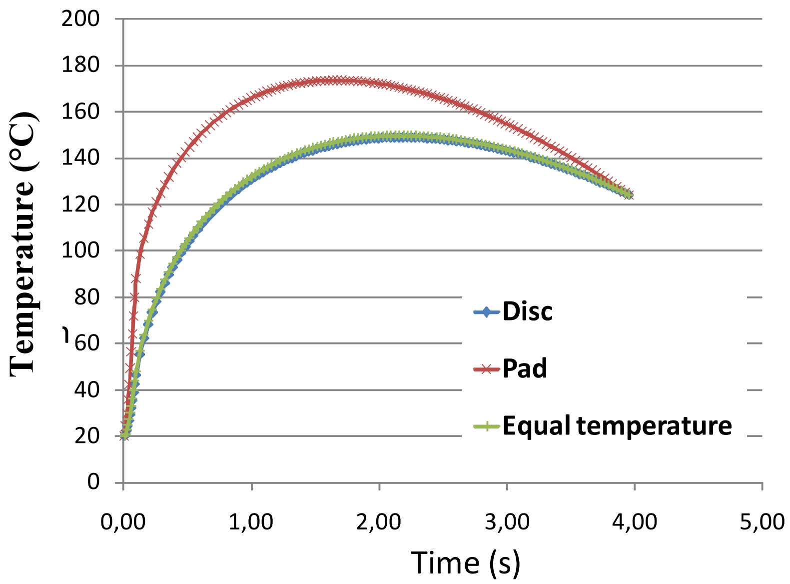

Figure 9. Evolution of disc/pad surface temperatures with the FEM model without and with thermal conductance, identified from the DEM analysis.

5 Conclusions

The present work focuses on the modelling of heat transfer and heat generation by friction in a braking system by using DEM approach for the disc/pad interface layer. A strategy of recuperation of the mechanical results proposed in this paper allows us to determine the evolution of temperature during 1s with a reduced computing time. Results show a maximal temperature in the interface layer and a temperature jump between disc and pad surfaces. The heat profile through the height of the interface layer shows a non uniform distribution with a maximal value near the disc surface.

Finally, a simulation of an automotive braking application has been performed using the FEM analysis including the identified thermal contact resistance. Results are in the same tendency as previous works estimating thermal gradient between the friction surfaces. Such results show that the DEM approach is relevant for the understanding of the location of heat generation and for the identification of the third body thermal behaviour. The model is more physically based as those found in the literature, even if properties of particles have still to be better characterized.

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