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# Friction at high sliding speed of WC-6Co pin versus steel disc AISI 1045: estimation of the contact temperature

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## **ABSTRACT**

Cemented carbide based cutting tools remain widely used in machining processes for their wear resistance. Knowledge and modelling of wear processes are of prior importance to get models to predict and increase the cutting tool service life. The present paper deals with some results of a research work that study relationships between wear and temperature in the case of pin-on-disc tribological experiments. Topic of the paper is focussed on the estimation of the pin/disc contact temperature by coupling experimental measurements and computational methods.

Friction experiments are conducted with WC-6Co pins against steel discs made of an AISI 1045 grade. Furthermore, WC-6Co pins are instrumented with two type-K thermocouples. A large sliding velocity range is considered in the study: from 100m/min up to 600m/min. The present paper is focussed on the 600m/min velocity. During these tribological tests, tangential forces and thermocouple temperatures are measured and monitored. From these values, the heat flux in the WC-6Co pin is estimated by two different ways. On one hand by considering the unidirectional Fourier law, on the other hand by estimating the heat partition coefficient [1] between the disc and the pin. The heat partition coefficient is determined from a physical approach based on the consideration of the mechanical power dissipated in the contact. In both cases, the heat transfer in the pin is then modelled by finite element methods. It is necessary to perform numerical analysis to estimate the pin/disc interface temperature because of the impossibility to measure it directly during friction tests. Results of experiments and of numerical simulations are compared.

## **1. INTRODUCTION**

The friction between two surfaces induces an increase of the temperature in the contact. This temperature considerably influences mechanical behaviour and physico-chemical properties of surfaces in the friction contact [1]. In machining processes, the tribological conditions at the tool/piece contact are severe. Consequently, tool wear remains a major problem. Cemented carbide tools (like WC-6Co) are commonly used for their excellent wear resistance. However for high solicitation conditions, the frictional heat generated at the contact interfaces leads to high temperature levels, which can sometimes reach 1000°C

[2–4]. In these processes, the temperature has a large influence on wear mechanisms. Thus, the knowledge of temperature levels at the contact interfaces becomes primordial. To understand the temperature influence on the tribological behaviour of tungsten carbide based tools, tribological tests are performed. Pin on disc tests are carried out to measure the evolution of the temperature in the pin and to estimate the interface temperature at the friction contact. The measurement of the contact temperature remains impossible nowadays. In order to calculate or to estimate this temperature, several works based on experimental investigations coupled with analytical and/or numerical methods have been achieved.

The estimation of the contact temperature has received extensive attention. Ertz and Knothe [5] considered analytical and numerical models to calculate the contact temperature between a wheel and rails. They supposed that the contact between wheel and rails had an hertzian form and that the thermal transfer was unidirectional. They show that the temperature generated in the contact remains confined in fine layers close to the surface. Regarding to the wheel bulk temperature, the temperature gap in this layer can exceed 200°C and affects the mechanical behaviour of the wheel. Laraqi and al. [6] investigated the temperature distribution and the thermal constriction at the contact interface. It is very difficult to estimate accurately a contact temperature since many conditions influence the model, such as boundary conditions, transient aspect of temperature evolutions in dynamic bodies, material properties and roughness of contact surfaces.

The temperature of the contact surface depends on the nature of surfaces. The surface roughness affects initial friction behaviour and surface temperature. In the first sliding time, the flash temperature is high and can reach the temperature melting point [6–8]. To take the surface roughness and the contact geometry into account, some authors introduce different shapes for heat sources (rectangular, square, circular and elliptic) [6, 9].

Hou and Komanduri [10] investigate the surface temperature in the grinding process considering a sliding friction of rough surfaces. The more contact surfaces are homogeneous, the less the constriction problem exists. Therefore, the average temperature of the surface remains homogeneous [11, 12].

One of the difficulties in the calculation of the temperature of the contact interface is the determination of the heat partition coefficient. In real cases, this coefficient depends on time and space. Lestyán et al. [7] studied surface temperature by considering a time dependent heat partition coefficient. They developed a special iterative algorithm based on incremental finite elements technique to calculate heat partition coefficient.

The aim of the present paper is to calculate the pin surface temperature on one hand by considering unidirectional Fourier law, on the other hand by estimating the heat partition coefficient between the disc and the pin.

## **2. EXPERIMENTAL SETUP**

Tribological tests are carried out using a high speed pin on disc tribometer (Fig.1). The disc is made of an AISI 1045 steel grade and is characterised by a 170 mm diameter and a 10 mm thickness. The chemical composition of this steel grade is detailed in table 1. The cemented carbide pin is made of tungsten carbide (WC) and of a 6 wt% cobalt based binder (Co). It has a cylindrical shape (diameter = 10 mm) with a truncated conic end with a flat circular surface of 2 mm in diameter (contact surface) [see fig.2].

Two K-type thermocouples (diameter 0.5 mm) are embedded in the pin to measure temperature evolutions. Thermocouple holes are 0.6 mm in diameter. To be sure that thermocouples are well embedded in holes, a silver slat is used. The first (TC1) and the second (TC2) thermocouples are respectively embedded at 1 mm and 3mm of the pin/disc contact surface [see fig.2].

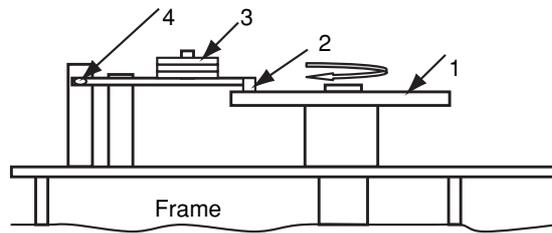


Figure 1 Tribometer description.  
1-Disc, 2-Pin, 3-Normal loading, 4-Friction sensor

Table 1 Mechanical properties and composition (wt%) of the AISI1045 steel grade

%C	%Si	%Mn	%P	%S	%Cr	%Ni	%Mo	%Cu	Re (MPa)	(0.2%) (MPa)	Rm (MPa)	A %
0,45	0,22	0,66	0,027	0,032	0,26	0,15	0,02	0,18	437		704	20,6

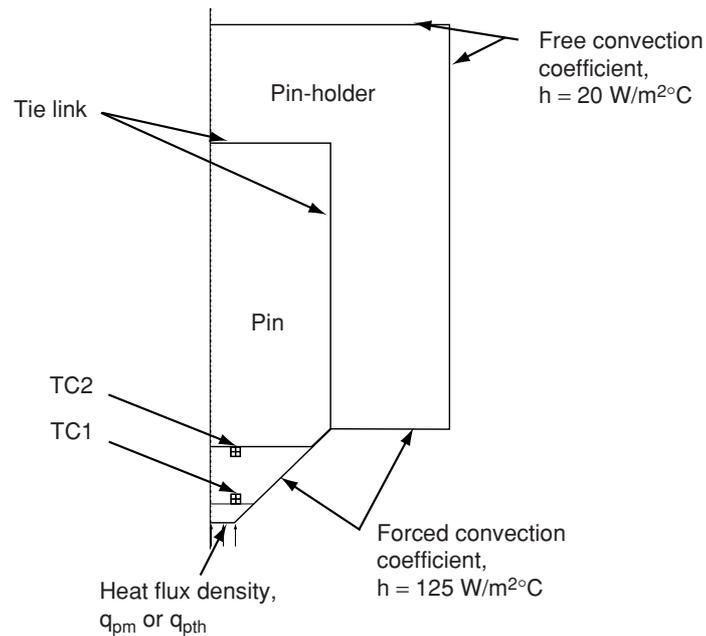


Figure 2 Axisymmetric configuration of the pin, of the pin-holder and boundary conditions (for a  $V = 600$  m/min sliding speed).

The normal loading is carried out by dead weights. The tangential (friction) force is measured using a strain gauge based sensor that is parallel to the friction plane. The dynamic temperatures (TC1 and TC2) of the pin and friction forces are recorded with a software developed on LABVIEW®. The friction coefficient is recorded during the tests too. Test conditions are listed in table 2.

Table 2 Test conditions

Sliding speed (m/min)	600
Normal load (N)	60

Before any test, the pin and the disc are cleaned using dry air and the pin is cleaned with ethanol in an ultrasonic tank.

### 3. NUMERICAL SIMULATION METHODS

The aim of the pin thermal simulation is to calculate the temperature at the pin/disc contact surface. This temperature is related to the power dissipated by friction in the contact. In this study, two approaches are considered to calculate the heat flux density flowing in the pin: (i) the first one is based on an heat partition coefficient ( $\beta$ ) and on the mechanical power dissipated in the contact by friction; (ii) the second one is based on the heat conduction by considering the Fourier law in an unidirectional case. In both cases, no thermal contact resistance is taken into account.

- **Approach 1:** Approach based on the mechanical power dissipated by friction.

In this approach, it is supposed that all the mechanical power dissipated in the contact is converted into heat. The generated heat is distributed between the pin and the disc. The calculation of the heat that flows in the pin is based on the determination of a heat partition coefficient ( $\beta$ ). This coefficient is related to experimental conditions. For high speed sliding tests, it depends on the dimensionless thermal number  $Pe$  (Peclet number). The total generated heat flux density ( $q_T$ ) and the heat flux density in the pin ( $q_{pm}$ ) per unit area per unit time are calculated using equations (1) and (2).

$$q_T(t) = \frac{\mu(t)PV}{\pi r^2} = \frac{F_T(t)V}{\pi r^2} \quad (1)$$

$$q_{pm}(t) = \beta \frac{F_T(t)V}{\pi r^2}, \quad (2)$$

$$\text{with } \beta = \frac{\lambda_p}{\lambda_p + \frac{\lambda_d \sqrt{\pi Pe}}{2}} \quad \text{and} \quad Pe = \frac{Vd}{\alpha_p},$$

Where  $\mu(t)$  is friction coefficient dependence of time and  $\lambda_p$  is the mean value of the pin conductivity.

- **Approach 2:** Approach based on a thermal unidirectional conduction case (temperature gradient).

During friction tests, the pin is continuously in contact with the disc. In the first periods of the test, the temperature evolution in the pin is highly transient. The accommodation phenomenon and the Biot number (which compares the resistances to the heat transfer in the bulk and on the surface of a body) can explain such a transient mode. However, for long test times, the temperature reaches a steady state. In this case, it is supposed that the conduction is the principal heat transfer mode. Based on this hypothesis, the heat flux density in the pin is calculated using the

Table 3 Thermal characteristics of the pin, of the pin-holder and of the disc

<b>Pin (WC-6wt%Co)</b>						
Temperature (°C)	20	100	200	300	500	600
$\lambda_p$ (W/m°C)	117	110	97	86	85	83
$C_p$ (J/Kg°C)	222					
$\rho_p$ (kg/m <sup>3</sup> )	14900					
$d_{pc}$ (m)	0.0035					
<b>Pin holder</b>						
$\lambda_{ph}$ (W/m°C)	13					
$C_{ph}$ (J/Kg°C)	500					
$\rho_{ph}$ (kg/m <sup>3</sup> )	7800					
<b>Disc</b>						
$\lambda_d$ (W/m°C)	54					
$C_d$ (J/Kg°C)	480					
$\rho_d$ (kg/m <sup>3</sup> )	7800					
$\lambda_{air}$ at 25°C (W/m°C)	0.025					

Fourier law for an unidirectional conduction case, considering equation (3).

$$q_{pth}(t) = \lambda_p \frac{\Delta T(t)}{dh} \quad (3)$$

Where  $\Delta T(t) = TC1(t) - TC2(t)$  is the difference between the temperature of *TC1* and *TC2* thermocouples,  $dh$  is the distance between these two thermocouples.

The calculated heat flux density is then applied at the pin/disc contact surface for numerical simulations (Fig. 2). Figure 2 presents the axisymmetric configuration of the pin with the applied boundary conditions. During friction, the disc rotation involves air movements. We considered a forced convection mode around the pin. Moreover, this forced convection mode is limited to the conical part of the pin and to the circular lower part of the pin-holder.

The value of the forced convection coefficient ( $h$ ) (eq. 4) is estimated using the Nusselt correlation number, ( $Nu$ ):

$$h = \frac{Nu * \lambda_{air}}{d_{pc}}, \quad (4)$$

where  $\lambda_{air}$  is air conductivity and  $d_{pc}$  is mean diameter of conical part of pin.

The considered thermal properties of the pin, of the pin-holder and of the disc are listed in Table 3. For the WC-Co pin material, the temperature dependence of its thermal conductivity is obtained by experimental measurements. The hot-disc method is carried out.

The commercial software ABAQUS® STANDARD 6.6.1 is used for numerical simulations. The transient option is considered to take into account the transient temperature phenomena. The pin and pin-holder are meshed with DCAX3

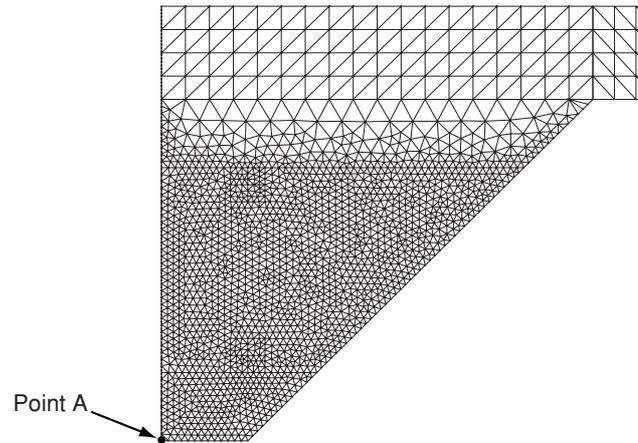


Figure 3 Meshing details in the conical part of the pin.

axisymmetric triangular finite elements. The complete meshing contains 9365 elements. To avoid the discretization size effect, finite elements are refined in the conical part of the pin (fig. 3).

## 4. RESULTS AND DISCUSSION

### 4.1. EXPERIMENTAL RESULTS

Figure 4 shows typical variations of the friction coefficient versus sliding duration under a load of 60 N at 600 m/min. It drops rapidly in a first step of testing. This phenomenon is due to the rupture of the contact surface asperities. Baek and Khonsari [13] studied the dependence of the friction coefficient with the surface roughness by considering stainless steel rubber coatings. They showed that the friction coefficient increases when increasing the surface roughness. It increases rapidly for low surface roughness levels ( $R_a < 0.12 \mu\text{m}$ ) and slowly for high surface roughness ones ( $R_a > 0.12 \mu\text{m}$ ). After this running-in stage, the friction coefficient decreases and reaches a steady state mode when the tribological contact is accommodated. Figure 5 shows the temperature evolutions measured by TC1 and TC2. The temperature curves show that the transient temperature spreads approximately between 0 and 200 s, before reaching a quasi steady state mode. The maximum temperature is close to 460°C for TC1 and to 420°C for TC2. Similarities are observed between the temperature evolutions and the friction coefficient ones. This is in agreement with the fact that surface heating is due to the energy dissipated by friction, mainly for the transient stage (Fig.4–5). The tribological behaviour of the contact interface influences the contact temperature and vice-versa.

### 4.2. NUMERICAL RESULTS.

#### 4.2.1. Approach 1: approach based on the mechanical power dissipated by friction.

In numerical simulations, the mechanical power dissipated in the contact is converted in a heat flux density that is applied at the contact pin surface. Figures 6 and 7 show the comparisons between the experimental temperatures recorded by the thermocouples and the computational temperatures extracted at the same depths in the pin simulations (Fig.2). At the test beginning (transient stage, between 0 and 200 s), it can be observed that the computational temperatures are lower than experimental ones and the deviation can reach 27% for TC1 and 17% for TC2. After this stage, in the quasi steady state mode, the computational temperatures show satisfactory agreement with experimental ones for the two

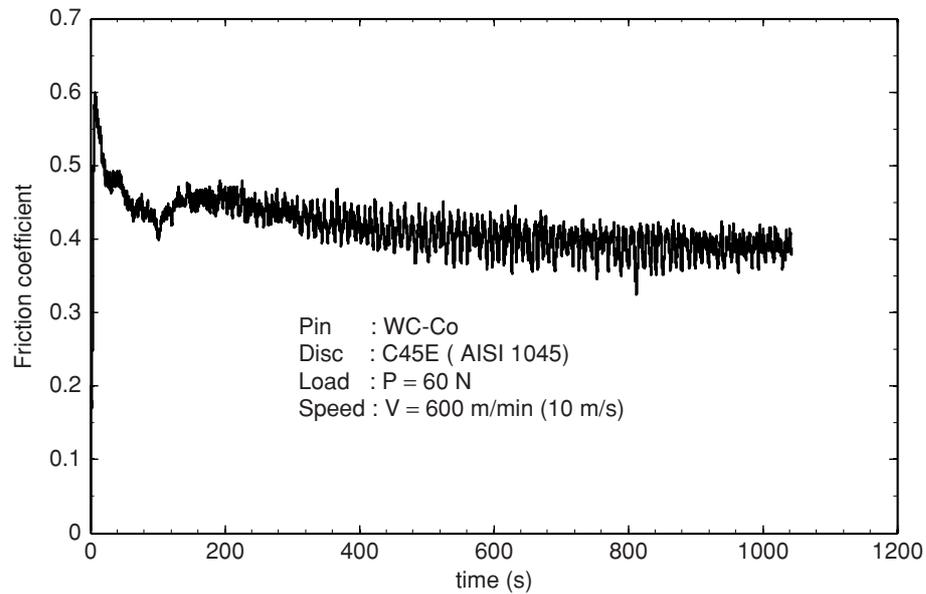


Figure 4 Friction coefficient versus sliding time during a tribological test performed at a 600 m/min sliding speed and under a 60 N normal loading.

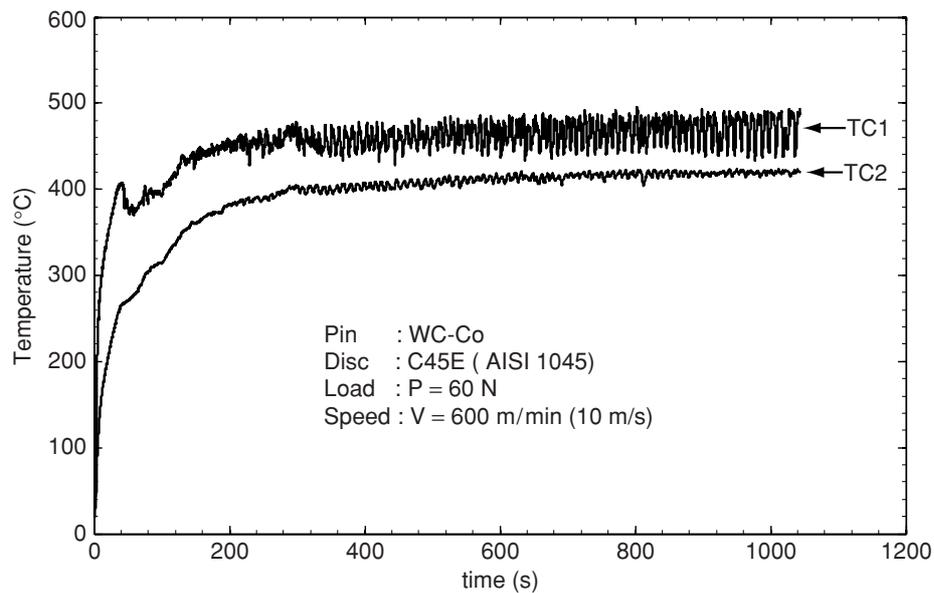


Figure 5 Temperature (measured by TC1 and TC2 thermocouples) versus sliding time during a tribological test performed at a 600 m/min sliding speed and under a 60 N normal loading.

thermocouples, since the maximum deviation is of 5% for TC1 and of 15% for TC2. It can be noticed that, in the quasi steady state domain, the computational temperatures become higher compared to experimental ones, especially for TC2. Komanduri and Hou [14] recently studied the heat dependence on the contact length, on the sliding speed and on the thermo-physical properties of the two elements of the sliding system. They suggested that the time

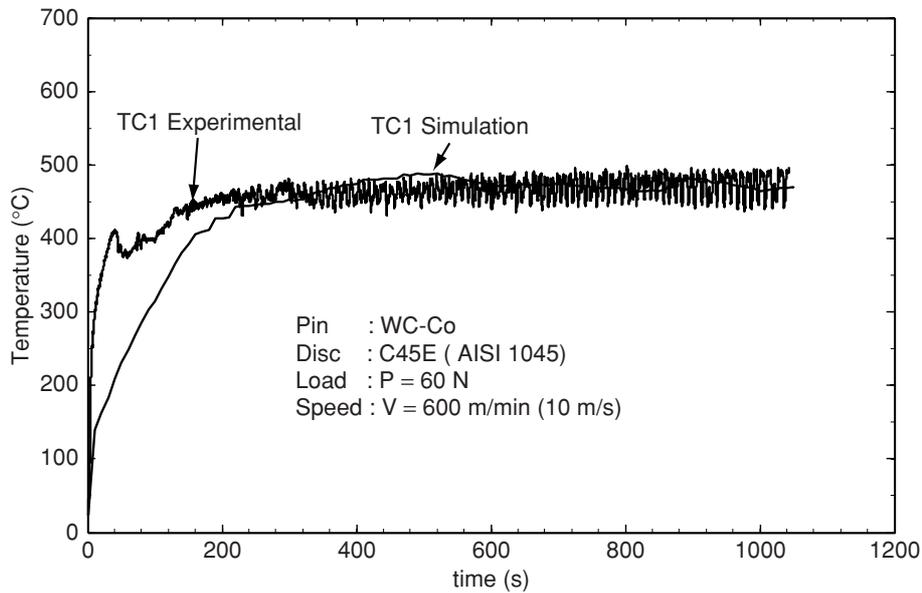


Figure 6 Comparison between the experimental and numerical (approach 1) temperature evolutions during a tribological test performed at a 600 m/min sliding speed and under a 60 N normal loading; case of the TC1 thermocouple.

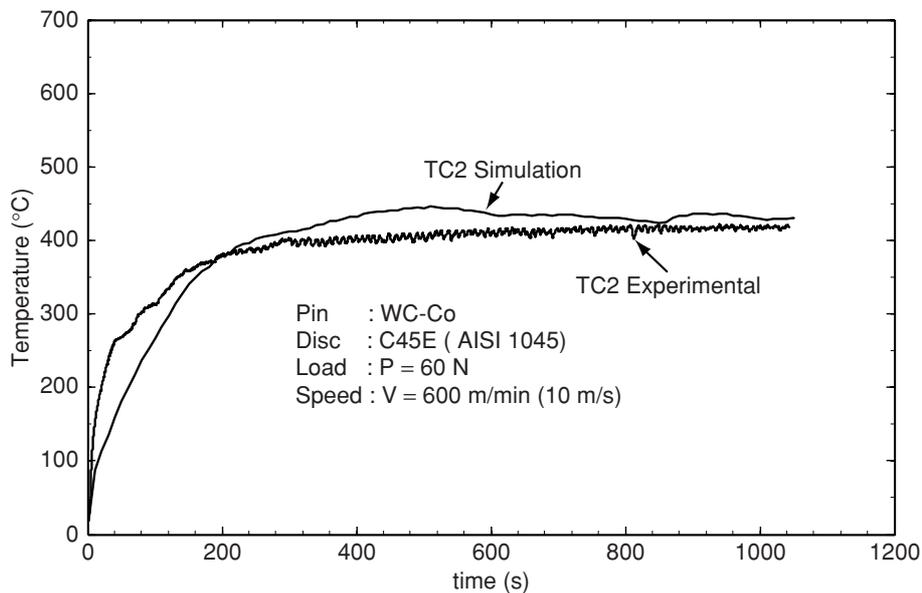


Figure 7 Comparison between the experimental and numerical (approach 1) temperature evolutions during a tribological test performed at a 600 m/min sliding speed and under a 60 N normal loading; case of the TC2 thermocouple.

stage is one of the parameters to be considered in the evaluation of the heat partition coefficient. They showed that the heat partition coefficient ( $\beta$ ) is not constant during the sliding test. This coefficient is higher in the transient stage and slightly constant in the quasi steady state stage. In our case, a constant value for the heat partition coefficient is considered.

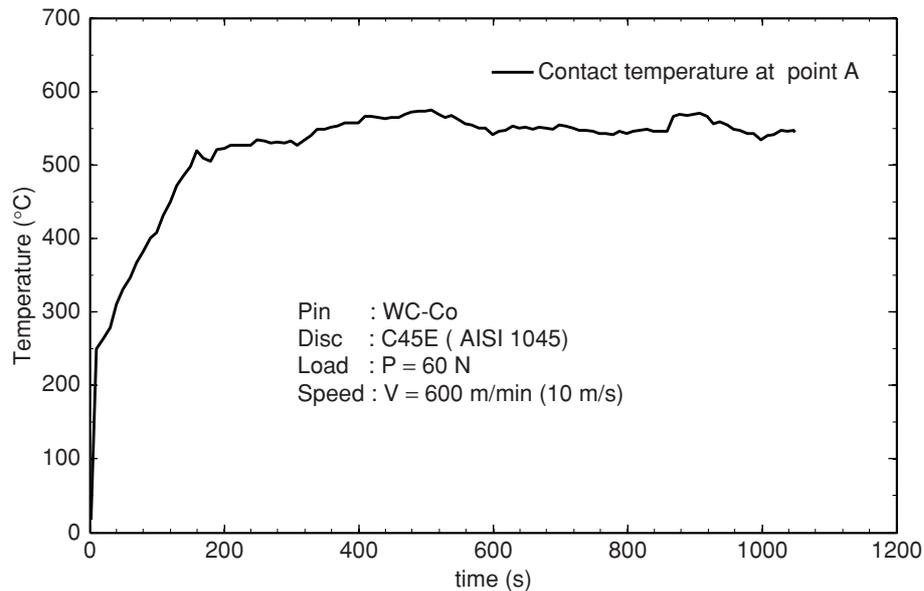


Figure 8 Evolution of the estimated contact temperature at point A (approach 1) in the case of tribological test performed at a 600 m/min sliding speed and under a 60 N normal loading.

It could explain part of the discrepancy between the measured temperatures and the calculated ones in the transient stage.

Figure 8 shows the evolution of the estimated contact temperature at the middle point of the contact surface (see point A on figure 3). As previously observed for TC1 and TC2 simulated temperatures, this evolution moves progressively from a transient domain to a quasi steady state one. In this last domain (200–1000s), temperature fluctuations are in the 523°C to 570°C temperature range. The temperature mean value is 550°C ± 11°C.

#### 4.2.2. Approach 2: approach based on a thermal unidirectional conduction case

In this case, the heat flux density determined from the two thermocouple measurements is applied at the contact pin surface. Figures 9 and 10 illustrate the comparisons between the experimental temperatures recorded by the TC1 and TC2 thermocouples and the computational temperatures. In this approach, an opposite phenomenon is observed compared to approach 1. In the transient stage, the computational temperatures are higher than those experimentally measured, especially for TC2. The deviation between experimental and computational temperatures can reach 12% for TC1 and 18% for TC2. After the transient stage, the computational temperatures decrease and move towards those measured in the quasi steady state domain. In this domain, the deviation is lower than 8% for TC1 and 10% for TC2. Both for the TC1 and TC2 numerical values, a temperature decrease is observed in the quasi steady state domain when the sliding time increases. This must be related to the decrease of the temperature gradient between TC1 and TC2 in this stage.

Figure 11 allows observing the evolution of the contact temperature (at the point A) estimated by approach 2. A transition from a transient domain to a quasi steady-state one is observed again. In the quasi steady state domain, temperature levels slowly decrease when the sliding time increase. As for TC1 and TC2 computational temperatures, this is related to the

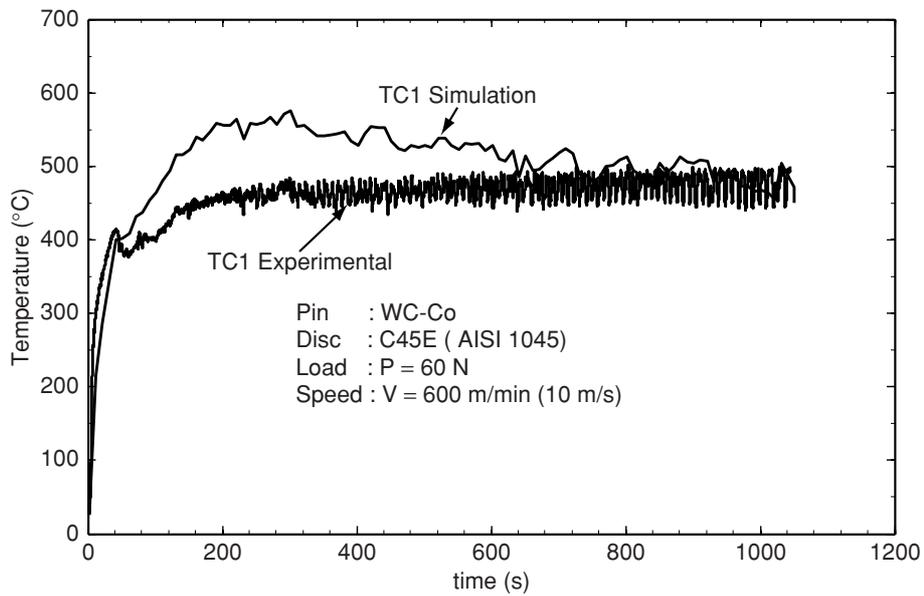


Figure 9 Comparison between the experimental and numerical (approach 2) temperature evolutions during a tribological test performed at a 600 m/min sliding speed and under a 60 N normal loading: case of the TC1 thermocouple.

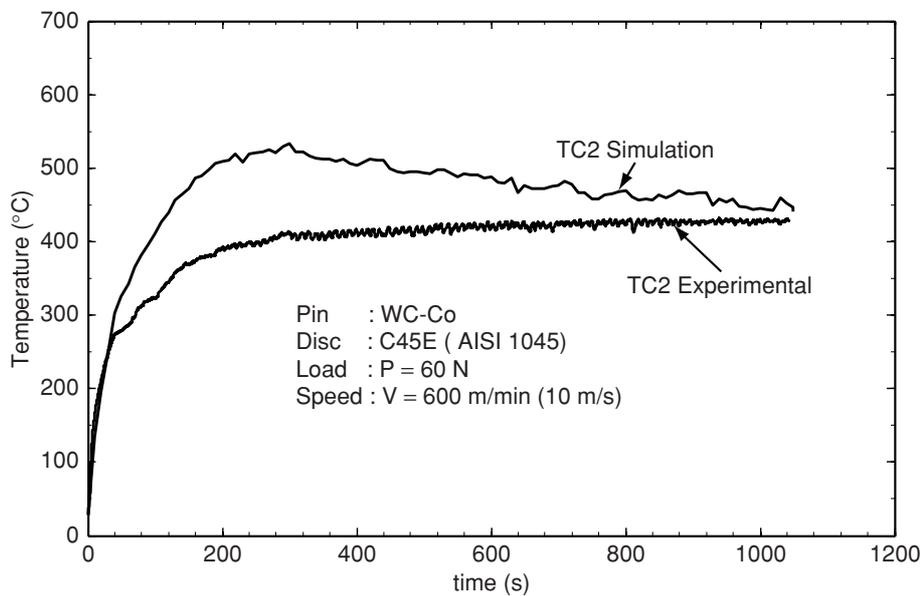


Figure 10 Comparison between the experimental and numerical (approach 2) temperature evolutions during a tribological test performed at a 600 m/min sliding speed and under a 60 N normal loading: case of the TC2 thermocouple.

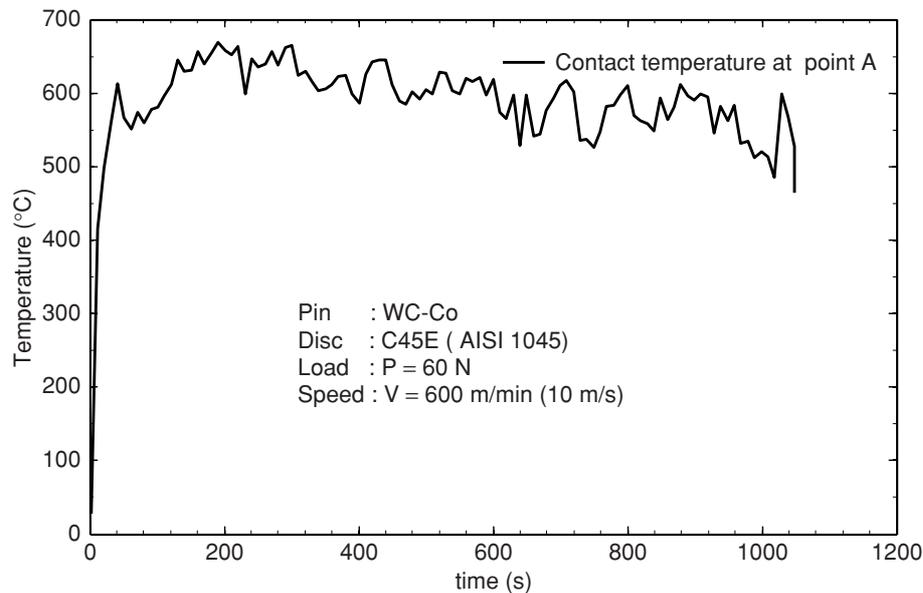


Figure 11 Evolutions of the estimated contact temperature at point A (approach 1 and approach 2) in the case of tribological test performed at a 600 m/min sliding speed and under a 60 N normal loading.

decrease of the temperature gradient. In this last domain (200–1000s), temperature fluctuations are in the 522°C to 640°C temperature range. The temperature mean value is  $590^{\circ}\text{C} \pm 32^{\circ}\text{C}$ .

The evolutions of the contact temperature obtained both by approach 1 and approach 2 can be compared. In the transient domain, the maximum deviation between the two estimated temperatures is equal to 25%. At the end of the quasi steady state domain, this deviation is reduced to 7%. This shows that the two approaches can be considered to estimate the contact temperature. Such temperature levels at the contact surface will influence physical and mechanical properties of materials. Therefore, the tribological behaviour will be affected too.

## 5. CONCLUSION

This paper presents two numerical methods to determine the contact temperature in tribology. These approaches are based on the determination of the heat flux density that must be applied at the contact surface using. In the first approach, the mechanical power dissipated in the contact and a heat partition coefficient are considered. In the second approach: the thermal gradient is calculated from the measurements of experimental temperatures. In both cases, the pin geometry and the boundary conditions are exactly similar.

When using the first approach, results show that:

- In a transient stage, computational temperatures are lower than measured ones.
- Nevertheless, computational temperatures are satisfactory in agreement with experimental ones since the maximum deviation is of 15%.
- In the quasi steady-state domain, temperature fluctuations are in the 523°C to 570°C temperature range.
- This approach can be improved by considering a time dependence of the heat partition coefficient: one for the transient stage and one for the quasi steady state domain.

For the second approach, results show that:

- In a transient stage, the computational temperatures are higher than the measured ones.
- At the end of quasi steady state domain, the computational temperatures are in agreement with experimental ones since the maximum deviation is of 10%.
- In the quasi steady-state domain, temperature fluctuations are in the 522°C to 640°C temperature range.

In the quasi steady state domain, contact temperature levels obtained by the two approaches are quite similar. The increase of the contact temperature in the pin is generated by friction at given normal load and sliding speed. Similarity of the results of the two approaches emphasises the thermal approach depends on the mechanical one. Thus, it is important to well estimate the partition coefficient and the heat boundary conditions following physical laws. Up to now, the height wear loss is not taken into account. Therefore, computational contact temperatures are underestimated. The absolute values of the estimated contact temperature levels show that physical and mechanical properties of materials could be affected, as the tribological behaviour.

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## NOMENCLATURE

$C_d$	Specific heat of the disc, J/(kg°C)
$C_p$	Specific heat of the pin, J/(kg°C)
$C_{ph}$	Specific heat of the pin holder, J/(kg°C)
$d$	Diameter of the contacting area, m
$d_{pc}$	Diameter of the conical part of pin, m
$dh$	Distance between the two thermocouples, m
$F_T(t)$	Tangential force (or friction force) dependence on time, N
$h$	Heat convection coefficient, W/(m <sup>2</sup> °C)
$P$	Load, N
$Nu$	Nusselt number
$Pe$	Peclet number
$q_T(t)$	Total heat flux density, W/m <sup>2</sup>
$q_{pm}(t)$	Heat flux density of the pin considering power dissipated, W/m <sup>2</sup>
$q_{pth}(t)$	Heat flux density of the pin considering temperature gradient, W/m <sup>2</sup>
$r$	Radius of the contact area, m
$TC1$	Temperature of the thermocouple located at 1 mm of the contact surface, °C
$TC2$	Temperature of the thermocouple located at 3 mm of the contact surface, °C
$V$	Sliding velocity, m/s
$\alpha_d$	Thermal diffusivity of the disc, m <sup>2</sup> /s
$\alpha_p$	Thermal diffusivity of the pin
$\alpha_{ph}$	Thermal diffusivity of the pin holder, m <sup>2</sup> /s
$\beta$	Heat partition fraction in the pin
$\lambda_d$	Thermal conductivity of the disc, W/(m°C)
$\lambda_p$	Thermal conductivity of the pin, W/(m°C)
$\lambda_{ph}$	Thermal conductivity of the pin holder, W/(m°C)
$\lambda_{air}$	Thermal conductivity of air at ambient condition, W/(m°C)
$\mu(t)$	Friction coefficient dependence on time
$\rho_d$	Density of the disc, kg/m <sup>3</sup>
$\rho_p$	Density of the pin, kg/m <sup>3</sup>
$\rho_{ph}$	Density of the pin holder, kg/m <sup>3</sup>
$\Delta T(t)$	(TC1-TC2)(t), (°C)