

Interactions between Third-Body Flows and Localisation Phenomena during Railway High-Energy Stop Braking

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ABSTRACT

During railway braking, dissipation of high energy due to friction leads to transient and localised thermal phenomena such as hot bands and hot spots. These localisation phenomena interact with third body flows and friction mechanisms activated in the contact. To study these couplings, an experimental approach has been developed, based on an inertial tribometer able to reproduce high thermo-mechanical brake-disc loadings.

This paper focuses on the coupling between flows of third-body and hot-band migration. Transient localised thermal phenomena are described for high-energy stop-braking. The monitoring of the disc track during friction in the visible and infrared wavelengths evidences the interactions between third-body flows and hot-band migration. The consequences relatively to load-bearing areas and third-body compaction in the contact are studied, allowing the proposal of a synthetic diagram of the coupling between friction mechanisms, third body flows, hot-band migration and first-body distortion.

1 INTRODUCTION

In railway applications, the braking function is an industrial problem of strategic importance, in terms of reliability, safety and specifications. Nevertheless, Current braking system and materials are now close to their limit of use, in terms of energy to be dissipated, due to the increases of speeds, masses, number of brakings, and the decrease of the intervals between 2 brakings as well as the duration between two stations.

As a consequence, during railway braking, a very high energy is dissipated by friction during a relatively short time, implying local thermal phenomena with high thermal gradients such as hot bands and hot spots [1,2]. The aim of this paper is to study interactions between local friction mechanisms and hot-band localisation phenomena during high-energy stop braking. We studied the friction phenomena using the third-body approach proposed by Godet [3,4]. We have shown that during braking a tribological circuit is activated and friction of pad and disc materials involves in the contact a third-body mainly composed of flat plates, which bear the load, fed by the flows of particles [5,6]. Figure 1 highlights third-body flows of the tribological circuit induced by the disc rotation, in particular the circumferential recirculation flow which permits to reintroduce particles in the contact. Several authors also studied third-body layers for automotive brake applications, the third body involving primary contacts, plateaus and patches [7-9].

To get information on the local and transient friction mechanisms involved in the contact, the use of a laboratory testing-device is more pertinent than a full-scale test bench. The section 2 presents the braking tribometer used in this study and the scale-conversion rule. Experimental conditions are given in section 3, while transient localised thermal phenomena activated during stop-braking are presented in section 4. The interactions between tribological mechanisms and transient thermal phenomena are analysed in section 5

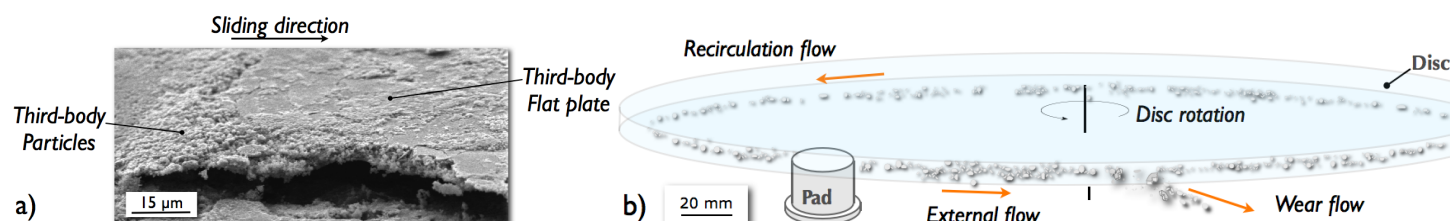


Fig. 1: Tribological circuit, a) Scanning electron microscopy (SEM) image of a third-body flat plate fed by particles, b) Flows of third-body particles activated by the disc rotation

in the case of hot-band migration. Finally, the section 6 gives a discussion about the coupling between physical mechanisms, third-body flows, hot-band migration and first-body distortion.

2. EXPERIMENTS

2.1 BRAKING TRIBOMETER

A specific braking tribometer has been designed in order to obtain at reduced-scale pad-disc contact conditions as close as possible to those observed at full-scale in railway applications [5,10]. Figure 2a shows its architecture. Using the braking torque measurement and the motor speed control, the deceleration is controlled in order to simulate the dissipation of kinetic energy. Therefore, for a stop-braking test, the mean contact pressure p is kept constant and the sliding velocity $v(t)$ decreases from an initial value v_0 (at $t = 0$) until standstill (at $t = t_f$), while the surface temperature $\theta(t)$ is obtained by means of heat dissipation by friction (Fig. 2b).

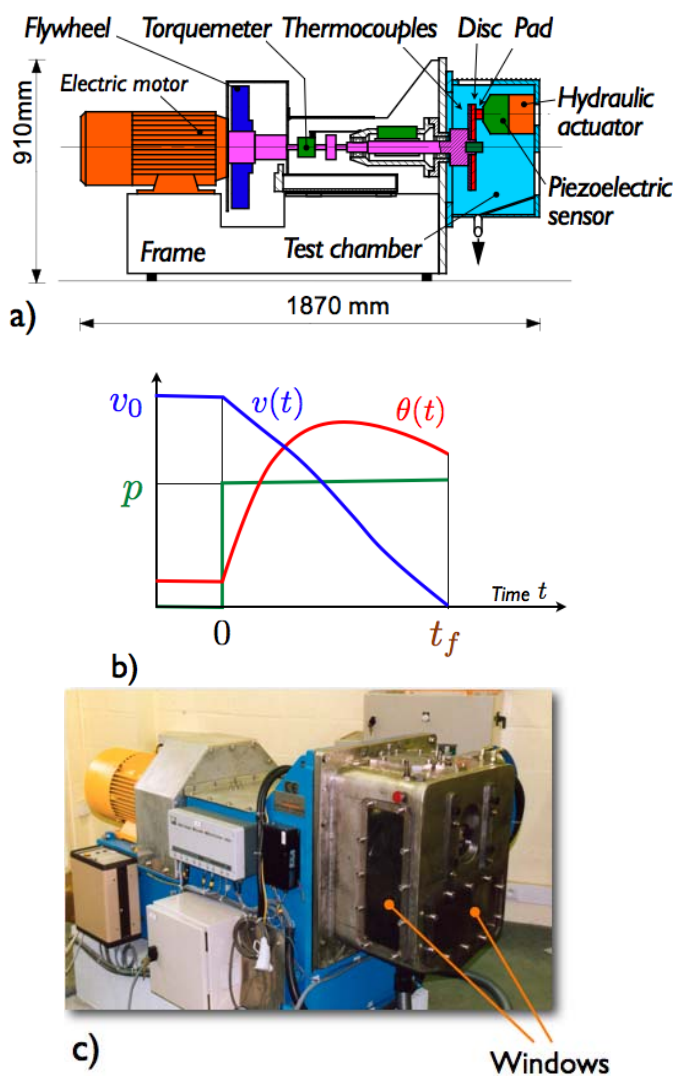


Fig. 2: Braking tribometer: a) Diagram, b) Stop-braking simulation, c) View.

The rotation speed is measured by a motor tachometer, the braking torque by a torquemeter, the pad-disc force by a load cell, and disc and pad mass temperatures by embedded thermocouples. Figure 2c shows the braking tribometer with the test chamber in the foreground: the windows set in the walls of the test chamber allow the use of specific instrumentations, as an optical pyrometer, a high speed video camera, and an infrared camera. Table 1 gives the main characteristics of the braking tribometer. The speed range permits reaching the high sliding velocities of the TGV very high-speed train. The kinetic energy dissipated during a stop braking can reach more than 1 MJ.

Tab. 1: Braking tribometer: main characteristics.

Electric motor power	11.5 kW
Rotation speed	up to 4 000 rpm
Normal load	up to 1 kN
Simulated inertia range	[1.2-20] kg m ²
Sliding velocity	up to 70 m s ⁻¹
Dissipated energy	up to 1.3 MJ

2.2 SCALE CONVERSION RULE

The scale of the tribometer was based on a similitude factor k , defined as the ratio of the energy dissipated at reduced scale q over the energy dissipated at full-scale Q . To reproduce the full-scale thermal loading in the disc pad contact, an ideal way is to respect the same similitude factor k for the disc rubbing surfaces, s_d and S_d , and for the pad friction surfaces, s_p and S_p , respectively at reduced and full scale, as shown in Fig. 3.

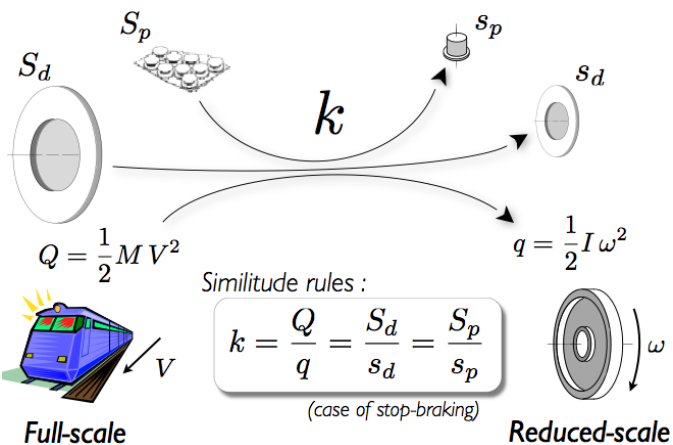


Fig. 3: Similitude rules.

Nonetheless, the simultaneous fulfillment of these three rules is very difficult, and so it is not easy to reproduce thermal conditions occurring at full scale. As an example, for the UIC standard railway brake pad, the pad surface covers one sixth of the disc-rubbing surface. To keep this ratio on the braking tribometer, the disk-rubbing surface would have a small radius, leading to a

pad location near the centre of rotation. This would imply a very high frequency of rotation and high velocity gradients in the contact.

To avoid this inconvenience, a scale conversion rule [11,12] has been developed using Newcomb's model, which gives the average temperature change of the friction surface $\Delta\theta(t)$ during a stop braking [13]:

$$\Delta\theta(t) = \frac{2\varphi_d(0)}{\sqrt{\pi}\xi_d T_f} \left(1 - \frac{2}{3}t\right) \sqrt{t}$$

where time t varies from 0 to T_f from the start to the end of braking, $\varphi_d(0)$ is the density of flux absorbed by the disc at the beginning of braking and ξ_d is the thermal effusivity of the disc material. With the respect of the following relation:

$$p v_0 \sqrt{t_f} = \gamma P V_0 \sqrt{T_f}$$

where P and p , V_0 and v_0 , T_f and t_f are respectively the mean contact pressure, the initial sliding velocity and the stop-braking duration at full- and reduced-scale, the same mean contact temperature variation $\Delta\theta(t)$ is obtained at the two scales. In this relation, γ is a constant, characteristic of material effusivities and geometries of the pad (symbol p) and disc (symbol d), as follows:

$$\gamma = \frac{s_d \xi_d + s_p \xi_p}{n s_d \xi_d + s_p \xi_p} \quad \text{with } n = \frac{s_p}{s_p s_d}$$

A non-dimensional conversion rule is then obtained:

$$\hat{p} \hat{v}_0 \sqrt{\hat{t}_f} = \gamma$$

by dividing the two sides of the conversion rule with $P V_0 \sqrt{T_f}$. The dimensionless parametric triplet $(\hat{p}, \hat{v}_0, \hat{t}_f)$ represents the non-dimensional experimental parameters. Hence the thermal representativeness of reduced-scale tests is based on compromises between these parameters.

All the dimensionless parametric triplets $(\hat{p}, \hat{v}_0, \hat{t}_f)$ satisfying the scale conversion rule correspond to the iso- $\Delta\theta$ surface drawn on Figure 5. That means they lead to the same mean contact temperature variation $\Delta\theta$ as at full-scale. The origin O corresponds to the triplet $(1,1,1)$, ie same mean contact pressure, same initial sliding velocity and same stop-braking duration at both scales. Using this triplet leads to a lower mean contact temperature variation at reduced-scale than at full-scale. Points P_1 and P_2 , on the surface, conform to the full-scale mean contact temperature variation. Point P_3 is a compromise for stop-braking duration ($\hat{t}_f = \gamma^2$; $t_f > T_f$), allowing conformity with mean contact pressure ($\hat{p} = 1$;

$p = P$) and initial sliding velocity ($\hat{v}_0 = 1$; $v_0 = V_0$); points P_1 and P_2 are compromises respectively for mean contact pressure ($\hat{p} = \gamma$) and initial sliding velocity ($\hat{v}_0 = \gamma$) in order to conform to stop-braking duration ($\hat{t}_f = 1$; $t_f = T_f$).

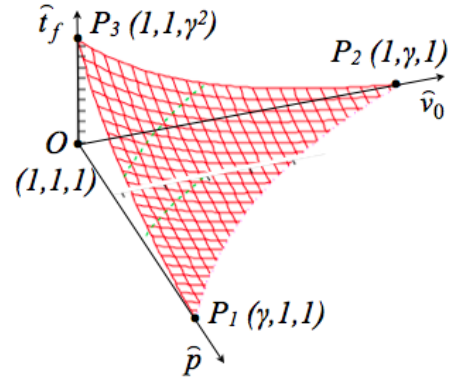


Fig. 4: Conversion rule iso- $\Delta\theta$ surface.

At last, all the points on the iso- $\Delta\theta$ surface are not strictly equivalent relatively to the thermal loading. It has to be noted that the choice of experimental parameters has a considerable influence on the obtained thermal contact loading: a long stop-braking duration leads to a more homogenous thermal loading of the friction surface, while a shorter braking duration leads to higher thermal gradients. As shown on Fig. 5, the respect of braking duration ($\hat{t}_f = 1$, point O) leads to the same density of dissipated energy and heat flux absorbed by the disc at full-scale ($\hat{q}_d = \hat{\varphi}_d = 1$). Then, thermal loadings are very close at both scales. On the contrary, as example for point P ($\hat{t}_f > 1$), the mean surface temperature $\Delta\theta$ is reached thanks to a higher dissipated energy ($\hat{q}_d > 1$) and a lower density of flux ($\hat{\varphi}_d < 1$) [11,12].

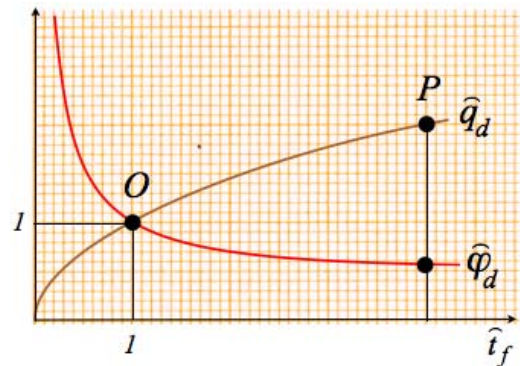


Fig. 5: Dimensionless densities of energy $\hat{q}_d$ and heat flux $\hat{\varphi}_d$ absorbed by the disc versus dimensionless braking duration $\hat{t}_f$.