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Heat transfer from a hot moving cylinder impinged by a planar subcooled water jet

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ABSTRACT

A hot moving (rotating) cylinder was heated up to 500–600 °C and then was cooled by a planar water jet impinging on a line parallel to the symmetry axis. The time dependent wall temperature was measured using embedded thermocouples and the corresponding wall heat fluxes were estimated through an inverse conduction method. In a recent paper, we showed that cooling rates depend on the subcooled temperature of the jet, the velocity of the jet and the surface-to-jet velocity ratio. Since the initial temperature of the cylinder was higher than the Leidenfrost temperature, we observed all the boiling regimes from film boiling to nucleate boiling. The objectives of this paper are firstly to describe the current conditions which exist in the Run Out Table in hot rolling mills, secondly to review the main experimental studies dedicated to jet cooling which have led to modelling heat transfer in boiling conditions and finally to propose new correlations taking into account the velocity of the wall.

1. Introduction

In the steel industry, cooling on the Run Out Table (ROT) after hot rolling is one of the most difficult process steps in the hot mill strip. The decrease in the temperature needs to be perfectly controlled because the mechanical properties of steel alloys are conditioned by the cooling rate ensured by these jets [1–4]. Generally, top cooling is carried out using a number of subcooled water jets which impinge perpendicularly on the hot steel surface while bottom cooling is done using sprays. The water jets are organized in a set called a header where two jets rows are either aligned or staggered. Complex flows are thus obtained, as a result of the interaction between the jets and the moving surface. In-depth knowledge on the heat transfer associated to that flow is therefore essential including knowledge of the interaction between the jet and the moving surface, the interaction between the jets and the interaction between the ramps.

Rates of cooling will vary between 15 and 1000 K/s depending on the required steel mechanical property. Although cooling technologies based on the so-called laminar water jets have been widely studied in the past, knowledge of these technologies remains incomplete which means it is difficult to attain optimum production. Obviously the kinetics of cooling depends on the various boiling regimes met during transient cooling (i.e. the metal slab is reheated before rolling, the temperature of the strip after rolling is about 900 °C and after cooling, the temperature should be between 200 and 500 °C, depending on steel grade). At the very beginning of the cooling phase, the temperature of the steel strip is above the Leidenfrost temperature so film boiling, transition boiling, critical heat flux (CHF) and nucleate boiling all occur. Controlling the cooling rate and the homogeneity thereof thus remains a major challenge for manufacturers aiming to produce steels with desired and homogeneous mechanical properties.

1.1. Description of the flow on the Run Out Table (ROT)

As previously described, the heat transfer can not be homogeneous when using water jets in the cooling system because of flow topology and because the water film depth above the hot surface is not constant. This depends on the distance from the impact zone of the jet but also on the ratio between velocity of the jet and the velocity of the moving surface. In a recent paper, Gradeck et al. [5] carried out experimental and numerical studies of the flow structure of a single water jet impinging on a moving surface. This work provided a valuable correlation to predict the position of the hydraulic water jump for operating conditions similar to those in ROT cooling systems. In a more recent paper, the flow pattern of multiple water jet impinging on a moving surface was numerically studied by Cho et al. [6] using the CFD Fluent package. Their computations clearly showed how flow patterns are dependant on the running conditions (flow rates, velocity of the surface). At low flow rates, hydraulic jumps were observed while with increasing values of the flow rates, the hydraulic jumps disappeared and a pool was observed. Moreover, a fountain effect was found to occur at times between two adjacent jets with consequent improvement of the

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Nomenclature

α	heat transfer coefficient ($W\,m^{-2}\,K^{-1}$)	r^*	dimensionless velocity, $\frac{V_S}{V_J}$
d	nozzle diameter (m)	x	distance from the jet axis (m)
q''	heat flux ($W\,m^{-2}$)	x^*	dimensionless distance, $\frac{x}{d}$
Re	Reynolds number		
Pr	Prandtl number	*Subscripts*	
ΔT_{sub}	subcooled temperature (K)	FB	film boiling
ΔT_{sat}	superheat temperature (K)	MFB	minimum of film boiling
T	temperature (°C)	TB	transition boiling
V_J	jet velocity at the impingement ($m\,s^{-1}$)	NB	nucleate boiling
V_n	jet velocity at the exit of the nozzle ($m\,s^{-1}$)	TC	transient conduction
V_s	wall velocity ($m\,s^{-1}$)	CHF	critical heat flux
λ	conductivity ($W\,m^{-1}\,K^{-1}$)	L	liquid
μ	dynamic viscosity (Pa s)	V	vapour
σ	surface tension ($N\,m^{-1}$)		

cooling flux. Monde et al. [7] gave an illustration of the interactions of neighbouring jets and the expected increase of the heat flux in this area. However in more recent study, Franco [8] did not find a fountain effect to occur whether the alignment of the jets was staggered or aligned. The large distance between jets (around 100 mm) in Franco's [8] experiments is the probable explanation.

1.2. Heat transfer associated with single impinging jet

Most studies of heat transfer associated with impinging jets have been carried out using a static surface which means that the dynamic conditions as mentioned above may have been far from industrial conditions [9–15]. However, most correlations from these studies are used as command and control for the ROT even though the influence of surface velocity was not addressed in these studies. Depending on the strip temperature, four different water cooling regimes may be encountered: (i) for high temperatures, only film boiling; (ii) transition boiling if the surface temperature of the strip is lower than the Leidenfrost temperature; (iii) nucleate boiling regime if the surface temperature is lower than the CHF temperature and finally; (iv) forced convection (Fig. 1(a)). A particular shape of boiling curve may be observed in the transition regime in the case of an impinging jet on a hot plate (Fig. 1(b)). After CHF, a first minimum of flux was observed after which the heat flux increased again and reached a high value. This "shoulder of flux" was found to falls down abruptly to film boiling regime for very high superheats. Miyasaka et al. [15], Ishigai et al. [9], Ochi et al. [10] and Hall [11], Robidou et al. [13] and Gradeck et al. [16] have reported the existence of a "shoulder of flux" beneath the jet (transition boiling regime) in the case of static surface. But Gradeck et al. [16] showed that heat transfer is radically modified when the impingement surface is moving. They found that the "shoulder of flux" beneath the jet axis collapsed and that the local boiling curves had the same shape upstream, downstream and beneath the centreline of the jet.

1.3. ROT heat transfer model

Given that cooling rates (i.e., heat transfer at the wall) are affected by the temperature of the strip, dedicated correlation needs be applied to the off-line or on-line models used in ROT in order to correctly predict the coiling temperature of the strip. Since the heat transfer model used in the ROT control is usually simple, consisting of a simple polynomial relationship [17–19], improvements should be done to consider four heat transfer regimes (Fig. 1) and two significant temperatures (Leidenfrost and CHF temperatures). In the following sub-sections, we give an overview of the correlations (or models) available for impinging jets in these four boiling regimes.

1.3.1. Film boiling

1.3.1.1. Stagnation zone. Zumbrunnen et al. [20,21] and Filipovic [22] obtained models for the film boiling regime by solving the boundary layer equations but as some strong assumptions have been assumed to simplify the Navier–Stokes equations and obtain an analytical solution, these models can be very far from the current application. For example neglecting waviness on the vapour–liquid interface as well as Kelvin–Helmotz instabilities lead to delay the minimum of film boiling (in comparison with measurements).

Liu and Wang [23,24] derived an analytical expression of the heat flux for the film boiling regime in the stagnation zone ($C = 1.414$). To fit the experiments, the constant C of the original expression was slightly modified to take into account the interfacial waviness of the vapor layer ($C = 2$):

$$Q''_{FB} = C\mathrm{Re}_J^{0.5} Pr_J^{0166} (\lambda_L \lambda_V \Delta T_{sub} \Delta T_{sat})^{0.5} \Big/ d, \tag{1}$$

where the thermal properties of water were evaluated at the film temperature of water and vapor properties evaluated at the film temperature of vapour.

Ochi et al. [10] proposed the following correlation

$$q''_{FB} = 3.18x10^5 (1 + 0.383\Delta T_{sub})(V_J/d)^{0.828} \tag{2}$$

A similar expression was found by Ishigai et al. [9] for a planar jet:

$$q''_{FB} = 5.4x10^4 (1 + 0.527\Delta T_{sub}) V_J^{0.607}. \tag{3}$$

For all of these correlations V is expressed in $m\,s^{-1}$, d is in mm and ΔT in K.

Robidou [25] used an innovative experimental device to measure boiling curves under steady state conditions and proposed the following correlation at the stagnation zone:

$$q''_{FB} = 5.38x10^4 (5.5 + \Delta T_{sub}) V_J^{0.6} \tag{4}$$

1.3.1.2. Parallel flow zone. Few models exist in the parallel flow zone ($x^* > 2$). Filipovic [22] developed a model for this regime based on the boundary layers equations and measurements achieved on a parallel jet. Hatta et al. [26] proposed the following correlation fitting the experimental data of Kokado et al. [27] obtained for a transient cooling:

$$\alpha = 200(2420 - 21.7T_L)\Delta T_{sat}^{-0.8}. \tag{5}$$