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Comparative study of the cooling of a hot temperature surface using sprays and liquid jets

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ABSTRACT

This experimental work aims at investigating the cooling of hot surfaces by using full cone sprays; comparison with the use of a liquid jet is also considered. The wall is a 175 mm diameter nickel disk and 5 mm thickness heated by electromagnetic induction up to about 800 °C. In the case of the sprays, the goal is to link the spray properties with the heat flux removed from the heated surface. For the spray, the influence of the mass flux distribution as well as the droplets properties on the cooling are studied by using three different spray nozzles. The mass flux distribution for each of the sprays is measured with the help of a patternator. The heat removed during the cooling phase is investigated with the use of infrared thermography while the droplet properties are characterized simultaneously by a Phase Doppler system. The Phase Doppler technique is mainly applied to study the statistical properties of the outcoming droplets in the vicinity of the heated surface. A decrease of the outcoming droplets size compared to the incoming one is well observed. Moreover, Phase Doppler device has also highlighted that the presence of the surface may have a significant influence on the upstream spray flow. Compared to the liquid jet, heat flux measurements have clearly demonstrated that the sprays lead a more spatial uniformity of the extracted heat flux and a better cooling efficiency. When spray cooling is considered, the influence of the mass flux on the heat flux and the cooling efficiency is in qualitative agreement with previous studies performed in similar conditions. In addition, a comparative study of performances, in term of liquid consumption and cooling duration, is performed by considering a similar surface temperature decrease for the three sprays and the liquid jet. Among the four experiments, the liquid jet has the longest cooling time as well as the highest liquid consumption. Furthermore, the best performances are reached for the spray having the highest mass flux value.

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1. Introduction

The use of water for cooling of hot surfaces in industry is unavoidable when high heat flux dissipation is required. Some emblematic examples are related to nuclear safety issues or to the thermal processing of alloys in steel industry. Cooling processes include pool boiling, impingement of liquid jets or sprays. In steel industry, pool boiling and liquid jets are widely used because they provide high dissipation rates but they have generally failed to ensure uniform cooling. Thus the homogeneity of the metallurgical phase can suffer from defaults.

The use of sprays seems attractive for several reasons: a better spatial uniformity of the cooling can be achieved associated with

lower water consumption for the same efficiency in term of heat transfer. Furthermore, lower water consumption involves less energy requirements for pumping and recycling the cooling water and thus an increase of the overall energy efficiency of the process. The experimental studies devoted to water spray cooling cover a wide range of cooling conditions. For practical reasons, the case of deposited droplets [1,2] single or multiple droplets was widely studied and well documented [3–10]. When sprays were used, several configurations were found, such as mono-sized sprays (impulse-jet technique to produce uniform size droplets) [11–15] or sprays polydispersed in size [16–26]. Furthermore, several categories of nozzles are available to generate various sprays with different geometries and liquid densities, droplet size and velocity distributions: full cone, hollow cone, flat fan or air-mist nozzles. However, only few authors have performed quantitative comparisons between the different spraying nozzles [4,21,26].

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Nomenclature

We	We number (–)
c_p	specific capacity (J/kg K)
d	droplet diameter (m)
h	enthalpy (J/kg)
$\dot{m}$	liquid mass flux (kg/m ² s)
$\dot{M}$	flow rate (l/mn)
q''	heat flux (W/m ²)
u, v, w	droplet velocity component (m/s)
p	liquid pressure (bar)
r	disk radius (m)
T	temperature (K)

Greek symbols

α	thermal diffusivity (m ² /s)
β	spray angle (°)
ε	emissivity (–)
λ	laser wavelength (m)

ρ	mass density (kg/m ³)
μ	dynamic viscosity (Pa s)
σ	surface tension (kg/m ²)
ξ	cooling efficiency (–)

Subscripts

b	laser beam
d	droplet
l	property of liquid phase
$Leid$	temperature at which minimum film boiling heat flux occurs
s	spray
sat	liquid saturation properties
sub	surface subcooling
v	property of vapor phase
w	conditions at front disk wall

In steel industry, the surface temperatures are generally above the Leidenfrost temperature at the beginning of the cooling phase and high water mass flux are required, which involved generally large-sized droplets. Experiments referred in the literature aim at characterizing the heat flux removed from the hot wall in conjunction with the droplets characteristics. Thermocouples inserted in the wall, coupled with an inverse conduction algorithm, are generally used to characterize the wall heat flux [5,6,17–19,26] whereas a granulometer (e.g. Phase Doppler analyzer system) is used to derive the velocity and diameter distributions of the droplets [18,19]. For instance, Al-Ahmadi et al. [26] and Cox et al. [14] have estimated the heat flux removed by sprays using thermocouples under conditions that are close to industry operations (droplet diameters up to several millimeters and mass flux up to 30 kg/m² s^{−1} respectively). More recently, infrared thermography (IRT) was also applied to estimate the heat flux extracted from a heated wall by using liquid jets [27], deposited droplets [14,15], single droplets or monodisperse droplet streams [10]. With the same objective, Jia and Qiu [18] have used thermocouples and a PDA device to estimate the heat flux and both incoming and outgoing droplets characteristics respectively. Therefore, they found that the heat transfer depends on the expulsion rate defined as the ratio of the outgoing to incoming masses fluxes. However, the mass flux in this study was low (1.20 kg/m² s^{−1} at maximum). Puschmann et al. [20] have realized an overall study by using a 2D-PDA device for the droplet properties and a patternator for the mass flux distribution as well as an IRT method to estimate the heat flux. They well validated their experimental set-up but the mass flux still remains low (5 kg/m² s^{−1} in maximum).

Usually, the Weber number and the mass flux are the two common parameters used in order to classify and qualify the cooling of a hot surface by sprays. Fig. 1 depicts the values of the Weber number and the mass flux of the main studies encountered in the literature. It can be observed that very few studies have been performed with high mass flux values [21–26]. In fact, the main studies were operated with a maximum mass flux of about 20 kg/m² s^{−1} and with a Weber number range of about [10–1000].

Except Jia and Qiu [18], the majority of the herein mentioned studies are limited to the investigation of the heat transfer issues but the link with the characteristics of the droplets and the hydrodynamic conditions of the impact with the heated surface is generally not investigated. Therefore, the objective of the present work is to compare the cooling properties and to link them with the incoming droplets characteristics and impingement features. In addition, the long term objective of this study is to work as much

as possible under the industrial conditions (i.e. mass flux at least equal to 100 kg/m² s^{−1}). Thereby, an experimental set-up was specially designed. Nevertheless, before injecting such mass flux values, the present study is limited to a maximum mass flux of about 8 kg/m² s^{−1} in order to validate the experimental set-up. In a future paper, similar experiments will be performed by increasing gradually the mass flux. In the present work, three full cone sprays are used in order to vary the mass flux distribution as well as the droplet velocity and size leading to Weber number up to about 800. The operating conditions targeted in the present study are shown in Fig. 1. Additionally, the comparison with cooling by a liquid jet will be also considered. For the three sprays, both the heat flux removed from the wall and the characteristics of the droplets interacting with the heated surface were characterized simultaneously respectively by using a 2D-PDA system and an IRT method combined with an inverse heat conduction model.

2. Sprays facilities

The experimental test rig, sketched in Fig. 2, was designed in order to work at a high water mass flux, approaching conditions used in steel industry for hot surfaces quenching.

Fig. 1. Classification of the sprays plotted in the 2D-plan We – mass flux of the main studies encountered in the literature for sprays cooling.

Fig. 2. Experimental spray facilities and implementation of the optical measurement devices (2D-PDA and IRT). Side view (i.e. X–Z plane) of the test rig. 1 – pump1; 2 – water tank; 3 – pump2; 4 – manometer; 5 – flowmeter; 6 – 3D traverse system; 7 – insulation part (quartz); 8 – nickel disk; 9 – retractable electromagnetic inductor; 10 – mirror; 11 – infrared camera; 12 – IR camera protective cover; 13 – laser probe; 14 – PDA receiver; 15 – main tank for water collecting.

As the spreading of outcoming droplets (after impact) occurs on a large area, a $3 \times 4 \times 0.30 \text{ m}^3$ buffer tank (15) was built in order to pick-up water after impinging the surface and to recycle it in a loop circuit. A first pump (1) transfers the water from the main tank (15) up to a second tank (2). Then, a centrifugal regulated pump (3) ensures the circulation of the liquid to the spray nozzle at the desired operating parameters. The injection pressure is controlled with a manometer whereas the water flow rate is monitored by a magnetic flowmeter. The cooling of a hot surface with sprays or with a liquid jet is basically a transient phenomenon. Therefore, an experiment consists in two phases: the first one is the heating phase of the surface to be cooled while the spray is not flowing, and the second phase corresponds to the cooling, when the spray or the liquid jet is applied. Since the cooling process occurs within a short time (typically 25 s in the present study), the time to establish the spray flow is not negligible compared to the cooling time. This time can be reduced by maintaining constantly the centrifugal pump at the same operating point. Therefore, a second similar spray (same nozzle) that is flowing out of the heated surface domain during the heating phase was used, with an identical hydraulic circuit. Two piloted electro-valves (valve A and B) allow switching from one spray to the other one.

The surface to be cooled (8) is a nickel disk (Radius $R = 87.5 \text{ mm}$ and thickness $e = 5 \text{ mm}$) because the thermo-physical properties of this metal are very close to those of steel. Furthermore, metallurgical phase transitions are observed only above 1000°C . The disk is heated by electromagnetic induction (Power Supply 12 kW; Five-Celes®). The inductor wire (9) is placed just above the front face of the plate during the heating phase and is removed before the beginning of cooling. With this contactless heating method, both front and rear faces are accessible for the implementation of optical diagnostics. The disk, placed on a ring made of an insulating material (7), is positioned horizontally below the nozzle A, so that its center is aligned with the nozzle axis.

Droplet size and velocity measurements were performed in the vicinity of the front face of the disk by means of a Phase Doppler system (PDA) (Items (14) and (13) in Fig. 2). Since the PDA device (emission system and receiver) are fixed, the disk and the ensemble of the two nozzles are held together on a 3D-traverse system (6) that makes possible the displacement of the measuring volume of the PDA. The origin of the X–Y–Z axis is taken at the exit

of nozzle A and the Z-axis direction corresponds to the spray flowing direction. The density of the heat flux removed at the front face is estimated by infrared thermography coupled with an inverse heat conduction model of the rear face of the disk. The infrared signal emitted by the rear face of the disk (opposite to the spray impact) is directed by a 45° mirror to an infrared camera in order to measure the temperature field. The mirror is protected against water expelled by the spray by the insulating part (7) and the IR camera by a protective cover (12).

Since the objective of the present work is to investigate the cooling capabilities of several sprays in parallel with droplet size and velocity distributions as well as different mass flux distributions, five Lechler® nozzles were used. The selection of the kind of the nozzles is guided by the main following arguments:

- The circular surface to cool has to be completely wetted. Therefore, full cone sprays are chosen.
- To change the range of droplet velocity and diameter as well as mass flux values, one can change the injection pressure, the flow rate or the cone angle. In the present work, the flow rate was been fixed and the nozzle are then chosen in order to modify the cone angle or the injection pressure.
- According to the objective fixed for the current study, the mass flux is limited to about $10 \text{ kg/m}^2/\text{s}$. It corresponds roughly to flow rates $\dot{M}$ of about 7 l/mn . The nozzles are then selected to respect this condition.

Nevertheless, the analysis of the measurements obtained with the five nozzles shows that some results are similar. Therefore, in order to do not overload the figures of the present paper, three sprays are judiciously selected and are referenced as spray number 1–3. Table 1 provides the parameters for the three sprays. They have the same flow rate but both sprays 1 and 2 has a cone angle of 45° while nozzle 3 has a cone angle of 60° . Thereby, to ensure that the spray covers all the surface area of the target, the height H along the Z axis (distance between the nozzle exit and the front face of the disk) is adapted according to the disk radius and the value of the spray angle. Although the mass flux is the relevant parameter for the study of cooling by sprays, we prefer to indicate in Table 1 the flow rate in order to precise that the same quantity of water is injected each time for the three sprays used. For

Table 1

Characteristic parameters of sprays and liquid jet and experimental operating conditions.

Cooling device	Spray angle ($^{\circ}$)	Flow rate $\dot{M}$ (l/min)	Injection Pressure (bar)	Nozzle-disk distance H (mm)	Injection temperature T_i ($^{\circ}\text{C}$)
Sprays 1	45	6.7	7	250	19
Sprays 2	45	6.7	3.7	250	19
Sprays 3	60	6.7	3.7	160	19
Liquid Jet		6.7		250	19

comparison purposes, a liquid jet was also used. In order to generate the liquid jet, a 9 mm-diameter tube was used with similar operating conditions as nozzle 1, i.e. at $Z = 250$ mm above the disk and with a liquid flow rate of 6.7 l/min (Table 1).

3. Experimental diagnostics

3.1. Phase Doppler measurements

The properties of the sprays (velocity and size of the droplets) are performed by using a Phase Doppler system (PDA). The Phase Doppler system is a 2D commercial classic PDA manufactured by Dantec-Dynamics®. An argon ion laser, operating in multiline mode ($\lambda_v = 514.5$ nm and $\lambda_b = 488$ nm), is used to perform velocity measurements in two directions of the space. Both PDA excitation volumes (green and blue one) was formed by using a LDA transmitter probe (Dantec-Dynamics Fiber-Flow® probe) aligned along the X-direction (label (13) in Fig. 2). The system includes a classic PDA receiver and the receiver (label (14) in Fig. 2) is operating in the first refraction mode with an off-axis angle of 60° . The focal front lens of the transmitter probe is 1200 mm while the focal length of the receiver is 510 mm, leading to a maximum detectable droplet size of 327 μm . Finally the signal is analyzed with the use of a P80 signal processor. Fig. 3 presents typical droplet size distributions measured with the PDA system for the three sprays in the central section ($X = Y = 0$) and at the height H where the disk will be located (Table 1). All size distributions peak in the vicinity of the same diameter value ($d \approx 40 \mu\text{m}$) but the spray 3 produces the largest droplets sizes.

The PDA technique was used for two purposes: firstly, to determine the properties of the three free sprays in absence of the disk and secondly, to characterize the impingement of the droplets on the heated surface. For the characterization of the free sprays, each one was moved in the X–Y plane by using the 3D-traverse system in order to cover the whole cross-sectional area. More precisely, as the disk radius is $R = 87.5$ mm, the range for X and Y is taken equal to $[-90 \text{ mm}; 90 \text{ mm}]$.

In order to characterize the droplets interaction with the heated plate, i.e. incoming and outgoing droplets, the PDA measurement

volume was positioned in the close vicinity of the front face of the disk. The impact of the liquid on the disk was characterized by droplet splashing phenomena and vapor production, which causes attenuation and a major disturbance of the signal received by the PDA receiver. In order to overcome this problem, the PDA receiver and the laser beams system were tilted of an angle of 45° (Fig. 4). The consequence of this geometrical arrangement was corrected in the calculation of the two velocity components, so as to obtain them in X–Y–Z axis system, i.e. the component w along the Z-axis and the component v along the Y-axis. With this configuration, droplets impinging the disk had positive velocities whereas droplets having negative velocities are considered as outgoing droplets.

3.2. Heat flux density measurements

The density of the heat flux (i.e. heat flux per unit of surface) removed at the front face of the disk was estimated by means an infrared camera combined to an inverse heat conduction method (IHCM). The infrared camera (Cedip® Jade III) is built with a focal plane array photonics detectors (InSb) working in the spectral range $[3\text{--}5 \mu\text{m}]$ and it is equipped with a narrow-bandwidth $[3.97\text{--}4.01 \mu\text{m}]$ spectral filter to fit with the expected experimental temperature range. The spatial resolution is 320×240 pixels/frame. The sampling rate of the camera can be adjusted according to the expected phenomenon duration. Since the cooling phase has a typical duration of about 25 s, the selected number of frames per run is about 2000. Thus, the sampling frequency for all experiments conducted in this study was adjusted at about 60 Hz. In order to measure accurately the surface temperature of the nickel disk, one face of the disk (rear face) was previously sandblasted in order to obtain a diffuse surface. In a second step, the surface was oxidized (with O_2 of air) at 800°C during two hours. The formed layer of oxide is sufficiently thin to avoid any undesirable effect on heat transfers and presents sufficiently stable radiation properties (measured value of the total emissivity, $\varepsilon_r = 0.64$). A measurement using a thermocouple is required for the calibration of the detector. A thermocouple, inserted in

Fig. 3. Typical droplet size distributions for the three, nozzles when the disc is not present.

Fig. 4. Front view (i.e. Y–Z plane) of the PDA arrangement (emission system and receiver).

the centerline of the rear face of the disk, was used to provide a reference temperature: this measurement is required to calibrate the whole measurement chain, i.e. to account for the mirror transmission and of the environment. The calibration is performed by heating the disk up to 800 °C and by recording simultaneously the infrared signal and the temperature measured by the thermocouple, while the disk is cooled naturally (radiative, convective heat transfer and residual conduction through the insulating parts).

Knowing the temperature field at the rear face ($Z = H + e$), an inverse heat conduction model is used to estimate the heat flux density removed by the sprays or the liquid jet on the front face ($Z = H$). The 3D internal conduction problem can be reduced to a 2D one by applying angular averaging, in order to obtain an estimate of the averaged value of the heat flux density $q''_{\text{spray}}(r, Z, t)$. Then, the 2D internal transient conduction reduced problem within the nickel circular disk is solved assuming the following hypothesis: constant nickel thermophysical properties, no conduction losses due to the disk holder (item (7) in Fig. 2) and no losses at the edge of the disk. A full description of the IHCM is available in previous papers [27–30]. Due to the presence of the insulation part, the disk area that is visualized by the camera is truncated. Thus, following heat flux results are presented for an area having a maximum radius of 70 mm.

3.3. Measurements of the mass flux distributions

When sprays are used, the cooling of a hot surface is strongly correlated to the mass flux distribution. Therefore, a patternator, composed of a row of twelve tubes connected to individual sampling vessels, was specially designed in order to measure the distribution of the mass flux for each nozzle. The diameter of each individual tube is 7.5 mm, leading to a surface of collection of 44.5 mm². The distance between two consecutive tubes is twice the diameter tube value. The patternator is placed in a manner that the top of the tubes are located at the same Z elevation from the front face. The use of the 3D-traverse system allows moving the spray in order to scan whole cross-sectional area in the X – Y plane. The mass flux was measured during a given time, on the order of 180 s per measurement. An error of about one second and 1 ml can be admitted, leading to an average uncertainty of about 5% on the mass flux. Typically, the mass flux distribution is described with a mesh having 11×11 measurement points for each nozzle. As PDA measurements, the range for X and Y is also [–90; 90 mm] in order to cover the entire spray cross-sectional. Despite of the number of measurements points used, it is verified that, by integrating the liquid flux distribution obtained, the total flow rate injected cannot be collected. Typically, about 85% of the initially injected water is collected for the three sprays.

4. Acquisition procedure and data reduction

As mentioned in Section 2, each experiment begins by heating the nickel disk before starting the cooling process. To heat up the disk, the inductor is placed as close as possible to the surface in order to ensure a fast heating. The heating phase is monitored by the IR camera and is stopped when the average rear surface temperature is on the order of 800 °C. A short standby time before starting the cooling phase is necessary to reach the uniformity of the surface temperature by conduction in the nickel. After switching on both sprays, both PDA system and IR camera are synchronized during the cooling phase. To attain this goal, the trigger output of the PDA processor is used to drive the IR camera acquisition and the spray operations (electro-valves A and B). The recording time Δt_r depends on the cooling time, which is given when the nickel surface temperature is equal to the initial liquid temperature ($T_l = 19$ °C). As different liquid flow rates are used, Δt_r varies from 20 up to 30 s according to the spray or liquid jet. The measurement is repeated twice for each device (sprays or liquid jet) in order to verify the repeatability of the cooling. The analysis of the results aims to correlate the local heat flux removed from the wall with the local mass flux and droplet properties distribution. As heat flux measurements are presented as a function of the disk radius, a reduction procedure of PDA and mass flux data is necessary to perform the further analysis. Basically, it consists in calculating a value for each of the measured quantities averaged over the azimuthal coordinate, within an element of crown centered on the nickel disk (radius r , thickness dr). Thereby, all averaged quantities (from IRT, PDA and mass flux measurements) depend only on one spatial variable r varying from 0 up to R (note that as mentioned before, for the IRT technique, the radius is limited to 70 mm).

5. Characteristics of the free sprays

Fig. 5a gives an example of the mass flux map obtained with the spray 1 in a cross-sectional area of the spray located at $Z = 250$ mm. It depicts clearly the non-uniformity of the mass flux distribution. The evolution of the averaged mass flux along a radius for the three nozzles is displayed in Fig. 5b. As the three sprays have the same injected flow rate, the curve for spray 3, being widely above the two other one, may be surprising. But it has been well verified that, by integrating within a crown, all curves lead to similar flow rate. It appears that the mass flux distributions of the three sprays are completely different. On one hand, both sprays 1 and 2 seem to have the same behavior with an almost identical value of the mass flux (at the vicinity of $4 \text{ kg/m}^2 \text{ s}^{-1}$) along the radius even if the mass flux of the spray 1 is slightly greater than the spray. On the other hand, the mass flux of the spray 3 decreases from the

Fig. 5. Typical mass flux distribution measured in a cross-section obtained with spray 1 (a) and along a radius for the three sprays after data reduction (b).

Fig. 6. Evolution of the droplet statistical properties along a radius obtained from the PDA measurements after data reduction for the three sprays: mean vertical velocity (a), mean horizontal velocity (b), SMD (c) and mean Weber number based on the SMD (d).

center from $8 \text{ kg/m}^2 \text{ s}^{-1}$ to $2.5 \text{ kg/m}^2 \text{ s}^{-1}$ at the edge. The major difference may be linked to the spray angle, which may explain this behavior. The other tested sprays (not displayed in the present paper), having also angles between 45° and 60° , confirm this trend.

Fig. 6 presents the statistical properties of the droplets along the disk radius derived from the 2D-PDA measurements. The evolution of the w velocity component (6a), the v velocity component (6b), the SMD d_{32} (6c) are plotted as a function of the radial coordinate r . As expected, spray 3 has the highest horizontal velocity (v component) due its higher cone angle. Moreover, it appears that the spray 3 has also the greatest SMD (as well as the mean diameter but not shown here) but not necessarily the highest vertical velocity. By observing Fig. 5b, it can be deduced that the high value of mass flux for the spray 3 may be due to the presence of larger droplets. In spray cooling, a normal Weber number based on the SMD and the mean statistic vertical velocity w is commonly introduced:

$$We = \frac{\rho_l d_{32} w^2}{\sigma_l} \quad (1)$$

where ρ_l and σ_l are the mass density and the surface tension respectively. The normal Weber number is plotted in Fig. 6d as a function of r . It tends to decrease with r for each the investigated nozzles. The spray 2 leads to the lowest values whereas the curves for both sprays 1 and 3 are almost superimposed. Indeed, even if the spray 3 has larger droplets, its vertical velocity being lower than for the spray 1, the product $w^2 d_{32}$ is almost the same for both sprays 1 and 3.

6. Investigation of the cooling phase

6.1. Influence of the wall on the incident droplets properties

To characterize the cooling of a heated surface by a spray, the incoming droplet properties are often measured in absence of the surface. However, the presence of the surface is likely to influence the trajectories of the droplets and therefore their impingement characteristics. The main goal of this section is to tackle this problem. The PDA results for the incoming droplets in the case of an impinging spray are compared with those obtained in the case of a free spray (without the disk). The case of spray 1 is illustrated in Fig. 7a when the measurement volume is at $Z = 240 \text{ mm}$, i.e. 10 mm above the center of the nickel disk, and when the disk is not heated. Similar experiments have been conducted when the surface has been heated. But, the analysis and the discussion of these results are beyond the scope of the present study and will be the object of another paper. A significant difference between the free spray and the spray in presence of the disk can be observed: the pdf of the droplet size measured in the case of the impinging spray exhibits a shift towards larger droplets sizes compared to the case of the free spray. This phenomenon can be attributed to the deviation of the flowfield by the disk, resulting in a deviation of the smallest droplets ($d < 50 \mu\text{m}$) which are likely to follow the deviated flow due to their small Stokes number. This observation can be confirmed by the careful observation of the evolution of horizontal velocity v as a function of the droplet diameter (Fig. 7b). The horizontal velocity component of the smallest incoming droplets increases significantly in the case the impinging spray, which reveals the deviation of the smallest droplets towards the disk edges. Therefore, the effect of the surface to be cooled on

Fig. 7. Comparison of the incoming droplets parameters measured in the case of a free spray and in presence of the disk in the case of spray 1 (disk not heated): pdf of the size distributions (a) and correlation size/horizontal velocity as a function of the diameter (b).

the spray flow can modify significantly the distributions of droplet size and velocity in the impingement region.

6.2. Interaction of the spray with the heated wall

The PDA results presented in this section are obtained by keeping PDA data acquired only during the Leidenfrost regime. By using heat flux results (see Section 6.3 below) it is possible to determine the time interval of the Leidenfrost regime (i.e. between the initial time when the spray is applied and the time when the Leidenfrost temperature is reached) and thus to associate corresponding PDA data. Measurements are performed at a distance of 10 mm above the front face of the heated wall at $X = Y = 0$. Fig. 8 describes typical droplet size distributions of incoming and outgoing droplets for the spray 1. The outgoing droplets are generally distributed on smaller size classes than the incoming one. The distributions for the outgoing and incoming droplets are peaking for a diameter value $d = 40 \mu\text{m}$ and $d = 50 \mu\text{m}$ respectively. More generally, the results show that the maximum detected droplet size is around $200 \mu\text{m}$.

The ratio between the outgoing Sauter mean diameter $d_{32,o}$ and the incoming Sauter diameter $d_{32,i}$ for the three sprays is presented in Fig. 9 as well as the ratio for the mean diameter ($d_{10,o}/d_{10,i}$). A reduction of the SMD and mean droplet diameter is observed for the three investigated cases. Even if there is no significant difference between the three sprays, the spray 3 presents the lowest reduction of the statistical diameters during the impingement. This could be related to the lower normal Weber number of spray 3 in comparison to the other (Fig. 6d) which

Fig. 8. Typical incoming and outgoing droplets size distribution.

Fig. 9. Ratio of the outgoing Sauter mean diameter to the incoming Sauter diameter and ratio of the outgoing mean diameter to the incoming mean diameter, for the three sprays.

involves a less efficient splashing on the heated surface. Indeed, some authors, as Moita and Moreira [7] and Dewitte [9], have shown that an increase of the incident We number leads to a decrease of the values of both SMD and mean diameter for outgoing droplets. Generally, the outgoing droplets, in the Leidenfrost regime, are mainly the results of rebound or splashing [7,8]; they may be also affected by evaporation, but the expected change in diameter for the largest droplet may be not significant [10]. Even it is not possible to identify the outgoing droplets resulting from a rebound or from splashing, the calculation of the incident Weber number for each of the size can shows which regime is dominant. The Weber number can be calculated for each droplet size classes:

$$We_i = \frac{\rho_l d_i w_i^2}{\sigma_l} \quad (2)$$

where ρ_l is the liquid specific mass and σ_l is the liquid surface tension, i relates to the i th size class, d_i is the median droplet and w_i is the corresponding vertical velocity.

Usually [7,8], the threshold between the rebound and splashing regimes is given for a Weber number $We = 60$. The size distribution of the incoming droplets is represented as a function of the droplet Weber number per size class in Fig. 10 for the three sprays. The Weber number was calculated by using the correlation of the size and the vertical velocity, measured by the PDA. It shows that all the distributions peak for a Weber number of about 100. The majority of the incoming droplets are impinging the disk with a Weber

Fig. 10. Droplet size distribution of the incoming droplets represented as a function of the Weber number for the three considered sprays.

number superior to 60, leading to the conclusion that the main regime of interaction with heated plate is splashing.

The Weber number can be calculated if the droplet diameter–velocity correlation is known. This correlation, which can be determined by the PDA system, is reported in Fig. 11 for both incoming (Fig. 11a) and outgoing droplets (Fig. 11b), in the case of the vertical velocity. The correlation for the incoming droplets is usual for the pressure effect atomizers, the smallest droplets having the smallest velocity. The correlation for the outgoing droplets shows the contrary: the smallest droplets are expelled faster than the large ones.

6.3. Investigation of the heat transfers rate

6.3.1. Investigation of the heat flux removed from the wall

Fig. 12 depicts a spatio-temporal map of heat flux density extracted by the spray 1 and 3 along the radius (horizontally) and as a function of the time (vertically). The color scale is kept identical for spray 1 and 3 in order to ease the comparison. The main comments are the following:

- For spray 1 up to $t \approx 7$ s, the heat flux is almost spatially uniform. A second phase is characterized by non homogeneous regions. The limit of the film boiling regime (Leidenfrost regime), where the heat flux is minimum, is reached for $0 < r < 30$ mm at $t \approx 13$ s. Then, for $0 < r < 30$ mm, the critical heat flux (CHF), corresponding to an inflection point, is visible at $t \approx 18$ s. From $t \approx 20$ s up to the end of the cooling phase, the nucleate boiling regime takes place and the removed heat flux decreases and exhibits an almost homogeneous distribution along the radius.

- In comparison, when spray 3 is considered, the CHF is reached very quickly ($t \approx 5$ s) throughout the radius, but the transition between the film boiling regime and the nucleate boiling regime cannot be clearly identified. Before reaching the CHF point, the heat flux presents strong spatial heterogeneities. Conversely, past the CHF point, the cooling rate is relatively homogeneous.

The boiling curves for the three sprays can be also determined: they represent the evolution of the extracted heat flux density as a function of the wall temperature T_w (Fig. 13). The boiling curves are plotted for two values of the radius: $r = 2.5$ mm (a) and $r = 70$ mm (b). First of all, as the vertical axis scale is the same for both positions, the spray 3 exhibits a strong spatial heterogeneity of the heat flux. Next, at $r = 2.5$ mm (i.e. close to the disk center), the heat flux extracted by spray 3 is clearly higher than for the two other. At $r = 70$ mm, the heat flux for sprays 2 and 3 are almost similar. For both positions, spray 1 leads to the lowest heat flux. The literature provides several correlations for the heat flux extracted in the film boiling regime. These correlations depend mainly of the SMD (or mean diameter), the vertical velocity, the mass flux or the We number. By observing the Fig. 5b, the heat flux for the spray 3 can be correlated with its mass flux value: between $r = 0$ and $r = 70$ mm, both heat flux and mass flux decrease. This observation is often noticed [5,6,25–27,31]. On the other side, this trend is not clearly verified for the two other sprays. The heat flux for the spray 1 is the lowest whereas it depicts higher mass flux than spray 2. It means that other parameters may influence the extraction of thermal energy by spray. According to Fig. 6a, it appears that the spray 1 presents higher mean vertical velocity values than spray 2. It could be assumed then that the increase of the initial droplet kinetic energy leads to a decrease of the cooling power. Nonetheless, the effect of the sole vertical velocity on the heat flux remains unclear and can be contradictory. Several authors [5,24] have revealed that heat transfer during the film boiling regime increases with decreasing the droplet velocity. Controversially, in other paper it was found that heat transfer increases with the velocity [6,31].

6.3.2. Investigation of the Leidenfrost temperature

The Leidenfrost temperature can be determined from the boiling curves at a given position along the radius, when the heat flux experiences a minimum value. Compared to spray 3, the Leidenfrost point is easy to identify for spray 1 and 2, i.e. when the mass flux is low. This observation is often mentioned in the literature: the increase of the Leidenfrost with the mass flux being related to the absence of a stable vapor film. However, the Leidenfrost temperature tends generally to increase with the mass flux

Fig. 11. Correlation size/vertical velocity of the incoming droplets (a) and of the outgoing droplets (b) for the three sprays.

Fig. 12. Evolution of the heat flux density extracted from the disk along the radius and as a function of the time for the spray 1 (a) and for the spray 3 (b).

Fig. 13. Boiling curves recorded at $r = 2.5$ mm (a) and $r = 70$ mm (b).

[26,31]. For instance, Yao and Cox [31] introduced the spray Weber number based on the SMD and the mass flux:

$$We_s = \frac{\dot{m} d_{32}}{\rho_l \sigma_l} \quad (3)$$

The mass flux is preferred because the direct effect of the droplet velocity is not clearly established. They have shown that T_{Leid} can be linked with the spray Weber number according to the following correlation:

$$T_{Leid} = 1400 We_s^{0.13} \quad (4)$$

Fig. 14 presents the evolution of the Leidenfrost temperature as a function of the spray Weber number for the three sprays where the Leidenfrost temperature is obtained at several positions along the radius (i.e. for each spray, one dot corresponds to one radial position). As the mass flux and the SMD is measured, it is possible to derive the spray Weber number and thus to plot the correlation of Yao and Cox. It appears that T_{Leid} increases with We_s , i.e. with the mass flux, which is in agreement with Yao and Cox [31] and Al-Ahmadi and Yao [26]. In addition, the best agreement with the experimental data is found for the highest mass flux. Usually, the boundary between low and high mass fluxes is not really defined and determined. For the considered sprays tested in this study, the Fig. 14 could show us where this boundary is. More precisely,

Fig. 14. Evolution of the Leidenfrost temperature as a function of the spray Weber number at several positions along the radius and comparison with the correlation of Yao and Cox [31].

we can assume that spray 3 is almost the case for high mass flux whereas the two other sprays are the case for low mass flux.

6.3.3. Determination of the cooling efficiency

Usually, the cooling ability of a spray can be quantified by the cooling efficiency ξ . It is given by [3]:

$$\zeta = \frac{q''}{\dot{m}[h_v + c_{pl}(T_{sat} - T_l) + c_{pv}(T_w - T_{sat})]} \quad (5)$$

It represents the amount of heat flux extracted by the spray q'' divided by the total heat flux which is required for complete evaporation of the liquid, which includes the heat flux required to heat the liquid up to its saturation temperature and to heat the vapor up to the wall temperature. Its evolution along the radius is presented in Fig. 15 for the three considered sprays and for a given wall temperature T_w (i.e. for a given time of the cooling phase). Here, T_w is chosen at 560 °C that corresponds to the film boiling regime.

It appears that the three sprays leads to comparable cooling efficiency with a mean value on the order of 6%. In addition, no significant evolution is observed along the radius. In several previous studies [4,11,16,21,22,24–26], it is often observed that an increase of the mass flux is correlated to a decrease of the cooling efficiency and it is in well agreement with Eq. (5). Therefore, the spray 3, having high mass flux, should have the lowest cooling efficiency. But, this spray has also the higher heat flux leading to increase the cooling efficiency in accordance with the expression of the Eq. (5). Yao and Cox [31] have highlighted that the cooling efficiency decreases with the spray Weber number. However, it is expected that the wall temperature T_w as well as the liquid temperature T_l can also influence the cooling efficiency. Based on experiments results derived from multiple investigations on different sprays, these authors proposed a correlation for the cooling efficiency, based on We_s , T_l and T_w :

$$\zeta = 8 \times 10^{-7} \left[\frac{We_s T_{sat}}{T_w - T_l} \right]^{-0.62} + 3.5 \times 10^{-7} \left[\frac{We_s T_{sat}}{T_w - T_l} \right]^{-0.2} \quad (6)$$

This correlation was validated over a wide range of operating conditions and falls in the range of the conditions of present study. Knowing the distribution of the liquid mass flux, the SMD (Figs. 5b and 6b respectively) and the wall temperature, the quantity $We_s T_{sat}/(T_w - T_l)$ can be calculated. Therefore, it is possible to compare the cooling efficiency obtained in the present study (Eq. (5)) with the correlation of Yao and Cox [31] (Eq. (6)). Fig. 16 depicts the evolution of the cooling efficiency along the radius obtained with the spray 1 for two wall temperatures (560 °C and 400 °C) that are above the Leidenfrost temperature (Fig. 14). The correlation of Yao and Cox (Eq. (5)) is also plotted for both temperatures.

For both considered wall temperatures, the correlation of Yao and Cox tends to underestimate the results of the present work. Nevertheless, for both cases (correlation of Yao and Cox and present work), the cooling efficiency increases with the wall temperature. This qualitative behavior is frequently observed in the literature in the case of the Leidenfrost regime [5,6,24,31] and is linked to an increase of the convective heat transfer within the vapor film.

Fig. 15. Evolution of the cooling efficiency along a radius for the three sprays (wall temperature is $T_w = 560$ °C).

Fig. 16. Evolution of cooling efficiency along a radius for the spray 1 (wall temperature is $T_w = 560$ °C). Comparison with the correlation of Yao and Cox [31].

7. Comparison of the cooling devices performance

The goal of this section is to compare the cooling performances of the three sprays and of the liquid jet. To make this comparison relevant, a common temperature range of cooling was defined in order to calculate:

- The thermal energy density extracted during the cooling time.
- The coolant water consumption over the defined cooling period.

Preliminarily, the cooling time was the time necessary to cool the disk surface from $\bar{T}_w = 546$ °C to $\bar{T}_w = 27$ °C this range embedding both film boiling and rewetting regimes. Here $\bar{T}_w$ refers to the space-averaged surface temperature, defined by:

$$\bar{T}_w(t) = \frac{1}{\pi R_1^2} \int_0^{R_1} T_w(r, t) 2\pi r dr \quad (7)$$

Note, that according the Section 3.2, the surface temperature is integrated for a maximum radius of $R_1 = 70$ mm. At each time step t of the cooling phase, the space-averaged heat flux removed from the wall was also calculated:

$$\bar{q}''(t) = \frac{1}{\pi R_1^2} \int_0^{R_1} q''(r, t) 2\pi r dr \quad (8)$$

Using Eqs. (7) and (8), the space-averaged cooling efficiency at each time step t can be derived:

$$\bar{\zeta}(t) = \frac{\bar{q}''(t)}{\dot{m}_0[h_v + c_{pl}(T_{sat} - T_l) + c_{pv}(\bar{T}_w(t) - T_{sat})]} \quad (9)$$

where $\dot{m}_0$ is the space-averaged mass flux. For the three sprays, $\dot{m}_0$ is determined from the data obtained with the patternator of the local mass flow rate (Fig. 5b):

$$\dot{m}_0 = \frac{1}{\pi R_1^2} \int_0^{R_1} \dot{m}(r) 2\pi r dr \quad (10)$$

Two other time integrated parameter can be deduced:

- The total (cumulative) energy density extracted by the cooling device:

$$E = \int_0^{\Delta t} \bar{q}''(r, t) dt$$

- The total power density extracted by the cooling device:

$$P = \frac{E}{\Delta t}$$

where Δt is the total cooling time.

Obviously, the case of the liquid jet is particular since the cross-section of the jet does not cover the whole surface of the disk.

Therefore, to perform a proper comparison with the case of the sprays, an equivalent value of $\dot{m}_0$ was used by dividing the jet liquid mass flow rate by the surface area considered in the spray cooling experiment, $\dot{M}/\pi R_1^2$. The main results are gathered in Table 2.

The evolution of the space-averaged heat flux as a function of the time is depicted in Fig. 17 for the three sprays and the liquid jet. The main comments are the following:

- Among the three sprays and the liquid jet, the spray 3 is the fastest device to cool the disk from a wall temperature of 546 °C up to 27 °C, with a cooling duration of about 12 s. It experiences also the highest heat flux value during the major part of its cooling time.
- Controversially, even if high heat flux can be extracted by the liquid jet, a long cooling time is experienced, with a duration of about 23 s, i.e. almost twice the value of spray 3. It can be attributed to the fact that the direct impingement of the jet occurs on a limited area of the surface to be cooled. In addition, the extracted heat flux presents strong fluctuations before reaching the CHF (Leidenfrost point being located at about $t \approx 17$ s) probably due to oscillating thickness of the vapor film separating the liquid and the solid wall. These fluctuations disappear beyond the Leidenfrost point when the disk surface is entirely wetted.
- The spray 1 shows relatively bad performances: the lowest heat flux and a quite long cooling time. As already noticed in Fig. 13, even if the spray 1 has a higher mass flux value compared to the spray 2, the extracted heat flux by spray 1 remains smaller.

The evolution of the space-averaged cooling efficiency is presented in Fig. 18 for the four devices. Sprays 2 exhibits the highest cooling efficiency, which may reveal the performances of spray 2 are better over all the cooling period. In the opposite, in spite of its potential to extract a high heat flux, the cooling efficiency of the liquid jet is the lowest compared to the spray systems. The cooling efficiency of spray 1 remains under of those of the two other sprays.

Fig. 19 describes the evolution of the cumulative energy density extracted from the wall (E in J/m²) as a function of time, in the above specified temperature range, i.e. between $\bar{T}_w = 546$ °C and $\bar{T}_w = 27$ °C for the three sprays and the liquid jet. The liquid jet extracts the highest quantity of energy and the three sprays removed roughly the same quantity of energy, but the duration of the cooling phase appears different for each of the sprays. By dividing the cumulative energy density by the duration of the cooling process, it is possible to derive a cooling power density and the liquid flow rate being known, the coolant water consumption can also be inferred (Table 2).

These different results lead to the following comments:

- In spite of its high relatively cooling power (10.5 kW/m²), the liquid jet exhibits the highest water consumption compared to the three sprays. This results to the high heterogeneity of the cooling rate at the surface.

Fig. 17. Evolution of the space-averaged heat flux as a function of the time for the three sprays and the liquid jet and for a similar surface temperature range.

Fig. 18. Evolution of the space-averaged cooling efficiency as a function of the time for the three sprays and the liquid jet and for a similar surface temperature range.

Fig. 19. Evolution cumulative energy density extracted by the three sprays and the liquid jet and for similar wall temperature range.

Table 2
Comparison of for the cooling performances of the three sprays and the liquid jet.

Cooling device	Total extracted energy (kJ/m ²)	Cooling duration (s)	Mean cooling power (kW/m ²)	$\dot{m}_0$ (kg/m ² /s)	Sprayed flow rate $\dot{M}$ (L/mn)	Coolant fluid consumption (L)
Spray 1	161	20.5	7.9	4.29	6.7	2.3
Spray 2	155	14.3	10.9	3.76	6.7	1.7
Spray 3	152	12.2	12.6	5.26	6.7	1.4
Liquid jet	235	22.7	10.5	7.25	6.7	2.74

- Among the four devices, the spray 3 leads to the best trade-off, since it exhibits the highest cooling power (12.6 kW/m^2) and the lowest water consumption. This probably could be related to its high mass flux value as well as to its droplets properties when they impinge the heated surface. Spray 3 is singularized by the largest spreading angle, the highest horizontal velocity and the lowest vertical velocity. But it has not the highest cooling efficiency. It could be linked to its higher mass flux in the vicinity of the center of the heated disk, which can thus result in a kind of heterogeneity of the cooling.
- Among the three sprays, it appears that the spray 1 presents the worst performances with the lowest cooling power (7.9 kW/m^2) and the highest water consumption. This can be related to its high impingement vertical velocity, trend already observed in the locally results (Fig. 13).

8. Conclusions

An experimental set-up was developed in order to investigate water cooling of high-temperature surfaces under conditions that are representative to an industrial situation encountered in steel processing. The cooling by a liquid jet and three polydisperse full cone sprays are compared. The wall, a 175 mm diameter nickel disk, is heated by an induction device up to 800°C before to be quenched. The heat flux was estimated with infrared thermography combined with an inverse conduction method and synchronized to a Phase-Doppler system. In first, the effect of the presence of the wall on the motion of the incoming droplets was highlighted. The trajectories of the smallest droplets, which behave as passive tracers, are deviated by the plate, which reduce their impingement probability on the heated target. The comparison between the statistics of the incoming and outgoing droplets shows a slight decrease of the mean Sauter diameter, which is attributed to the disintegration of the largest droplets into to smaller one and to some evaporation of the liquid.

When spray cooling is considered, the cooling efficiency decreases with the mass flux and the Leidenfrost temperature increases, which is in qualitative agreement with other studies of the literature. Nevertheless, the role of the mass flux on the heat flux removed is still unclear. In the present work, the heat flux seems better correlated with the impingement vertical velocity, where an increase of this parameter leads to a decrease of the heat flux. Finally, a comparative study of performances, in term of liquid consumption and cooling duration, is performed by considering a similar surface temperature decrease for the three sprays and the liquid jet. Among the four devices, the liquid jet has the longest cooling time and the highest liquid consumption. On the contrary, the best performances are reached for the spray having the highest mass flux value together with the highest horizontal velocity.

Future works will be devoted to investigate the case of sprays of higher mass flux with an extended range of injection conditions. In addition, high speed camera visualisation will be performed in order to describe more thoroughly the interactions of the spray with the heated plate and to identify the different hydrodynamic regime (rebound, splashing, and film deposition).

Conflict of interest

None declared.

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