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Evaluation of heat transfer in quenching processes with impinging liquid jets

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ABSTRACT

Liquid jets and sprays often are used in metal-working processes like hot rolling, heat treatment or casting for intensive cooling of hot metal specimen and surfaces. Due to the potential evaporation of the quenching liquid during boiling, the local fluid flow as well as the heat transfer are extremely dynamic processes. Therefore, a numerical simulation model has been developed for flow and heat transfer to analyze the entire cooling process in the jet quenching process including all boiling regimes.

In this contribution the focus is on quenching of hot metal surfaces with impinging water jets. The impingement liquid jet flow behavior and the wetting front behavior of the jet on the surface of a plate have been analyzed. As result of the simulations, the local surface temperatures and the temperature gradients at the cooled surface are evaluated as well as the local heat transfer rates.

The model validation is done by means of different experimental approaches. Therefore, the progression of the wetting front at the quenched surface and the temperature distribution on the back side of thin metal plates during jet quenching has been evaluated.

1. Introduction

In heat treatment or metal forming processes for metallic semi-finished products and structural parts, water jets or sprays are often used for an intensive and controlled quenching [1]. In this processes temperatures of the metal surface to be cooled are well above 1000 °C for steel alloys or 600 °C for aluminum alloys. By rapid quenching of the material it is intended to obtain the required material and component properties for practical use. An important requirement for the cooling process is the controlled setting of a spatially and temporally defined course of the heat transfer or heat flow rate in order to avoid temperature inhomogeneities in the component. These may reduce the component quality by residual stresses or inhomogeneous material properties [2]. The heat transfer mechanisms while quenching with liquid jets are to be seen in analogy to different flow boiling phenomena that subsequently occur during quenching, as have been summarized in Ref. [3]. Due to the high surface temperatures, flow boiling takes place in different boiling phases as for instance film boiling above the Leidenfrost temperature. Below the Leidenfrost temperature wetting of the surface takes place and leads to transition boiling where the heat transfer increases rapidly. The heat transfer rate reaches its maximum at the critical heat transfer rate when complete nucleate boiling occurs. When the surface temperature further decreases below the boiling point, just forced convection occurs [4]. Due to the density difference of liquid water and vapor the hydrodynamics of the surface water film is disrupted. The growing bubbles and its sudden expansion leads the liquid film to splash off from the surface. The mechanisms behind this dynamic heat and fluid flow process are not well understood. Conventional experiments to characterize the heat transfer are not able to derive the surface temperature directly. The surface temperature is approximated by thermocouples mounted close to the surface or by approximation from the back side surface temperature of thin plates during quenching (inverse heat transfer). Therefore a heat and fluid flow model is developed to analyze the entire process and derive significant information like the temperature of the wetted surface and the heat transfer rate from the surface. Simulations of the quenching process of metal plates by impinging liquid water jets obtain high temperature gradients inside the material in normal direction to the surface. Therefore, the systematic error in such experiments with inverse back-calculation of the surface temperature, as in Karwa [5], can be evaluated.

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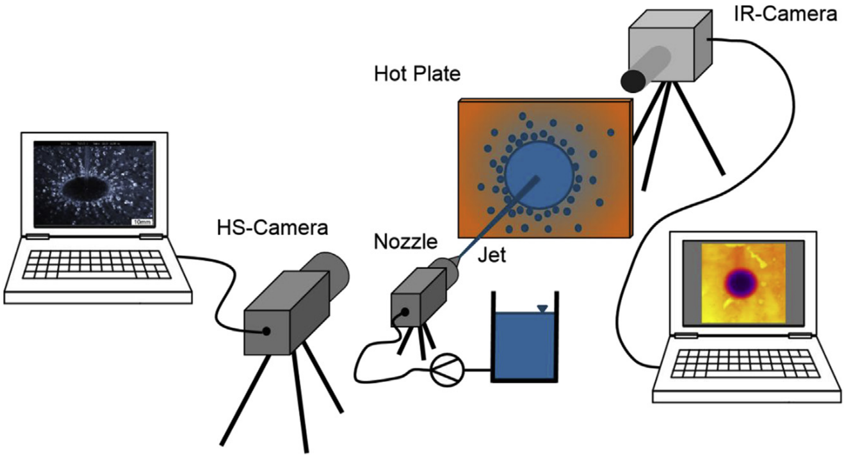

Fig. 1. Schematic representation of the experimental setup.

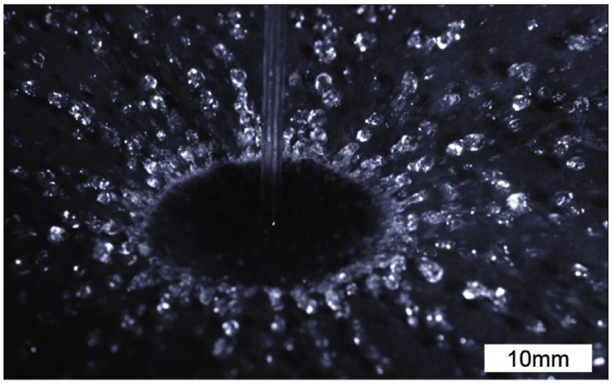

Fig. 2. Image of an impinging water jet (d = 1.3 mm) on a hot surface at $T_{surf} > T_{Leidenfrost}$.

Table 1
Sink and source terms in conservation equations for the evaporation case.

Evaporation $\dot{m} < 0$	Liquid	Vapor
Energy	$S_{e,1} = -\alpha_1 \rho_1 c_{P,1} \frac{(T_1 - T_{sat})}{\Delta t}$	$S_{e,2} = -\dot{m}\, c_{P,1}(\mathrm{T_{sat}} - \mathrm{T_2})$
Mass	$S_{mass,1} = \frac{S_{e,1}}{\Delta h_v} = \dot{m}$	$S_{mass,2} = -S_{mass,1} = -\dot{m}$
Momentum	$S_{mom,1} = \dot{m}\cdot(u_i)$	$S_{mom,2} = -\dot{m}\cdot(u_i)$

2. Experimental setup

In the experimental investigation hot plates have been quenched from high temperatures by an impinging jet. The spreading of the wetting front during jet quenching of the hot plate is tracked by recordings of a high-speed camera. Local temperature measurements during quenching of the plate have been done at the backside of the thin plates (2 mm) by an infrared camera. The determination of the surface temperature directly beneath the impinging jet on the front side of the plate is not feasible in the present investigation. The experimental setup is illustrated in Fig. 1.

Stainless steel plates (200 × 200 × 2 mm) have been heated in an oven up to a homogeneous initial temperature (in the range between 500 and 850 °C). The backside of the plate has been coated to receive a high and constant emission coefficient (approximately 1) necessary for the infra-red recordings. The emissivity coating is a high temperature resistant silicone resin based specific paint.

For the IR camera measurements a calibration experiment has been performed. The temperature profile of a thermocouple (TC) of a 10 mm thick stainless steel plate is recorded 1 mm below the surface of. In addition, the surface temperature was recorded with the IR camera. The plate temperature has been hold and at constant temperature the deviation between both recordings is ± 1 K. During plate cooling, due to the depth of the TC, a temperature difference of 11 K from TC to the IR camera was observed for cooling in air from 500 °C. The temperature measurement of the IR camera has a coefficient of determination of 99.21.

The circular jet is adjusted at a constant water mass flow rate. The nozzle diameter is varied between 0.8 and 1.3 mm. Water outlet velocities have been varied from 10 to 30 m/s. The water temperature is set to 22 °C. A distance of 15 cm between the nozzle and the plate is necessary to allow a manual insertion of the hot plate in the quenching fixture. Before initialization of the jet impingement, a splash guard system is used in order to avoid start-up irregularities of the water jet. The system is taken out of the jet flow immediately before the quenching process starts, so that the fully formed water jet hits the hot plate.

Fig. 2 shows a camera snapshot of the impinging jet on the hot surface. In the area close to the impingement/stagnation point, the surface is wetted and a radial film flow evolves. The bright area indicates the nucleate and transition boiling regimes. Here, due to the momentum caused by the liquid evaporation process, the water film lifts off from the surface, disrupts and forms droplets. These droplets slide outwards without further wetting the surface and practically without any cooling effect of the surface. From these visual observations the wetting front position and its velocity have been determined. The wetting front position and propagation velocity as well as the temperature distribution at the back side of the plate are the main parameters for validation of the numerical model.

3. Model specifications

The numerical heat and fluid flow process model describing the entire quenching is based on a Eulerian multi-phase model [6], that is properly extended by appropriate sink and source terms for the ability to calculate the mass, heat and momentum fluxes during boiling [7–9]. The conservation equations for energy, mass and momentum are given in Eqs. (1)–(3) for two phases i (1 = liq, 2 = vap). Table 1 represents the associated sink and source terms for the two phases (liquid and vapor) in case of evaporation. For condensation, the terms are formed analogous.

$$\frac{\partial}{\partial t}\rho_i + \nabla\cdot(\rho_i u_i) = S_{mass} \tag{1}$$

$$\rho_i \frac{\partial}{\partial t} u_i + \rho_i (\nabla\cdot u_i) u_i = -\nabla p + \nabla\cdot\bar{\bar{\tau}} + \rho_i g + S_{mom} \tag{2}$$