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General principles of controlled water cooling for metallurgical on-line hot rolling processes: forced flow and sprayed surfaces with film boiling regime and rewetting phenomena

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Abstract — Recent progress in metallurgy processes has made it necessary to perform accelerated cooling on-line after rolling (manufacture of plane products like strips or plates, or long products like rods or extrusions). For such heat treatment the cooling rate, the temperature fall and the temperature homogeneity have to be closely controlled. Then, owing to the temperature of the products, it is of paramount importance to perfectly know the cooling efficiency through the film boiling regime and when rewetting occurs. A theoretical model, well correlated with experiment, described the cooling with forced convecting sub-cooled water in the film boiling regime. It is based on transient thermal diffusion of a temperature step through the water layer, the turbulent diffusive properties of which are dependent on the water flow configuration. It is shown also, for a horizontal plane plate cooled with sprayed water jets on both faces, that the structure and general orientation of the water flow has a direct influence on the spatial distribution of the cooling heat fluxes. Consequently, the transposition principles from experiments conducted with small samples to industrial scale are pointed out. In addition, the rewetting phenomenon is analysed through the contact temperature concept between the liquid water and the surface of the product being cooled. For this phenomenon the nature of the very superficial layer is of first importance (scale or oxidations). Based on such models and analyses, numerous cooling facilities of various sizes have been developed by Bertin Technologies and are widespread all over the world: big size for steel plates (5 m wide by 20 m long), reduced size for aluminium rods (diameters from 10 to 35 mm) and aluminium extrusions (typical extrusion section within 300 mm by 300 mm). © 2001 Éditions scientifiques et médicales Elsevier SAS

film boiling / rewetting / sprayed surfaces / metallurgy cooling / accelerated cooling

Nomenclature

a	thermal diffusivity	$m^2 \cdot s^{-1}$
Cp	specific heat	$J \cdot kg^{-1} \cdot K^{-1}$
C_R	cooling rate	$K \cdot s^{-1}$
d	rod diameter	m
d_e	inside duct diameter	m
d_H	hydraulic diameter	m
e	plate thickness	m
g	gravity acceleration	$m \cdot s^{-2}$
h	heat transfer coefficient	$W \cdot m^{-2} \cdot K^{-1}$
H	height of water on plate	m
k	thermal conductance	$W \cdot m^{-2} \cdot K^{-1}$
Pr	Prandtl number $= \mu Cp/\lambda$	
Q_s	mass flux	$kg \cdot m^{-2} \cdot s^{-1}$
Re	Reynolds number $= \rho V d_H/\mu$	
St	Stanton number $= \alpha/(\rho V Cp)$	
t	time	s
q	volume flow rate per weir unit length	$m^{-2} \cdot s^{-1}$
V	liquid velocity	$m \cdot s^{-1}$
x	distance, length	m

Greek symbols

α	convective heat transfer coefficient	$W \cdot m^{-2} \cdot K^{-1}$
λ	thermal conductivity	$W \cdot m^{-1} \cdot K^{-1}$
ϕ	heat flux density or heat flux for short	$W \cdot m^{-2}$
ρ	density	$kg \cdot m^{-3}$
θ	temperature	K
$\Delta\theta$	temperature step	K

Subscripts

c	at contact

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sat	fluid saturation condition usually at atmospheric pressure
t	turbulent
o	liquid at entry or bulk
m	mean value for a given area
p	plate
s	superficial
sup	superheated liquid
x	local value at distance x

Superscript

*	dimensionless value

1. INTRODUCTION TO METALLURGICAL COOLING

There is a steady tendency in metallurgical industry for promoting continuity in processing metal transformation from production to half finished state (plates, beams, extrusions, rods). By doing so, certain intermediate cooling or reheating can be suppressed. For example, an accelerated and controlled cooling, on-line after rolling, help a lot to strongly improve the mechanical characteristics of the corresponding products. High cooling rate, either constant or well controlled, and precise final temperature are required to optimise simultaneously the cooling and the thermomechanical treatment.

Such metallurgical coolings concern such a huge quantity of products that the preferred cooling fluid is water at ambient temperature. The product initial temperature at the starting of cooling is roughly 800 °C with steel, and between 400 and 500 °C with light aluminium alloys. Within this range of starting temperature, the water cooling begins always in the situation of film boiling and ends with nucleate boiling. The superficial temperature of the metal is the main parameter when cooling with liquids and the heat flux will highly fluctuate versus this temperature according to Nukiyama [1]. Following the cooling, the superficial temperature decreases till the so-called rewetting temperature. Above this value, the heat flux is rather constant. Below this value, the heat flux increases a lot with the decrease of the surface temperature.

This behaviour is encountered with forced flow and sprayed surfaces. For numerous forced flow cooling applications, one tries to avoid the film regime. There have been rather few efforts to examine the detailed aspects of convective film boiling as recognised by Carey [2]. More studies have been performed on sprayed surfaces that mainly concern:

- the behaviour of individual droplets impacting on hot surfaces, with mass flux well below 1 kg·s^{-1}·m^{-2}, see Lee et al. [3];
- the dilute sprays with low water mass flux, up to 2 kg·s^{-1}·m^{-2}, see Deb and Yao [4];
- the sprays with small surfaces, less than several 10 cm^2, like for the two previous references and the results of Viannay and Moureau [5] for mass flux up to 13 kg·s^{-1}·m^{-2}.

The influence of the orientation of the surface relative to gravity and the size of the surface are rarely investigated.

In the present study, sprayed surfaces are in the range 1–100 m^2 and the mass flux range is 1–50 kg·s^{-1}·m^{-2}. Added to the specific constraints due to the use of liquid, are the specifications wanted by the metallurgist: high cooling rates, precise final temperature and a good final temperature homogeneity all over the product.

The following three main needs have to be fulfilled to fit the cooling specifications:

- to find a way to obtain high heat fluxes during film boiling regime;
- to have a good knowledge of the parameters conditioning the heat flux distribution in such a way as to avoid residual strains due to spatially nonhomogeneous cooling;
- to know the criteria that influence the transition mechanisms from film boiling to rewetted regime to be able to keep the cooling heat flux steady during the transition, with flow rate adjustment, for example.

To answer these questions an original scientific approach has been developed, along with cooling devices dedicated to metallurgical cooling. We describe hereafter this approach that is quite general and can be useful for other applications. This includes:

- a theoretical and experimental heat exchange model for film boiling with subcooled forced flow;
- a description of the spatial cooling heat flux arrangement resulting from spraying a plane plate;
- a criterion that specifies the beginning of the transition from the film boiling regime towards the nucleate boiling regime.

2. FILM BOILING MODEL WITH SUBCOOLED FORCED FLOW

The basis of this model has been published by Viannay and Karian [6].

This model is partly experimental and partly theoretical. It entails the following steps:

- detailed phenomenological model elaboration justified with order-of-magnitude calculations; this step is the key of the approach: it assembles very simple elementary physical phenomena;
- mathematical model elaboration issued from the phenomenological model aiming at comprehension of the experimental results;
- experiments conducted in accordance with the above approach.

In fact, these three steps were jointly worked out in such a way as to reach an overall coherent approach. This model has been proved useful to understand also the cooling of large horizontal surfaces sprayed from the top.

2.1. Phenomenological model

This model describes the cooling of a very hot wall with a confined forced flow of a subcooled liquid experiencing the film boiling regime. The assumed disposition is given *figure 1*:

- the product to be cooled is running across a duct, the section of which suits the product geometry (i.e. rectangular or circular);
- the duct extremities are fitted with inlet and outlet sections designed for the liquid circulation;
- the fluid is kept subcooled (i.e. the fluid temperature is always smaller than its boiling point temperature);
- the duct opposite wall to the product is considered adiabatic, it is called the cold wall.

This model postulates that:

- there exists a vapor film between the hot surface and the liquid;

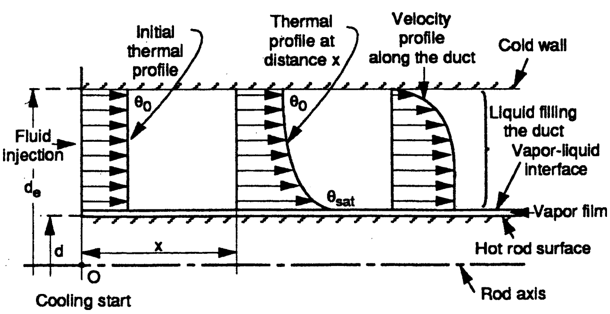

Figure 1. Schematic disposition for cylindrical geometry with thermal and velocity profiles.

- the temperature of the liquid–vapor boundary is the liquid boiling temperature;
- the flow inside the duct is turbulent due to shear stresses along the cold duct wall;
- there is no shearing between the vapor and the liquid at the corresponding interface;
- the product speed has no influence on the heat transfer rate;
- the vapor film thickness is such that the heat flux by conduction and radiation across itself is equal to the turbulent diffusion heat flux inside the liquid; there is no continuous vaporisation (the vapor film generation has a negligible heat transfer contribution);
- then, the heat flux density is not a function of the hot product temperature in this film boiling regime, it is only dependent on heat transfer inside the liquid;
- the cooling liquid is considered quasi-homogeneous with an equivalent or turbulent thermal conductivity equal to $\lambda_t = \lambda\alpha/k$. α is the turbulent convective heat transfer coefficient within the liquid inside the duct, and k is the liquid thermal molecular conductance per unit of the hot surface, calculated from the solid conduction formula for the radial heat transfer through a cylinder, applied for the liquid cylinder around the rod. k (equation (1)) is a function of the duct geometry;
- the liquid, entering the duct at a uniform temperature θ_o, experiences a temperature step at the vapor interface equal to $\Delta\theta = \theta_{sat} - \theta_o$;
- heat is transferred within the liquid only by conduction from the vapour interface towards the cold wall, using the concept of turbulent conduction;
- the semi-infinite medium model of conduction is applicable to the liquid as long as the residence time within the duct is short (indicative value less than 1 s), and the liquid layer is thick enough.

These hypotheses are schematically visualized in *figure 1.*

2.2. Mathematical model

We indicatively give the mathematical expression for the cylindrical duct and product, geometry. The corresponding plane geometry formulas are given in [6]:

- thermal molecular conductance of the liquid annulus, per unit of rod surface:

$$k = \frac{2\lambda}{d \ln(d_e/d)} \tag{1}$$

- hydraulic diameter for Reynolds number:

$$d_H = (d_e - d)\left(1 + \frac{d}{d_e}\right) \tag{2}$$

- turbulent thermal conductivity:

$$\lambda_t = \alpha \frac{d}{2} \ln \frac{d_e}{d} \tag{3}$$

- turbulent thermal diffusivity:

$$a_t = \frac{\lambda_t}{\rho Cp} \tag{4}$$

- Stanton number definition and expression for convective heat transfer (Kutateladze [7]):

$$St = \frac{\alpha}{\rho V Cp} = \frac{0.023 Re^{-0.2}}{1 + 2.14 Re^{-0.1}(Pr^{2/3} - 1)} \tag{5}$$

- dimensionless time:

$$t^* = \frac{4a_t}{d^2} \frac{x}{V} \tag{6}$$

- dimensionless heat flux density definition and expression at a distance x:

$$\phi_x^* = \frac{d\phi_x}{2\lambda_t(\theta_{sat} - \theta_o)} \approx (\pi t^*)^{-0.5} + 0.5 - 0.25\left(\frac{t^*}{\pi}\right)^{0.5} + \frac{t^*}{8} \tag{7}$$

The expression of the dimensionless heat flux density is the asymptotic form valid for $t^* < 1$ from Carlsaw and Jaeger [8], for heat transfer by conduction inside a semi-infinite medium, with a temperature step on the inside cylindrical boundary. The local heat flux ϕ_x along the liquid stream is then proportional to $(\theta_{sat} - \theta_o)$, and dependent on the flow via λ_t and the convective heat transfer coefficient α.

2.3. Experimental results

Experiments have been performed with several geometries (plane and cylindrical), and several liquids (water, oil and water with additives). Local heat fluxes were measured by the transient method, an easy method when dealing with high fluxes. Experiments for the cylindrical geometry were conducted with a hot rod of diameter 32 mm and cooling tubes of diameters 35.4 and 50 mm, the fluid velocity varying from 0.7 to 11 m·s^{-1}. Experimental dimensionless heat fluxes are compared with the

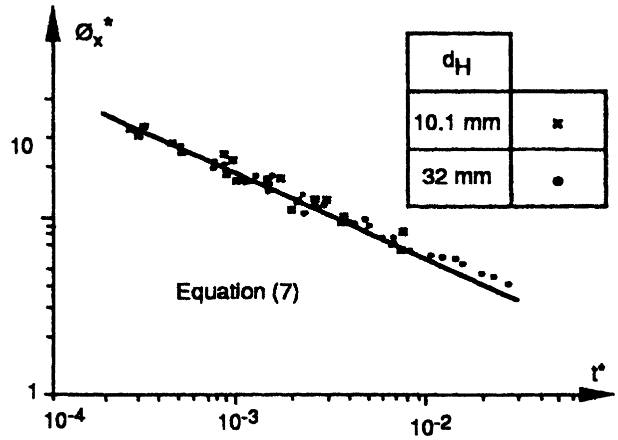

Figure 2. Cylindrical duct. Variations of ϕ_x^* as a function of t^*. Comparison between experiments and model.

mathematical expression in *figure 2* for cylindrical geometry. There is a very good agreement between the model and the experience. No adjustment of the Stanton convective correlation was necessary.

This means that the phenomenological approach is valid. Then, this approach has been extended for the analysis of cooling with sprayers (see the following section).

One important feature issued from these results is that the heat flux is rapidly decreasing along the flow, starting from the injection point, the rough relation being like $x^{-0.5}$. As a result, along noticeable distances from the injection, high heat fluxes occur up to 10 MW·m^{-2} and even above. This heat flux can be modulated through the liquid velocity, the liquid thermophysical properties and its temperature.

3. FILM BOILING WITH SPRAYED SURFACES

3.1. Intoduction to linear pneumatic sprayers

The previous model has been worked out for confined liquid flow through ducts. It has been proved very useful to design numerous industrial cooling machines for which high heat fluxes were required but with a moderate range of variation. For such machines it is not easy either to modify the duct geometry or to run the flow rate within a large range.

For other very important industrial applications, it is necessary to cool very large plane surfaces (several meters) very homogeneously, and at an easily adjustable high cooling rate, that is with high heat fluxes.

It has appeared that for such requirements, linear air-plus-water sprayers with constant air flow rate and variable water flow rate gave the best compromise, for a fine adjustment of heat fluxes and products with various sizes.

3.2. Water flow structure influence on the spatial heat flux distribution, with linear sprayers

When looking at the spatial heat fluxes distribution one can see that it is very sensitive to:

- the droplet jet orientation versus the orientation of the surface to be cooled and the gravity direction;
- the way the water leaves the surface after impingement.

This is well illustrated through heat fluxes measurements performed with three arrangements similar to typical industrial situations.

For the following three cases the nozzle plane of symmetry is vertical and the surface to be cooled is horizontal:

Case 1. The jet impinges on the top of the plate. The cooled top surface of the product being not too large (equivalent to the transversal jet impact), water leaves the plate transversely at a velocity comparable to the droplet velocity and making a rather thin coat of water (few millimetres). This case represents the cooling of small objects, like aluminium extruded sections.

Case 2. The jet impinges on the bottom of the plate. Like the previous case, water leaves the plate transversely in a thin coat that is maintained against the plate by the impinging water. Out of the impingement, water falls down leaving the plate. This case represents cooling from the bottom of objects of any size.

Case 3. Like for case 1, the jet impinges on the top of the plate, not directly onto the metal surface but rather on a thick coat of water (several centimetres). This case represents large surfaces cooling (above 10 m^2). For example: to cool heavy steel plate (5 m by 20 m, or 2.5 m by 40 m), twenty parallel sprayers (5 m long) are displayed at a 1 m pitch, along the cooling tunnel (20 m long), inside which the plate travels transversely to the sprayers. For such a disposition the total flow rate to be evacuated is large and has to exit laterally from the surface along the edges of the plate, under gravity forces.

Figures 3 and *4* show the above-described experimental dispositions. For these experiments the linear nozzle length was comparable to the transverse impact size, around 1 m, the spray shape being then a dihedron the axis of which is the nozzle slot. This kind of linear pneumatic sprayer has been developed by Bertin Technologies with low air pressure and constant air flow rate, primarily to cool heavy steel plates behind mills. They are conceived for generating a uniform pattern of water along the slot (longitudinal axis). It has been shown that the corresponding cooling was uniform along this axis. The transversal axis is perpendicular to the plane of symmetry of the slot (and of the spray).

A special arrangement has been designed to reach a comparable hydraulic behaviour to the one of case 3, using only one sprayer with a shorter length than the one for the industrial application. Water is confined in a box, the bottom of which is the hot plate. The height of the vertical walls parallel to the sprayer are higher than the expected layer thickness and lateral walls height are adapted to adjust the predicted height of water onto the plate for the given application (see *figure 4*). The water leaves the plate in that situation above these lateral walls, not shown on *figures 3* and *4*. The wanted water height, for example, to simulate a plate 5 m wide, for a high mean mass flux of 15 $kg{\cdot}s^{-1}{\cdot}m^{-2}$, is 78 mm given by the weir formula found from Comolet [9]:

$$q = \left(\frac{2}{3}H\right)^{3/2}\sqrt{g}$$

q is the lateral flow rate on each side of the plate per lateral length unit, as a function of the water height H.

The hydraulic arrangement is completed with a masking device that permits one to display the instrumented hot plate below the spray, and wait till the hydraulic regime is steady before the cooling starts. The plate is of copper or aluminium with thickness around 10 mm, in such a way that its transversal conduction time constant is small compared with the cooling time. The thermocouples equipped the opposite side of the sprayed side. The local heat flux, for a given plate thickness e, is deduced from the plate cooling rate C_R, measured with the imbedded thermocouples:

$$\phi_x = \rho e C p C_R$$

ρ and Cp are the plate medium properties.

Figure 3 gives the transversal distribution of the mass flux locally impacting on the plate and the local resulting cooling heat flux. The comparison between flow rate and heat flux distributions shows that the cases 1 and 2 are

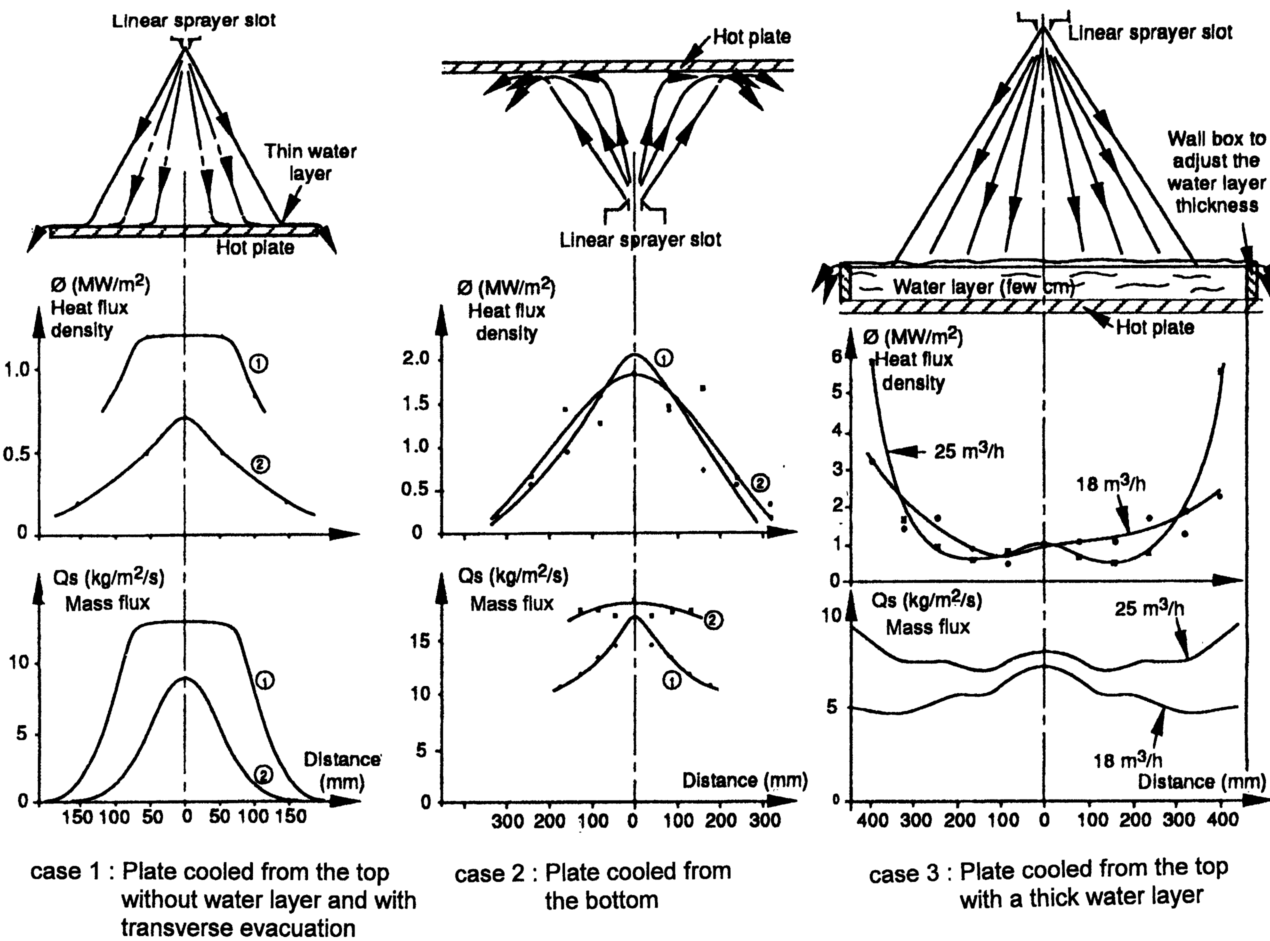

Figure 3. Influence of spray disposition on cooling heat flux.

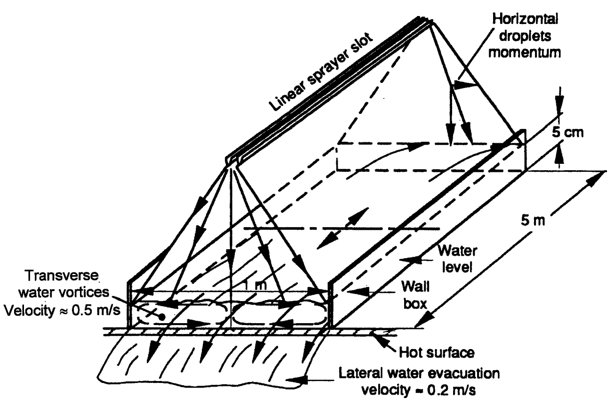

Figure 4. Scheme of the water vortices induced by the droplets' momentum.

very similar with a corresponding evolution of the two distributions. For case 3 the two distributions are rather in antiphase, while the mean heat flux variation versus the total flow rate stays comparable with cases 1 and 2. When looking at mean values calculated along the total transverse sprayed surface it is found that for the three cases, the variation of the mean heat flux versus the mean mass flux is expressed like

$$\phi_{\mathrm{m}} = K Q_{\mathrm{sm}}^{0.7}$$

K being a constant that can change with the disposition.

The peculiar behaviour seen for case 3 is explained through the analysis of the flow pattern on the top plate. At the exit of the nozzle the droplets have a horizontal component of velocity parallel to the plate, the corresponding momentum is transferred to the top of the water layer at the impact. This creates flow recircula-

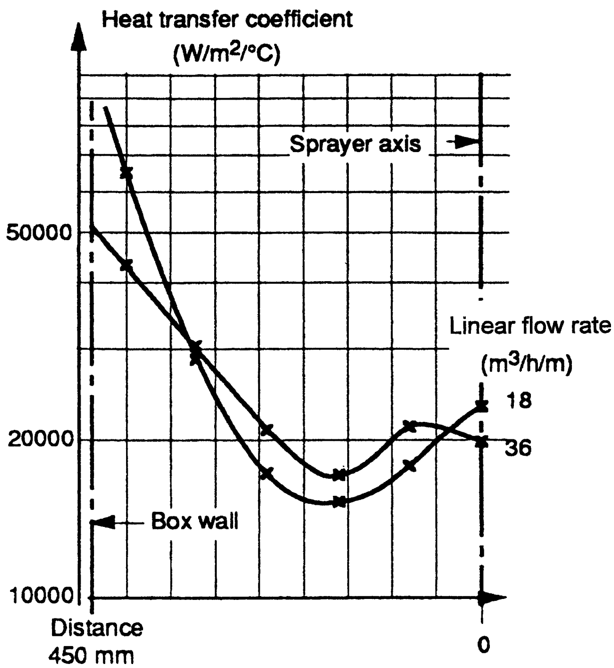

Figure 5. Convective heat transfer coefficient inside the water vortex.

tions, shown in *figure 4*. Below one sprayer, two vortices are generated with reverse rotation. On top of the layer, flow goes from the plane of symmetry towards the edges of impact, the reverse occurring on the bottom of the layer that is near the plate surface. These vortices have been evidenced experimentally (with coloured fluid injection), with the disposition shown *figure 4*. On industrial application they occur when several sprayers are displayed contiguously, like described in case 3 for cooling large heavy steel plate. The measured transversal velocity, around 0.5 $\mathrm{m \cdot s^{-1}}$, for typical conditions, is noticeably higher than the lateral velocity: 0.2 $\mathrm{m \cdot s^{-1}}$ maximum value at the edge of the plate.

Applying the same heat transfer model as the one described in section 2, we consider the vortex flow just above the plate. Inside this layer a thermal diffusion starts at the transverse edges and develops along the flow towards the plane of symmetry. This mechanism explains the decrease of the heat flux from the edges towards the plane of symmetry. The corresponding heat transfer coefficient h is given in *figure 5* calculated from

$$h_x = \frac{\phi_x}{\theta_{\mathrm{sat}} - \theta_x}$$

θ_x is the local bulk water temperature. This heat transfer coefficient is rather insensitive to the flow rate variation. This means that the water transverse velocity in the layer just above the plate is roughly constant, due to the fact that the total droplet momentum is constant for this kind of sprayer with constant air flow rate whatever the liquid flow rate is.

The good transverse (relative to the plate) cooling homogeneity is obtained for cases 1 and 2, with the specific design that guarantees a uniform air and water distribution along the slot. In addition for case 3, the vortex phenomena neutralise the effects of water lateral exit that could create transverse nonhomogeneous cooling.

4. REWETTING: TRANSITION BETWEEN FILM REGIME AND WETTED REGIME

One of the key requirements for cooling devices is to reach a precise final temperature. The rewetting phenomena, that is the transition between film boiling regime and nucleated boiling regime, is a rapid one. One important feature is the sudden enhancement of heat flux (three or four times larger), that occurs at a given temperature (rewetting temperature) dependent on the metal.

The so-called rewetting temperature in what follows is the superficial temperature of the object being cooled, that is the temperature of the oxidized superficial layer if any. It is around 500 °C for steel and around 250 °C for light alloys and metals that are difficult to oxidize. The appreciation of this temperature is usually subject to errors due to the thermal gradient through the superficial oxide layer. But these errors are not so important compared with the importance of the following phenomena. Moreaux et al. [10] showed that the rewetting temperature was highly depending on the effusivity of the superficial layer of the cooled surface. This physical parameter ($(\lambda \rho Cp)^{0.5}$) governs the thermal exchange when putting into contact the surfaces of two bodies at uniform but different temperatures.

At the very beginning of the contact, the common contact temperature θ_{c} for the two bodies is given by Carslaw and Jaeger [8]:

$$\frac{\theta_1 - \theta_{\mathrm{c}}}{\theta_{\mathrm{c}} - \theta_2} = \left(\frac{\lambda_2 \rho_2 Cp_2}{\lambda_1 \rho_1 Cp_1}\right)^{0.5} \tag{8}$$

This equation expresses the equality of the flux exchanged at the contact from one body to the second one, the contact temperature being unique. This shows that the contact temperature, that is the temperature leading to the rewetting, cannot be directly measured with classical means and is always lower than the measured mean metal temperature. θ_c stays steady up to the time during when the two bodies can be considered semi-infinite from the thermal conduction point of view. Practically the body with the highest effusivity will impose its temperature as contact temperature. Oxides having generally a lower thermal conductivity and then a lower effusivity than that of metals, even superficial oxidation will noticeably increase the rewetting temperature.

Calculating with equation (8) the contact temperatures and using the rewetting temperatures measured by Moreaux et al. [10] and by Choi and Yao [11], we found values between 180 and 200 °C. This corresponds to rather high superheating temperatures for liquid water at atmospheric pressure, that is physically conceivable, according to the following observations. Very small drops of water dispersed in an oil bath can stay liquid for a very long time at temperature above 160 °C, in the absence of nucleation site inside the droplets (experiment of one of the authors). This is also currently observed in regulated temperature bath using thermal fluids, and also when frying potatoes in oil.

Tima et al. [12] have experimentally shown that on a wall heated at a very high heating rate the first bubbles appear (starting of the vaporization), at a wall temperature that depends on the rate of its variation and then on the instantaneous and local rate of heating. The maximum values are not far from the spinodal temperature, also called self-vaporization temperature. This spinodal temperature is the highest superheating temperature that a liquid can undergo in the absence of any nucleation seed. It is given by Carey [2] using Van der Waals gas state equation.

Then, the rewetting phenomena starts when the contact temperature between the cooling liquid and the solid being cooled is just below some temperature θ_{sup} below the spinodal temperature that is around 200 °C for water. The contact temperature is calculated with equation (8) and the thermophysical properties of the liquid for one body and of the superficial layer of the solid for the second body. Above this temperature even a very fugitive contact of liquid is not possible and the vapor film is steady. Below this temperature a liquid contact can last and progressively the vapor film is disappearing.

Figure 6 shows the rewetting situation and the corresponding profiles of temperature in the solid and the liquid at the instant of the transition. This shows that the superficial solid temperature at which rewetting will occur depends on:

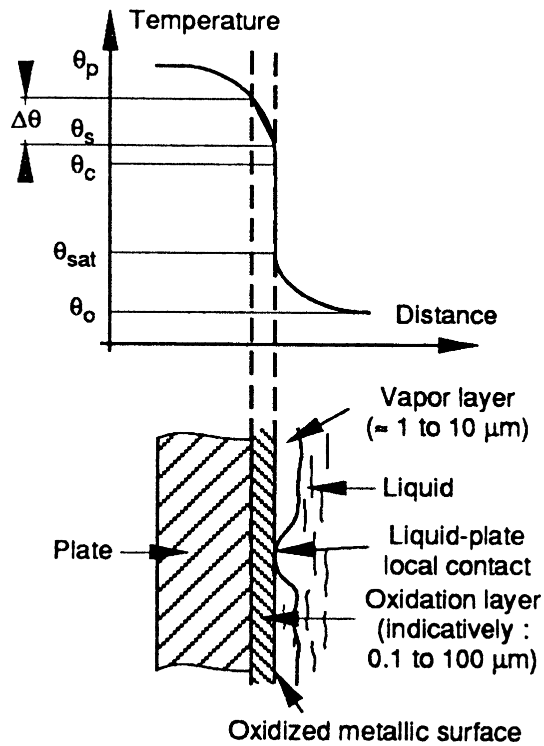

Figure 6. Temperature profile at the starting of the liquid-plate contact (rewetting).

- the heat flux that the liquid is able to dissipate, it is a function of the local flow structure and the water subcooling;
- the local liquid saturation temperature, that is usually the boiling temperature at atmospheric pressure;
- the liquid effusivity;
- the maximum instantaneous temperature of the liquid in superheated state, depending on the subcooling of the liquid and of its cleanliness in term of nucleation seeds;
- the effusivity of the superficial layer of the solid. This layer is predominant rather than the bulk metal for the contact temperature phenomena and then the occurrence of the rewetting.

The superficial temperature θ_s that gives a contact temperature θ_c equal to θ_{rew} is the true rewetting temperature. This temperature is not directly measurable. Practically one measures a mean bulk metal temperature that differs from the superficial one by:

- a temperature drop inside the metal depending on the cooling heat flux, the thermal conductivity and the thickness of the body;

• a temperature drop through the superficial oxide layer depending also on the cooling heat flux, the thermal conductivity and the thickness of the layer.

5. INDUSTRIAL APPLICATIONS

Bertin Technologies has designed two kinds of cooling machines.

The first one cools cylindrical rods (diameter from 10 to 35 mm), of light aluminium alloys. This rod is mainly used for wire drawing in the manufacture of electrical cables. This machine is simple, the rod is running horizontally, then, it comprises a horizontal cooling tube with an injection device at rod entry and a water diversion device at the rod exit. These two devices have to insure the tightness. The rod and water velocities are rather high (5–20 $m \cdot s^{-1}$). Local heat fluxes are above 10 $MW \cdot m^{-2}$. Numerous machines are used by aluminium transformers.

The second one cools high quality heavy steel plates (1–10 cm thick), used for pipe lines and boat hulls. These machines comprise linear pneumatic sprayers with low pressure air (around 60 mbar), with lengths of up to 5 m, corresponding to the plate width, placed in parallel above and below the plate line pass, transversely to the plate displacement. The water flow rate is individually controlled for each sprayer along the machine as a function of the predicted superficial plate temperature, in such a way as to control the heat flux along the cooling (from 0.3 to 1.5 $MW \cdot m^{-2}$ in film boiling regime). A well adapted thermal control model makes these machines a very flexible cooling tool. Four such machines are running in the USA, Korea, China and France.

Bertin Technologies has also adapted the linear pneumatic sprayer to extrusion cooling under the registered trade mark SPRAYLINE, with a standard length of 300 mm. This sprayer is currently used to build cooling tunnel, some of them with sprayers displayed on the four internal faces.

6. CONCLUSION

We have shown that for film boiling with subcooled forced flow, the heat transfer through a vapor liquid interface subjected to moderate shear stresses is mainly depending upon the contact time duration with this interface, the liquid turbulent diffusivity, and the liquid temperature.

Concerning film boiling with sprayed water on hot plates we have shown that:

• The heat flux spatial distribution is tightly connected to the flow pattern. This pattern depends primarily upon the overall concept of the water injection, and the size of the installation.

• Cold experiments (i.e. hydraulic experiments) will have to recreate as precisely as possible the flow structure expected on industrial situation in such a way as to foresee the heat flux distribution with hot plates.

• When cooling from the bottom in film boiling situation, the liquid stays in the vicinity of the plate only, under the impacting water or, in some places, due to stagnancy caused, for example, by roller supporting the plate. Where water is not stagnant, the local heat flux is only depending on the local mass flux. In the stagnant zone, the structure of the resulting flow and the total flow governs the local heat flux.

• It is not possible to reach spatially uniform heat flux when cooling large surfaces. At the very most, uniformity along one axis is possible, like the one obtained with the linear sprayers developed by Bertin Technologies. Uniform cooling is then obtained running the plate through the cooler to give every point of the surface the same cooling "story" assuming that the cooling is homogeneous transversally to the speed direction.

Rewetting has to be considered very carefully for industrial cooling installations when the final temperature is precisely required. A precise final temperature and a constant cooling rate cannot be reached by just acting globally on the liquid flow rate, due to the enhancement of heat flux at rewetting. A flow rate control has to start at the beginning of rewetting and for an efficient such control one needs an exact rewetting model. The "true" rewetting temperature will depend on the following parameters:

• on metal side: initial heat flux in film boiling regime, kind and thickness of metal, kind and thickness of the oxide layer;

• at solid–liquid interface: state of cleanliness of the metal surface; kind, cleanliness and temperature of the liquid;

• on liquid side: initial heat flux in film boiling regime highly connected to the kind, the temperature and the flow structure of the liquid.

The rewetting model will have to consider all these parameters. The flow rate control will be either spatial or temporal whether the cooling process is under running the product or static.

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