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# Identification of the heat flux generated by friction in an aircraft braking system

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ABSTRACT

In this paper, we present an estimation of the heat flux generated by friction under real aircraft braking conditions using an inverse method. The estimation is performed considering an assumption of 1D transient model and multiple interfaces. This model takes into account a non-perfect description of the thermal contact. Then, an identification of the generated heat flux by friction in the different interfaces from experimental data is performed using a linear temporal evolution parameterization of this parameter. The reconstruction of the thermal field from identified generated heat fluxes provides some very low residues. The comparison between the thermal energy identified by the inverse method and mechanical energy absorbed on the test bench validates the results.

## 1. Introduction

Modern civil planes are prone to take-off and land several times a day. Companies' willingness to reduce turnaround time (TAT) and to reduce the weight of the brake, leads to very high loads and temperature in carbon disks. Although temperature estimation or measurement seems natural, friction heat flux at disk contacts is not well-defined. Besides, 3D model robustness and accuracy are non-optimal, since the causes of heat fields are not perfectly determined. The aim of this study is to characterize in-landing boundary conditions at carbon disk surfaces from thermal measurements on an experimental device. Among all the type of inverse heat conduction problems (IHCP), we are interested here in the identification of an unknown heat flux boundary condition. Inverse heat conduction problems are highly ill-posed, to the extent that any small input modification results in a pronounced modification of the solution. Many investigations have presented several methods in improving the stability of IHCP [1–7]. Although many works deal with experimental and numerical analysis of direct heat transfer problem in the automotive or rail brake disc [8–14], very few studies concerning IHCP in disc brakes have been published. Furthermore, most studies often use numerical simulations as data [15]. Ghadimi [16] presents an inverse algorithm based on the Artificial Neural Networks and the Sequential Function in order to estimate the heat flux absorbed by the locomotive brake disc. However, all these references present braking systems comprising only one disc. Fittingly, aircraft braking systems are composed of several discs in friction with multiple interfaces. Few references [17–21] deal with this configuration and all of them develop Finite Element models. However, none specify the generated heat fluxes to be considered in these numerical models in order to determine the resulting temperature fields. In this study, we aim to identify these generated heat fluxes from experimental data, using one dimensional IHCP method. In order to solve the inverse problem, a simple direct model has to be developed. A Finite Difference Method (FDM) has been chosen to solve the direct problem.

Different regularization techniques can be used to stabilize the IHCP procedure as truncation or penalization [22], [23]. In order to identify the evolution of parameters, the estimation could be based on the function specification method using future time step [2]. Finally, for the problem of identifying the evolution of the heat flux generated by friction in an aircraft braking system, the parameterization of this evolution is used in the IHCP [24–26].

## 2. Brake thermal direct model

### 2.1. Presentation

For this study, the brake core is composed of nine carbon discs, including five stators and four rotors. The wheel and brake assembly is

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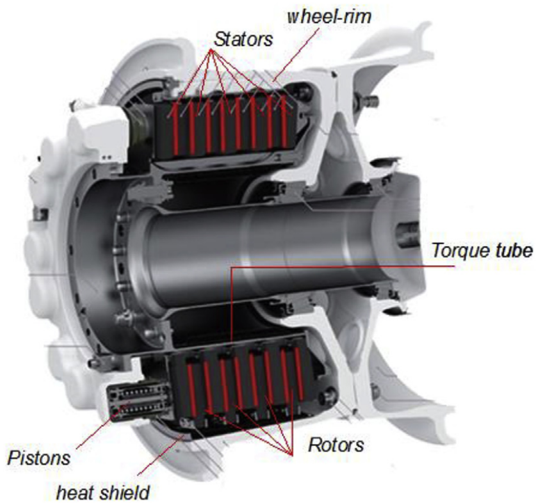


**Fig. 1.** Studied aircraft braking system (see also Fig. 2).

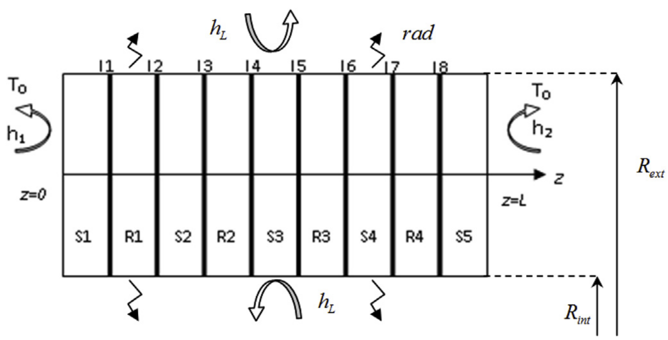


**Fig. 2.** Representation of the studied 1D system.

presented in Fig. 1. We propose to develop a simple thermal 1D model on the braking period, while the wheel is in rotation. During this period, we can consider that only the discs are subject to high temperature gradients. After this phase, which lasts a few seconds, the heat will be transferred to the other part of the braking system (the wheel, torque tube, piston housing …). The short scale of time allows us to consider brake surrounding temperatures, close to initial conditions.

During this time scale, a heat flux is generated by friction on the sliding contact surfaces between rotors and stators. Spatial and temporal heat flux distribution will condition the heat gradient within the brake system, and that is why its characterization is crucial. Currently, hypotheses based on brake mechanical behavior and efficiency are made but they have never been verified experimentally.

### 2.2. System modeling

At each interface between stators and rotors, the thermal contact is supposed to be non-perfect. Bardon [27] proposed an expression to describe the interfacial heat exchange. This approach introduces two contact parameters in addition to the generated heat flux: the sliding thermal contact resistance and the intrinsic heat partition coefficient. Both parameters are dependent upon the thermal constriction resistances. An equivalent expression has been proposed by Tseng [28] to study heat transfer for rolling systems. During the braking, heat propagation is supposed to be unidirectional in the entire system, following the transverse direction to the discs as is shown in Fig. 2. *kth* rotor is noted as $R_k$, *kth* stator as $S_k$, and the *kth* sliding interface as $I_k$.

Carbon discs can then be considered as simple successive rings, with $R_{ext}$ as their external radius, $R_{\text{int}}$ as their internal radius, and with a non-perfect thermal contact model between each disk. At first, boundary conditions are set with Fourier conditions at left and right extreme boundaries. The heat equation is established in eq. (1) for our one-dimensional problem in *z* direction. Moussa et al. [7] used a similar approach in their study of the heat transfer at the grinding interface between a glass plate and a sintered diamond wheel.

$$\begin{cases} \frac{1}{a}\frac{\partial T}{\partial t} = \frac{\partial^2 T}{\partial z^2} - \frac{h_L}{\lambda}C_0(T(z,t) - T_0) - \frac{\sigma\varepsilon_L}{\lambda}C_1(T^4(z,t) - T_S^4) \\ \qquad - \frac{\sigma\varepsilon_L}{\lambda}C_2(T^4(z,t) - T_T^4) \\ \lambda\left.\frac{\partial T}{\partial z}\right|_{z=0} = h_1(T - T_0), \quad -\lambda\left.\frac{\partial T}{\partial z}\right|_{z=L} = h_2(T - T_0), \quad T(z, t=0) = T_0 \\ \varphi_{l_k}(t) = \alpha_k\varphi_{gk}(t) + \frac{T_{cl_k}(t) - T_{cr_k}(t)}{R_{TSC_k}}, \\ \varphi_{gk} = \varphi_{l_k}(t) + \varphi_{r_k}(t) \quad \text{on each interface } I_k \\ \text{with } C_0 = \frac{2}{R_{ext} - R_{\text{int}}},\ C_1 = \frac{2R_{ext}}{(R_{ext} - R_{\text{int}})^2},\ C_2 = \frac{2R_{\text{int}}}{(R_{ext} - R_{\text{int}})^2} \end{cases} \tag{1}$$

Where $\varphi_{l_k}(t)$ and $\varphi_{r_k}(t)$ are the heat flux entering the left and the right-sided disc, respectively, $R_{TSC_k}$ is the thermal contact resistance, $\varphi_{gk}(t)$ the generated heat flux, $T_{cl_k}(t)$ and $T_{cr_k}(t)$, respectively, are the contact temperatures on the left and right at the interface *k* between a rotor and a stator at the time *t*. Because the frictional materials are identical, the local heat partition coefficient $\alpha_k$ is assumed to be equal to 0.5 for each interface according references [27, 28]. Time evolution of the generated heat flux $\varphi_{l_k}(t)$ will be discussed in part 2.3. Lateral losses by convection on the sides of the discs are taken into account with the ambient temperature, which is supposed to be constant. Lateral losses by radiation are calculated with the heat shield temperature $T_S$ (on the external radius), and with the torque tube temperature $T_T$ (on the internal radius). During the braking period, there is just a small variation of these surrounding temperatures. Therefore, we make the assumption that they remain constants for the radiation model. A parametric study is conducted in part 4 to analyze the effect of $h_L$ and $\varepsilon_L$. We show that the convective and radiative heat losses (regardless of $h_L$ and $\varepsilon_L$ physical values) are negligible comparatively to the generated heat flux. These losses are less than 3% of the friction energy.

### 2.3. Parameterization of the problem

The different thermal contact parameters are: $\varphi_{l_k}(t)$ and $R_{TSC_k}$, $\alpha_k$. The local heat partition coefficient is supposed to be equal to 0.5 (carbon/carbon sliding contact). In the aircraft braking system case, where the heat generated heat flux is important [29], the impact of the contact resistance on the temperature is insignificant. It will be difficult or impossible to identify this parameter [29], [30]. Therefore, the value of the sliding contact resistance will be imposed on $R_{TSC} = 10^{-4}m^2KW^{-1}$. The impact on the identification procedure of this assumption will be insignificant on the other parameters, under 2% if $R_{TSC} <= 10^{-3}m^2KW^{-1}$.

We propose to estimate a linear heat flux between step times. We use a generated heat flux parameterization with a hat function basis such as shown below in Fig. 3 [26]. The parameterization is an efficient regularization process for the inverse problem [25].

The choice of the time increment $\Delta t = t_{ip} - t_{ip-1}$ is based on the supposed variation form of the generated heat flux study. Indeed, if $\Delta t$ is too small, it will be difficult to identify $\varphi_{ip}$. In fact, the heat flux must have sufficiently varied over a time increment in order to identify the different function values. We note $\varphi_{ip}^k$ as the different heat flux $\varphi_{ip}$ at the interface *k*. $t_{\max}$ corresponds to the stop of the wheel.

### 2.4. Identification procedure

The different $\varphi_{ip}$ have to be identified. The time vector is imposed and is chosen from the assumed rate of heat generated flux change during the braking period. The time increment will be less in the first