HEFAT2011
8th International Conference on Heat Transfer, Fluid Mechanics and Thermodynamics
11 – 13 July 2011
Pointe Aux Piments, Mauritius

CHARACTERIZATION OF HEAT FLUXES DURING WATER SPRAY COOLING OF THICK ROTATING PLATES

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ABSTRACT
Controlling temperature and cooling is, in many industrial applications, a key parameter for production quality and hardware lifetime. Although water spray cooling is widely used for processes involving high temperatures, it is still difficult to characterize the magnitude and distribution of heat fluxes. Water spray cooling of thick rotating plates is studied experimentally, taking thickness into account. A semi-inverse method is used to compute surface heat fluxes. Results obtained are presented in the form of boiling curves, also called Nukiyama curves, and validated by simulation.

NOMENCLATURE

λ	[W/mK]	Thermal conductivity
$\dot{Q}$	[W/m^3]	Volumetric heat generation density
q''	[W/m²]	Surface heat flux
CHF	[W/m²]	Critical Heat Flux
HTC	[W/m²K]	Heat Transfer Coefficient
T	[K]	Temperature
T_{sat}	[K]	Wall temperature
T_w	[K]	Water temperature
r	[m]	Cylindrical coordinates radius
z	[m]	Cylindrical coordinates elevation
e	[m]	Measurement cell thickness
d_{32}	[m]	Sauter Mean Diameter

INTRODUCTION
Water spray cooling of high temperature surfaces is widely used, especially for electronics [1][2], energy production [3] and in mechanical industries [4][5]. For the last one, temperature control of production equipment is a key parameter for product quality and tools lifetime. While allowing high surface heat fluxes, water spray cooling is made complex by a strong spatial heterogeneity. Critical Heat Fluxes (CHF) are influenced by parameters like surface shape, and droplet size and distribution.

In order to perform later predictive numerical simulations, the cooling of thick rotating plates under flat water spray has been tested experimentally, in industrial conditions. These plates are used to support high temperature products (from 600°C to 900°C) during their cooling cycle.

Water spray cooling

Numerous studies have aimed at characterizing the heat fluxes during water spray cooling. In 1934, Nukiyama conducted pool boiling experiments. The heat flux obtained were presented as a function of the temperature difference (T_{sat}-T_w) [6]. The Nukiyama boiling curve is useful to describe the boiling regimes. More recently, Jouhara and Axcell performed quenching experiments and identified different modes for the collapse of film boiling which arise. [3].

Water spray cooling was in particular studied since the late 60s, and a great number of correlations were proposed to predict heat transfer coefficient q'' [1,4,7]. Most of them involve parameters linked to the spray, such as Sauter mean diameter and droplets impact velocity. These parameters are difficult to obtain or measure in an industrial context. In order to tackle this problem, Estes and Mudawar proposed two correlations to evaluate d_{32} and q'' as functions of nozzle diameter and current operating parameters. [7]. Moreover, several research have investigated the heat flux sensitivity to several geometrical parameters. Silk and al. showed that it is possible to significantly increase the CHF by spraying over fin structures, whose dimensions are much larger than the vapour film thickness [8]. Moreira and Panão found evidence that intermittent spray allows a better efficiency and proposed a new relationship including spraying time and frequency [9]. Wendelstorf and al. took the wall oxydation into account and evaluated the loss of efficiency as the oxydation layer thickness increase [10]. This result agrees with the conclusions of Sehmbey and al., which indicate that a polished surface dramatically increases the heat transfer. [11].

This oxydation can rapidly appear in industrial operating conditions, and cannot be evaluated in time with any precision. As a consequence, these correlations should not be used to predict the heat transfer during the tools lifecycle but can be considered with interest for validation purposes.

Most of these studies were performed in the laboratory, with test rigs designed to lower measurement errors. Samples are mostly heated by electrical cartridges [1,8,9,12,13], furnaces [4,10,14] or in some cases by induction [15]. Sample temperatures are measured in major cases by using thermocouples [1,4,8,12,13], although recent studies used IR thermography [2]. Heat fluxes are computed from temperature histories even by direct methods, assuming 1D heat conduction for thin samples [4,7,9,12], or by semi-inverse or inverse methods, determining with simulations the parameter values lowering the error between real and numerical results [5,15].

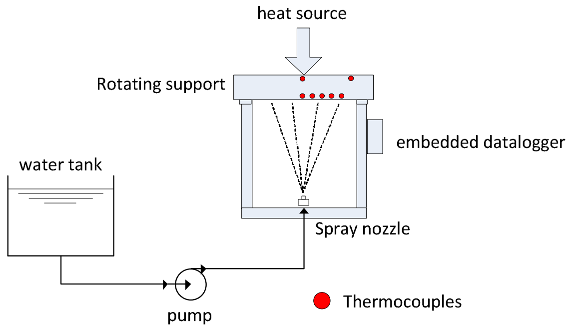


Figure 1 Experimental apparatus

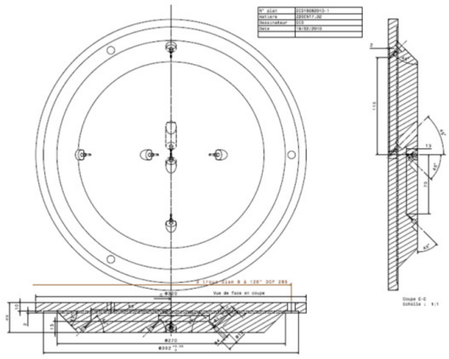


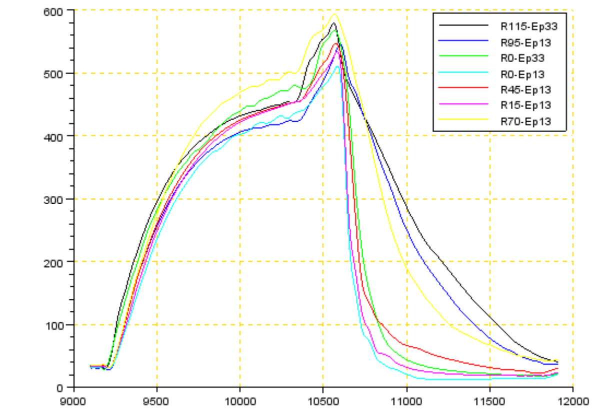
Figure 1.b Thermocouples disposition

EXPERIMENTAL PROCEDURE

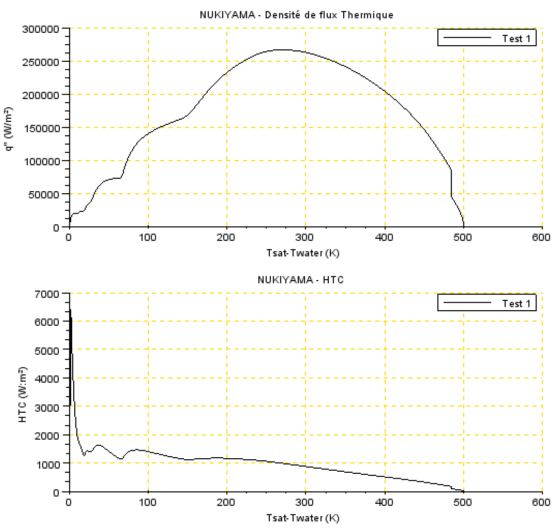
Description of the apparatus

The apparatus aims at determining the thermal behaviour of the rotating support plates in operating conditions. As the electromagnetic, thermal and humidity conditions cannot be controlled, the hardware was adapted to this environment. The samples are screwed onto a rotating support trough a insulator, and driven by an electrical motor. The nozzle is supplied with water by a pneumatic pump, allowing an accurate pressure and flow rate control. The sample plates are drilled and K type thermocouples are placed on five different radii and two thicknesses (Fig.1&1.b). The thermocouples are placed with a 45° angle, which corresponds to the isotherms observed in preliminary simulations. A temperature data logger has been embedded to the rotating part, allowing the recording of cooling cycles without using wireless transmissions, which would be disrupted by environmental conditions. A heating and cooling cycle takes about 20 minutes, and temperatures are recorded every 0,2s.

Experimental data were recorded, varying the surface shape (flat surface or fins), water supply pressure and the flat spray cone angle (from 25° to 90°). After each test, temperature histories, including one or many heating/cooling cycles, are extracted from the data logger, processed. The results are presented as a form of Nukiyama boiling curves. (Fig.2)



(a)



(b)

Figure 2 Temperature history for all sensors (a) and an example of boiling curves (b) On top : surface heat flux q". On bottom: Heat Transfer Coefficient HTC

Data processing

Heat transfer in the solid phase are described by the heat equation :

$$\lambda \nabla^2 T + \dot{Q} = \rho C_{p,s} \frac{\partial T}{\partial t} \quad (1)$$

It is not possible to simplify $\nabla^2 T$ for this case. Indeed, the thickness of the plate and the spatial heat fluxes heterogeneity prevent horizontal isothermal layers. As a consequence 1D heat conduction cannot be assumed.

Let us consider a measurement cell centered on a thermocouple. This sensor being placed near the surface, we can identify the boundary heat flux extracted by the spray as the source/sink term $\dot{Q}$ in eq. (1)

Every time step, a temperature contour is computed by bilinear interpolation within the domain limited by the sensor array (Fig. 3).

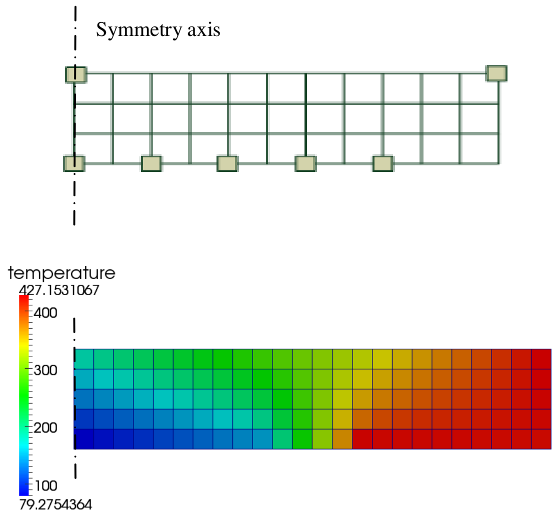


Figure 3 Domain discretization used for interpolation and resulting temperature map. Please note that only the right side of the plate is represented here.

The laplacian $\nabla^2 T$ is estimated by the means of finite differences method around the measure point, in cylindrical coordinates:

$$\nabla^2 T = \frac{1}{r}\frac{1}{\Delta r}\left[(r+\Delta r)\frac{1}{\Delta r}\left[T(r+\Delta r,z) - T(r,z)\right] - (r-\Delta r)\frac{1}{\Delta r}\left[T(r,z) - T(r-\Delta r,z)\right]\right] + \frac{T(r,z-\Delta z) - 2T(r,z) + T(r,z+\Delta z)}{\Delta z^2} \quad (2)$$

Surface heat flux q" can be determined by adding the cell thickness, which depends on the distance between the thermocouple and the wall surface :

$$q'' = e \times \dot{Q} \quad (3)$$

$$HTC = \frac{e \times \dot{Q}}{T_{sat} - T_w} \quad (4)$$

RESULTS AND VALIDATION

In order to validate this method, the computed boiling curves were added to a finite volume model. Numerical results are compared to experimental temperature data (Fig. 4).

For each test, the Root Mean Square Error (RMSE) is calculated to evaluate the accuracy. We observed a good agreement between numerical results and real tests for the sensors placed near the center of the spray, with RMSE less than 20K, which represents less than 5% relative error. Meanwhile, the error increase for sensors placed at the disk extremity , with RMSE up to 100K. Many reason can explain this error increase. The principle hypothesis is the bad evaluation of lateral convection. Initially based on air characteristics, it seems that the vapour film still has some influence to the heat transfer, even if the water spray is far from there. Another reason should be the low number of sensors placed in this area and the resulting interpolation errors, due to limitations in the data logger capacity. Priority was given to the discretization of the wetted area, which is our main interest here. These errors should not affect our confidence in the boiling curves obtained.

CONCLUSIONS

An experimental investigation of the water spray cooling of thick plates is being performed. A semi-inverse method has been proposed to recover the Nukiyama boiling curves from recorded temperature histories. The resulting data has been tested and validated with simulations. The feasibility of such a procedure in an industrial context has been proven, and will be extended in order to collect a knowledge base describing the tools behavior all along their lifecycles.

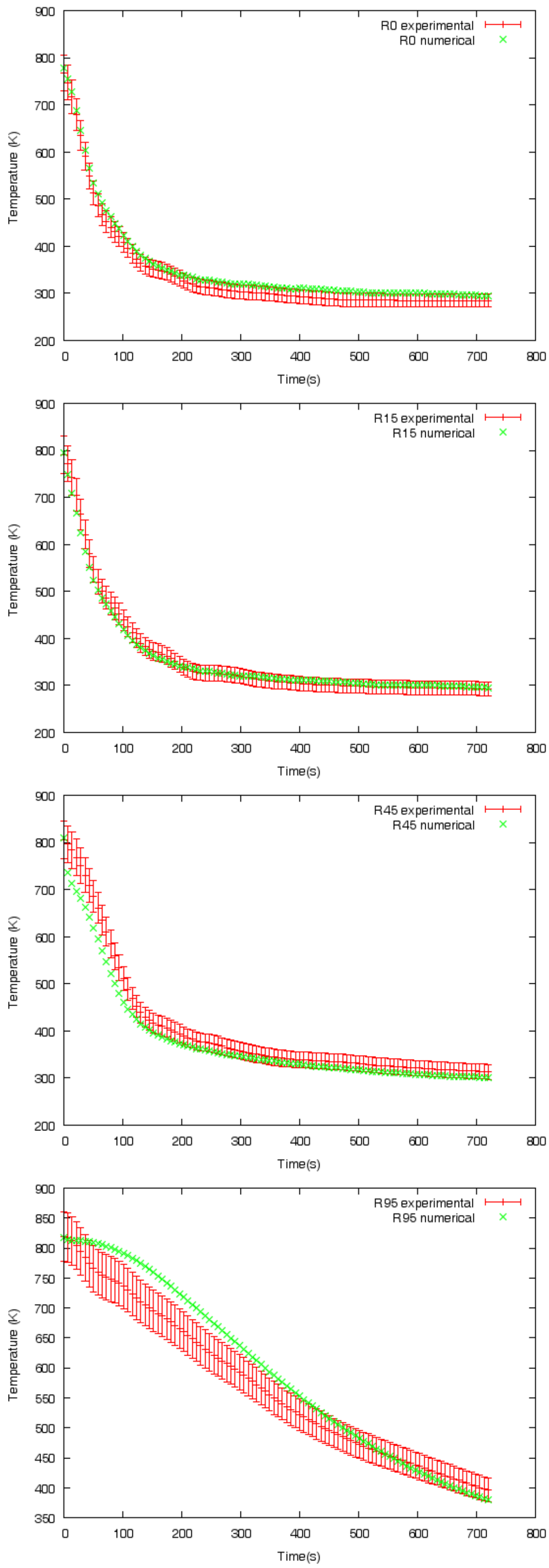


Figure 4 Comparison between experimental results (with 5% error bars) and corresponding simulations at the center of the plate surface and 15mm,45mm,95mm radius

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