

## INVERSE DETERMINATION OF TRANSIENT LOCAL HEAT TRANSFER IN A LIQUID MICROLAYER.

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### Abstract :

*It is difficult to measure the temperatures distribution in a liquid micro layer. That's why in this article the inverse estimation technique is applied to determine the local surface heat flux in transient state. The results of the local thermal performances analysis during cooling by a miniature jet impinging a horizontal surface are presented.*

### Résumé :

*La détermination de la répartition locale de la température de surface dans une micro-couche liquide est très complexe. Cet article présente les résultats d'analyse inverse des mesures sur le refroidissement d'un disque chauffé par un minijet d'eau réparti sur la surface d'échange sous la forme d'une micro-couche liquide. La distribution locale des densités de flux surfaciques est obtenue par résolution du problème inverse de conduction de la chaleur en transitoire.*

### Key-words:

**inverse problems; transient state; local heat transfer coefficient**

### 1 Introduction

Several experimental and theoretical studies on the heat transfer and the hydrodynamics of liquid jet impingement have been reported in the literature [1-2]. Numerous studies were conducted to evaluate the average heat transfer coefficients, but the characterization of the local heat transfer at the heat exchange surface for unsteady state has not obtained much attention. The heat transfer for jet impingement is influenced by different physical parameters such as : the velocity, the water jet temperature and the heat exchange surface, the jet orientation , etc.

Extensive research has been conducted to understand the heat transfer characteristics in the stagnation zone at the steady state because of the stagnating mass impacts at high speeds on the impingement surface. When a circular liquid free jet strikes a flat plate, it spreads radially in a very thin micro film along the heated surface. A hydraulic jump, that is associated with a Rayleigh-Taylor instability, appears [3]. The local distribution of the heat transfer coefficient is a function of the local liquid film thickness and the flow velocity.

It is known that the local heat transfer at the heat exchange surface depends on the axial profile of the temperature in the thin liquid film (Fourier law). Measurement of the temperature distribution in the liquid film along the axial and the radial directions without perturbing the flow is very complex using the micro sensors or the induced fluorescence laser. For this reason, inverse heat conduction problem (IHCP) has been solved and used in order to determine locally the transient distribution of the thermal boundary conditions at the wetted surface using only the temperature measurements inside the disk.

In the present work, the local heat transfer for unsteady-state was studied by solving the IHCP and using the microsensors' responses placed inside the disk. The iterative regularization

method was used to solve the inverse problem under analysis [4]. The method was based on the conjugate gradient method used to minimize the residual functional and the residual discrepancy principal as the regularizing stopping criterion.

## 2 EXPERIMENTAL SETUP

The experimental setup is described in detail in reference [3]. Fig. 1 shows the heater assembly where the experimental cylinder (1) was made of a brass block and it was heated by using one cartridge heater (2) of 200 W that has a 10 mm diameter and 40 mm length. The electric power input was measured using an ammeter and a voltmeter (uncertainty 2%). The heating cylinder was thermally insulated with Teflon (3) on all faces except the cooling face in order to prevent the heat loss. The thickness of the Teflon was 30 mm and its thermal conductivity is 0.23 W/mK. The heat losses were estimated to be lower than 7%. The experimental disk was 50 mm in diameter and 10 mm thick. The heat exchange surface was polished and oriented horizontally. The temperatures inside the experimental cylinder were measured using 50 Chromel-Alumel micro-thermocouples of 100  $\mu\text{m}$  diameter (uncertainty of  $\pm 0.2^\circ\text{C}$ ). The thermocouples were placed at axial and radial locations below the cooling surface of 0.6, and 8 mm as shown in Fig. 2. Twenty five thermocouples were placed at 0.6 mm below the wetted surface at radial intervals of 3.5 mm. Twenty five thermocouples were located at 8 mm below the wetted surface at radial intervals of 3.5 mm. The thermocouples were located at different concentric circles of radii 8 mm, 11 mm, 15 mm, 19 mm and 22 mm.

For each experiment, the flow rate was first adjusted with the regulating valves. Initially, the disk was covered in order to prevent the liquid to wet the heat exchange surface. The hydrodynamic boundary layer is fully established and steady. For a fixed total heat flux, the disk was heated continually and the wall temperatures were monitored. When the thermal steady state was reached, the heat exchange surface was rapidly cooled with the impingement liquid jet (Fig. 3). The time-dependent local wall temperatures were recorded, until the experimental disk reached a new steady state (Fig. 4). A Labview data acquisition system (11) was used to collect the temperature data for each operating condition. The local surface temperature and heat flux were determined by solving IHCP using these measurements. The tests were conducted for a circular jet falling by gravity on the heated disk with the nozzle diameter of 2.2 mm or 4 mm.

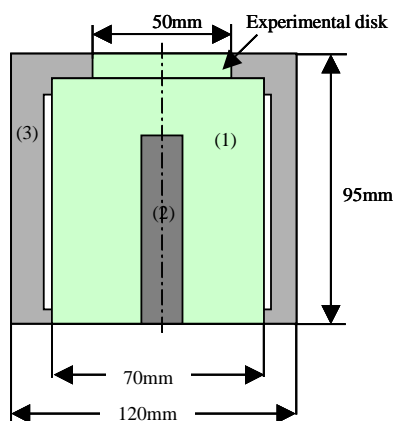


FIG. 1 – Experimental apparatus

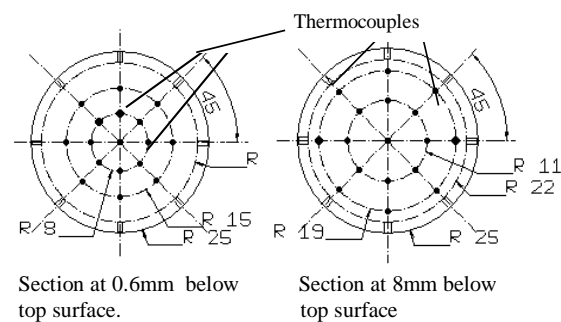


FIG. 2 – Thermocouple locations

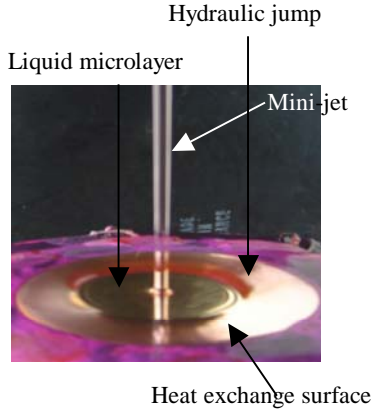


FIG. 3 – The impingement liquid jet.

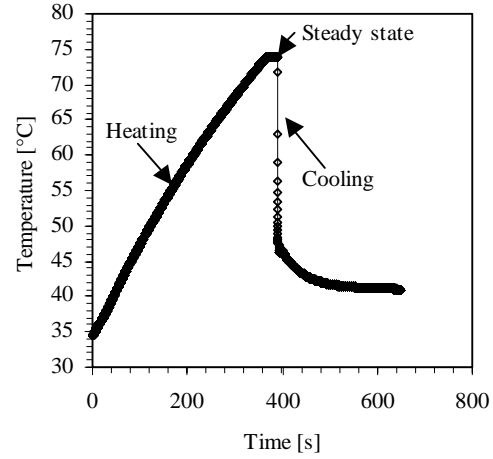


FIG. 4 – Wall temperature measured at the stagnation zone.

### 3 PHYSICAL MODEL

The physical model considers a disk of thickness  $E$  and radius  $R$ . The disk was thermally insulated with Teflon on all faces except the wetted and the heated faces. The physical model of an unsteady heat conduction process is given by the following system of equations:

$$\frac{\rho C_p}{\lambda} \frac{\partial T(r,z,t)}{\partial t} = \frac{\partial^2 T(r,z,t)}{\partial r^2} + \frac{1}{r} \frac{\partial T(r,z,t)}{\partial r} + \frac{\partial^2 T(r,z,t)}{\partial z^2}, \quad 0 < r < R, \quad 0 < z < E, \quad (1)$$

$$\frac{\partial T}{\partial r}(0,z,t) = 0, \quad \text{where } 0 < t \leq t_f, \quad 0 \leq z \leq E \quad (2)$$

$$\frac{\partial T}{\partial r}(R,z,t) = 0, \quad \text{where } 0 < t \leq t_f, \quad 0 \leq z \leq E \quad (3)$$

$$T(r,z,0) = T_0, \quad \text{where } : 0 \leq r \leq R, \quad 0 \leq z \leq E \quad (4)$$

$$\lambda \frac{\partial T}{\partial z}(r,E,t) = Q_w(r,E,t), \quad \text{where } : 0 < t \leq t_f, \quad 0 \leq r \leq R \quad (5)$$

$$T(r,0,t) = f(r,t), \quad \text{where } : 0 < t \leq t_f, \quad 0 \leq r \leq R \quad (6)$$

The distribution of the local heat flux  $Q_w(r,E,t)$  at the heat exchange surface ( $z=E$ ) is unknown. It was estimated by solving the IHCP using the temperatures  $T_{\text{meas}}(r_n, z_n, t)$  measured at nodes  $(r_n, z_n)$  inside the disk. The solution of the inverse problem is based on the minimization of the residual functional defined as :

$$J(Q_w) = \frac{1}{2} \sum_{n=1}^{N_{\text{meas}}} \int_0^{t_f} [T(r_n, z_n, t; Q_w) - T_{\text{meas}}(r_n, z_n, t)]^2 dt \quad (7)$$

where  $T(r_n, z_n; Q_w)$  are the temperatures at the sensor locations computed from the direct problem (1)-(6). The minimization was carried out by using the conjugate gradient algorithm [4]. The heat flux  $Q_w(r,E,t)$  was approximated in the form of a cubic B-spline and the IHCP was reduced to the estimation of a vector of B-Spline parameters. The conjugate gradient procedure is iterative. For each iteration (it), the successive improvements of desired parameters were built. The descent parameter was computed using a linear approximation as follows:

$$\gamma^{it} = \frac{\int_0^{t_f} \sum_{n=1}^{N_{meas}} [T^{it}(r_n, z_n, t; Q_w) - T_{meas}(r_n, z_n, t)] \theta^{it}(r_n, z_n, t; \delta Q_w) dt}{\int_0^{t_f} \sum_{n=1}^{N_{meas}} \theta^{it}(r_n, z_n, t; \delta Q_w)^2 dt} \quad (8)$$

The variation of temperature at the sensor locations  $\theta^{it}(r_n, z_n, t; \delta Q_w)$  resulting from the variation of heat flux  $\delta Q_w(r, E, t)$  was determined by solving the variational problem. The variation of functional  $J(Q_w)$  resulting from temperature variation is given by :

$$\delta J(Q_w, \delta Q_w) = \sum_{n=1}^{N_{meas}} \int_{t_0}^{t_f} [T(r_n, z_n, t, q_w) - T_{meas}(r_n, z_n, t)] \theta(r_n, z_n, t) dt \quad (9)$$

where  $\theta^{it}(r_n, z_n, t; \delta Q_w)$  was determined at the sensor locations  $(r_n, z_n)$  by solving the variational problem.

Using the Lagrange multiplier method, the necessary condition of the optimization problem was obtained from the following equation:

$$\delta L(Q_w, \delta Q_w) = 0 \quad (16)$$

where  $\delta L(Q_w, \delta Q_w)$  is the variation of the Lagrangian functional. Equation (16) requires that all coefficients of the temperature variation  $\theta(r, z, t)$  were equal to 0. To satisfy this condition the necessary conditions of optimization were defined in the form of adjoint problem.

If the direct problem and the adjoint problem were verified, the variation of Lagrangian functional becomes:

$$\delta L(Q_w, \delta Q_w) = - \int_0^{t_f} \int_0^R \psi(r, E, t) \delta Q_w(r, E, t) dr dt \quad (23)$$

The vector gradient can be verified by the following equation [5] :

$$J'_{Q_w}(r, E, t) = -\psi(r, E, t) \quad (24)$$

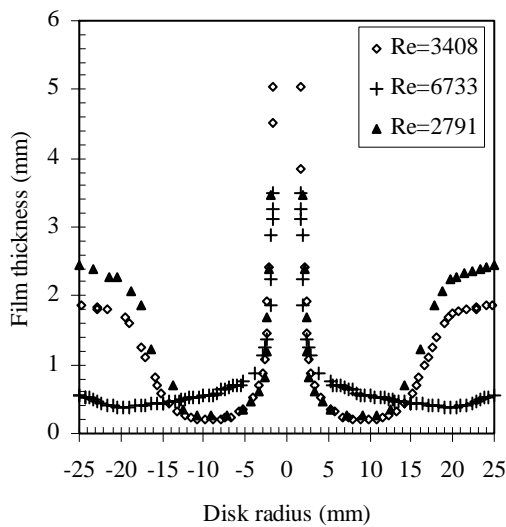
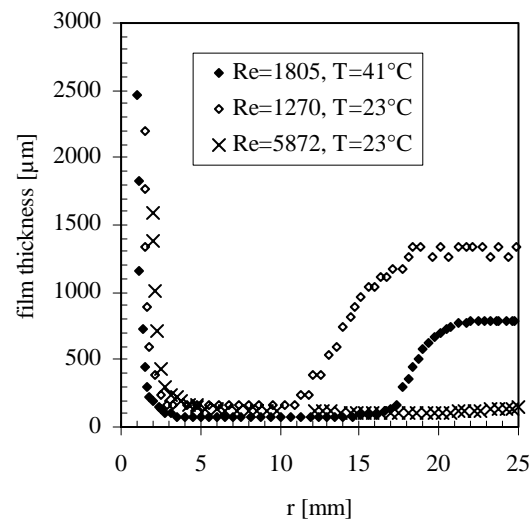
The details of the calculation procedure are given by Louahlia-Gualous et al. [4] and will not be repeated here.

## 4 Results and discussions

The numerical procedure is verified by using the temperature data calculated from the direct problem for a known circumferential heat flux density to simulate the measured temperatures inside the cylinder for the inverse problem [3]. The number of parameters is equal to 8.

### 4.1. Local evolution of the liquid film thickness

The measurements of the local liquid microlayer depth near the heated disk and the velocity profile are along the radial direction. Fig. 5 presents a sample example of the local liquid layer depth ( $\delta$ ) measured for three values of the inlet Reynolds number ( $Re=6733$ ,  $Re=3408$ , and  $Re=2791$ ) based on the diameter of the nozzle that used equal to 2.2 mm for these experiments. The inlet temperature of the water is of 32°C and the nozzle-heat exchange surface spacing of 95 mm. The heat flux inside the cylinder is of 50 W. Fig. 5 shows the presence of the three distinct zones : the impingement zone, the zone where the liquid layer depth is approximately uniform, and the final zone where a hydraulic jump is formed. The radius at which the liquid layer depth increases, is termed as the hydraulic jump radius that depends on the inlet Reynolds number as shown in Figs. 5 and 6.

FIG. 5 – Local film thickness,  $d=2.2$  mmFIG. 6 – Local film thickness for  $d=3.3$  mm.

#### 4.2. Inverse estimation of the heat transfer coefficient in the thin liquid film

The local heat transfer was treated in the steady and unsteady states. For inlet Reynolds number of 7600, Fig. 7 shows an example of the temporal temperature measured for different radial locations at 0.6 mm below the heat exchange surface. During experiments, the heat flux imposed inside the experimental cylinder was 45 W, the nozzle-heat exchange surface spacing was 30 mm, and the inlet temperature of the liquid was 42°C ( $\pm 0.2$ °C). At the steady state, the wall temperatures were 78°C. When the heat exchange surface was wetted, the wall temperatures decreased continually and reached a stable value during a short period. It is shown that the temperature at the stagnation zone was lower than the temperature in the zones far from the impingement zone.

The IHCP was solved using the measured temperatures at 0.6 mm from the heat exchange surface. The local thermal characteristics were estimated using the temperatures measured at the bottom surface ( $z=0$ ) as the boundary condition to solve the direct problem. Fig. 8 shows the unsteady evolution of the predicted surface heat flux at different radial locations on the cooling surface ( $z = E = 8$  mm). The surface temperature was low in the stagnation zone and in the impingement zone where the heat flux is high. The difference between the wall and the liquid temperatures is high at the moment when the impinging jet impinged the heat exchange surface and it decreases with time. Therefore, for each radial location the heat flux decreases with time and follows the same trend. The heat flux decreased after the impingement zone because the liquid spreads along the radial direction as a very thin film.

For both sides of the disk, the radial distributions of the surface heat flux presented in Figs. 8 for different times, are not uniform along the radial direction, and they are high in the impingement zone. After this zone, the heat transfer decreased because the liquid covered the entire heat exchange surface. Therefore, the local liquid flow rate decreased in spite of the decrease of the film thickness. When the radius  $r$  became higher than approximately 0.018 mm, the heat transfer deteriorates because of the formation of the hydraulic jump where the velocity of the flow became relatively negligible. For each time, the local heat flux and the local heat transfer coefficient follow the same trend. Beyond 64s, the curves of the heat flux and those of heat transfer coefficient were the same with time because of the steady state.

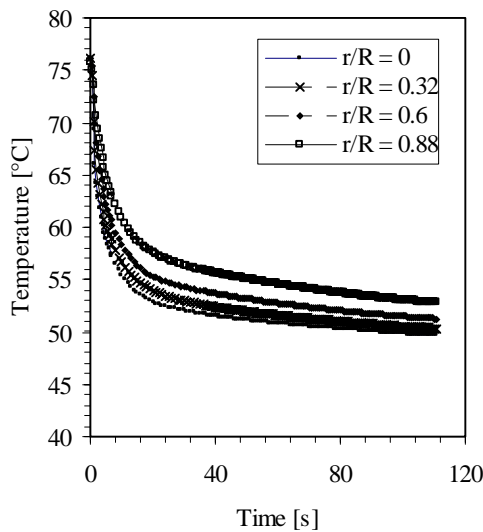


FIG. 7 – Measured temperatures in disk.

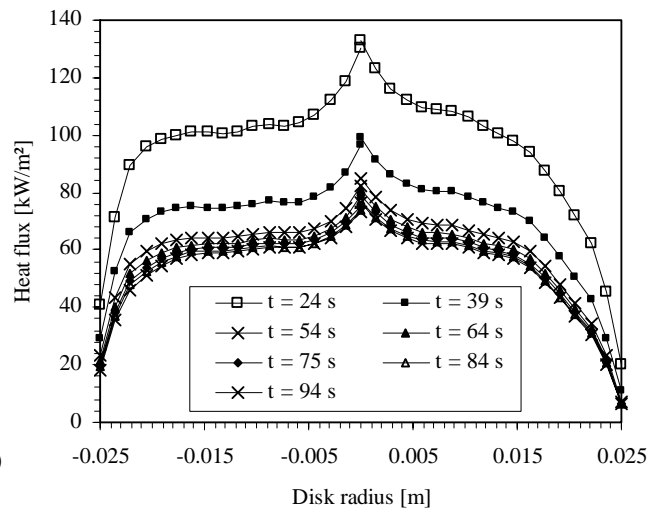


FIG. 8 – Surface heat flux estimated by IHCP.

## 5 Conclusions

The unsteady study of the local heat transfer was conducted for different experimental conditions. For each radial location, the heat flux decreases with time and follows the same trend. The heat flux was high in the impingement zone and it decreases after this zone because the liquid spreads along the radial direction as a very thin film. For each time, the surface temperature was lower in the stagnation zone and the maximum heat transfer occurs in the stagnation zone and falls off with the radial location because the local flow rate decreases and the thermal boundary layer increases with radial distance.

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