Continuous Casting Secondary Cooling: Laboratory Heat Transfer Measurements and Accuracy Considerations by Comparison with the Real Situation

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In secondary cooling of continuous casting, it is very important to know the cooling heat flux for the actual spray cooling situation with respect to various parameters like the local position, the nozzle types, distances, and the water and air flow rates, to be able to control the cooling conditions precisely. As heat flux measurements on a casting machine are too challenging, experimental laboratory test rigs are designed and used for measurements by different research groups. Therefore, metal probes of different dimensions and materials are heated up to the desired temperature and then exposed to spray nozzles. The heat flux is usually measured by temperature sensors immersed in the probe body, and then determined from the measured temperature using inverse modelling methods. Herein, the differences between the real and laboratory conditions are focused on using a mathematical heat transfer simulation model. The influence of strand surface temperature, nozzle spray water flow conditions, and Leidenfrost effect are pointed out. A procedure to use heat flux data measured on a test rig for cooling control on a real caster despite the different conditions is proposed.

1. Introduction

During the secondary cooling of continuous casting, water or water/air spray nozzles are used to keep the strand surface temperature within the desired range. Controllable spray intensities are used to locally adjust the required cooling intensity. Therefore, it is very important to know the cooling heat flux for the actual spray cooling situation to be able to control the cooling conditions precisely. As neither strand surface temperatures nor cooling heat fluxes can be measured during the casting process, the actual strand surface temperatures are often calculated by a thermal real-time simulation of the strand based on the actual casting and spray cooling conditions.\[1\]

Such a simulation requires knowing the actual cooling heat fluxes as boundary conditions. Figure 1 shows the secondary cooling principle with water sprays: the cooling heat flux is highest in the impingement region of the spray, whereas it is lower in the other regions of the roll gap. During a transition of the strand between two rolls, the strand surface temperature usually increases first because the actual cooling heat flux is lower than the average heat flux. As the surface approaches the center of the spray impingement, the surface temperature decreases until the cooling heat flux becomes lower and the surface temperature starts to increase again because of the heat flux originating from the hot strand core. Generally, the cooling heat flux density for a specific spray nozzle arrangement depends on the position $x$ on the strand surface with respect to the spray impingement position on the strand surface temperature profile $T_s(x)$ and on the nozzle spray water mass flow rate $\dot{m}_w$.

$$\dot{q}(x) = \dot{q}(x, T_s(x), \dot{m}_w) = \alpha(x, T_s(x), \dot{m}_w) \cdot (T_s(x) - T_w) \quad (1)$$

Usually only averaged heat fluxes and spray water densities are considered, as if the water was sprayed homogeneously over the whole strand surface, resulting in a homogeneous heat flux. The averaging makes the real-time temperature simulation simpler and faster as the computational grid can be much coarser. Furthermore, the averaging is some kind of filtering required for the stability of the nozzle flow control to obtain the desired surface temperatures. Therefore, empirical relations based on laboratory measurements for the averaged cooling heat flux as a function of the averaged spray density

$$\bar{q} = \bar{a} \cdot (\bar{T}_s - T_w) \quad (2)$$

$$\bar{a} = c \cdot (\bar{\dot{m}_w})^\beta \quad (3)$$

are often used\[2,3\] where $T_s$ is the average strand surface temperature, $T_w$ is the spray water temperature, $\bar{a}$ is the average and temperature-independent heat transfer coefficient (HTC), $\bar{q}$ is

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the average surface cooling heat flux, \( \dot{m}_{\text{fl}} \), is the averaged water spray density (mass per surface area and time), and \( c \) and \( p \) are coefficients. Equation (3) does not consider the influence of air, as investigated, e.g., in the study by Preuler et al.\(^4\) These kinds of relations are satisfying for a wide range of casting operation parameters.

For a long time, nozzle configurations have been optimized by homogenizing the spray density, assuming that the local heat flux depends only on the local spray water density, and therefore the intended homogeneous cooling heat flux should be achievable by a homogeneous spray pattern (e.g., the study by Schwertfeger (ed.)\(^3\) Chapter 2.3, whereas Figure 3.3.42 shows the contrary). Recent laboratory experiments turned out that especially in the overlapping region of nozzles, a homogeneous spray pattern does not necessarily guarantee a homogeneous cooling heat flux (the study by Schwertfeger (ed.)\(^3\) Figure 3.3.42\(^,\)\(^4,\)\(^5\)). As the accuracy requirements increase continuously and as new enhanced casting technologies (e.g., very high cooling rates for thin-slab casters\(^6\) and very low cooling rates for local surface temperature increase) necessitate an extension of the range of validity, more detailed information on the cooling heat flux is needed.

According to the common understanding of the predominant cooling mechanism in the secondary cooling zone, a vapor film develops between the impinging spray water and the hot strand surface and above the vapor film, a water film arises.\(^1,\)\(^3,\)\(^7,\)\(^8\) Only a small part of the heat transfer from the hot surface contributes to the vaporization of the spray water, whereas the main part of the heat flux penetrates the vapor film and heats up the film water. For continuity reasons, the film water has to drain off. Depending on the strand surface orientation, it can either detach from the strand surface or follow the surface; moreover, it can leave the roll gap through the gap between the rolls or on the lateral ends or be retained by the subjacent roll (Figure 1b). Therefore, strand surface regions with a low or zero spray density can be cooled by draining or retained cooling water (the study by Schwertfeger (ed.)\(^3\) Figure 3.3.39).

A further common understanding of the cooling mechanism is that for usual strand surface temperatures, the HTC is independent of surface temperature.\(^3\) In contrast, recent measurements\(^8,\)\(^9\) show that the higher the spray density, the more a temperature influence becomes effective; especially, the Leidenfrost temperature increases and can reach temperature ranges of usual strand surface temperatures which drastically increase the heat transfer, as shown in Figure 2a.

Apparent, all available data for the relation between heat transfer and spray cooling intensity are based on measurements, whereas theoretical considerations and numerical process simulations seem both too challenging due to the complexity of the cooling mechanism. Therefore, we will focus on laboratory measurement facilities with support from simple mathematical models that shall help to sharpen the image of the cooling process.

As heat flux measurements on a casting machine are too challenging, experimental laboratory test rigs have been designed and used for measurements by different research groups.\(^4,\)\(^5,\)\(^7,\)\(^8,\)\(^10\) In many cases, local heat fluxes at the spray impingement center are measured for varying spray densities up to \(30 \text{ kg m}^{-2} \text{ s}^{-1} \), measured locally at the heat flux measuring position.\(^7,\)\(^8,\)\(^10\) Now these results are often used as a basis for empirical relations between averaged spray density and averaged heat flux. Relations determined for a local heat flux and local spray water density can only be applied for an average heat flux and average spray water density (and vice versa) if the relation is strictly linear (e.g., \( p = 1 \) in Equation (3)). The investigated spray densities in the previously published results are exceeded by about factor 10 in modern thin-slab casters. Figure 2a shows significant changes in the heat transfer measured for high spray densities, which clearly demonstrates that extrapolating relations for the heat transfer for lower spray densities toward higher spray densities will not properly predict the heat transfer.
2. Mathematical Models

2.1. Heat Transfer Model

A simple 1D transient model for the heat transfer in the strand shell (or laboratory probe body used for cooling experiments) is presented in the following. The main purpose is to demonstrate the influence of various effects under real and laboratory conditions as a kind of “what-if” scenario without the claim of exact results. In addition, the model can be used for the fast evaluation of heat flux and surface temperature based on immersed sensor temperature measurements in laboratory setups, as described in the following paragraphs. As a third application, the model can help to compensate unconsidered effects in a laboratory setup or differences between the real and the laboratory cooling processes.

The model describes the situation under real casting conditions and can be easily adapted to laboratory conditions, as described later. A strand fixed coordinate system is used which moves in casting direction. Time \( t \) and position \( x \) are related by \( x = v_c \cdot t \) for a constant casting speed \( v_c \). The distance from the strand surface is described by the \( y \) coordinate. The temperature field in the strand shell is decomposed in two components:

\[
T(y, t) = \overline{T}(y) + T'(y, t) \tag{4}
\]

The first component \( \overline{T}(y) \) is the temperature field averaged in the casting direction, considering the heat flux from the liquid core to the strand surface. The second component \( T'(y, t) \) is the local temperature difference field caused by locally changing cooling conditions in casting direction due to the spray cooling.
without considering the heat flux from the liquid core to the strand surface.\textsuperscript{[14]} For simplicity, the averaged surface temperature and surface heat flux are assumed to be constant in time; therefore, the surface temperature \((y = 0)\) is

\[
T(0, t) = \overline{T} + T'(0, t)
\]  

(5)

and the surface heat flux is

\[
\dot{q}(t) = \dot{q} + \dot{q}'(t)
\]  

(6)

where \(\dot{q} < 0\) is the average cooling heat flux and \(\dot{q}'(t)\) is the difference between actual and average cooling heat flux.

If we suppose we know the average heat flux and surface temperature, only the local temperature difference field \(T'(x, t)\) has to be calculated. For an initial condition \(T'(x, 0) = 0\) and for a surface heat flux \(\dot{q}'(t)\), the solution is

\[
T'(y, t) = \theta(y, t) \ast \dot{q}'(t)
\]  

(7)

where \(\ast\) denotes a folding operation in time and

\[
\theta(y, t) = \frac{1}{b \sqrt{\pi t}} e^{-\left(\frac{x^2}{2at}\right)}
\]  

(8)

is the pulse response to a unit Dirac pulse obtained by differentiating the analytical solution of the 1D transient heat transfer equation for a step-wise change of the heat flux at the surface of a semi-infinite body where \(a\) is the body’s thermal diffusivity and \(b\) is its thermal effusivity.\textsuperscript{[17]} The application to temperature differences in the strand shell with a finite shell thickness is justified as the temperature differences due to the inhomogeneous surface cooling situation decay fast and become negligibly small within a surface distance of a few millimeters. As an example for the situation shown in Figure 3a, the temperature difference at a distance of \(y = 30\) mm from the shell surface is 1.5 K over the shown \(x\)-axis range, whereas it is over 200 K at the shell surface \((y = 0)\).

For time discrete systems, the folding operation is a convolution of two vectors that can be calculated efficiently using the fast Fourier transformation.\textsuperscript{[18]} Now the initial condition \(T'(y, 0) = 0\) is generally not correct as it would imply a homogeneous cooling for \(t < 0\). The passage of the strand beyond one nozzle row after the other can be seen as a quasiperiodical process, where the influence of the initial condition disappears quickly. Therefore, we start our calculation with a homogeneous initial condition one or two nozzle rows before the nozzle row we want to look at and discard the data of the previous nozzle row(s).

Figure 3. Upper diagrams: surface temperature during the passage of a spray nozzle under real conditions at the beginning of a) a conventional slab casting and b) a thin-slab casting secondary cooling zone and under corresponding laboratory conditions with different starting temperatures and isothermal initial conditions calculated with the 1D transient thermal model; lower diagrams: corresponding surface heat fluxes.
Using Equation (6), the surface heat flux difference

$$\dot{q}'(t) = - \max(\dot{q}_R(T(t)), \dot{q}_H(t, T(t))) - \dot{q}$$

(9)

where \(\dot{q}_H(T) > 0\) is the radiative heat transfer and \(\dot{q}_H(t, T) > 0\) is the cooling heat flux due to spray cooling including radiation. For the spray cooling heat flux, a Gauss-shaped spray density over time is assumed. Based on the actual spray density and surface temperature, the cooling heat flux is calculated with the formula proposed by Wendelstorf et al.[8] In Figure 2b, the results of this formula are plotted as solid lines in the valid range from 3 to 30 kg m\(^{-2}\) s\(^{-1}\) spray density. Higher spray flow densities are meanwhile standard for high-performance cooling in thin-slab casting, but no comparable measurement data seem to be published by now. In the sense of the aforementioned “what-if” character of the model, the function proposed by Wendelstorf et al.[8] is empirically extrapolated to represent a similar behavior, as shown in Figure 2a, for spray densities \(n'_W > 30\) kg m\(^{-2}\)s\(^{-1}\) to

$$\alpha(\tau, \mu)/\alpha_{ref} = 190 + \tanh\left(\frac{\mu}{80000}\right) + 3.26 \cdot \tau^2 \cdot \left(1 - \tanh\left(\frac{\tau}{128 + \mu}\right)\right)$$

(10)

where \(\alpha_{ref} = 1\) Wm\(^{-2}\)K\(^{-1}\), \(\tau = (T_S - T_W)/\Delta T_{ref}\), where \(\Delta T_{ref} = 1\) K is the dimensionless local temperature difference between strand surface and spray cooling water, and \(\mu = n'_W/\mu_{ref}\), where \(\mu_{ref} = 1\) kg m\(^{-2}\) s\(^{-1}\) is the dimensionless local spray density. Inserting Equation (8) and (9) (with \(\dot{q}_H = \alpha \cdot (T_S - T_W)\) and the local spray density \(n'_W(t)\)) and Equation (10) in Equation (7) yields an implicit equation for the temperature field as the heat flux depends on the resulting surface temperature. Therefore, the solution is calculated iteratively, using the temperature result of the previous step for the calculation of the heat fluxes. Usually a converged solution is obtained after less than ten iterations.

Figure 3a shows an exemplary model result of a real casting situation of a conventional caster at the beginning of the secondary cooling zone for a casting speed of 1.2 m min\(^{-1}\) with a maximum local spray density of 30 kg m\(^{-2}\) s\(^{-1}\). In addition, a comparable laboratory situation without average heat flux (i.e., isothermal initial conditions and \(\dot{q} = 0\)) and different starting temperatures is shown. The middle start temperature is chosen such that the total heat flux in the experiment simulation is equal to the total heat flux of the strand simulation. The diagram shows the surface temperatures and heat flux densities of a strand surface element moving from left to right between two roll contacts with a nozzle cone located at the center. The influence of the surface temperature on the heat flux can be clearly seen. The strand surface temperature has always a steeper slope than the experimental surface temperature as there is a permanent heat flux through the shell in the real situation. Nevertheless, the average temperatures of simulation and experiment are approximately equal if the total heat flux is equal. During spray cooling, the surface temperature decreases from 1105 to 880 °C, whereas the average temperature is 1030 °C. This low level of the minimum temperature, occurring in the region of the highest cooling heat flux, illustrates that is important to consider the local temperature instead of the average temperature for the correct determination of the heat flux. In Figure 3b, the simulation result of a thin-slab casting situation with a casting speed of 6 m min\(^{-1}\) and a maximum spray density of 250 kg m\(^{-2}\) s\(^{-1}\) is shown. Although the underlying heat transfer relation is only a rough estimation, the result shows impressively how a high Leidenfrost temperature can lead to an unstable cooling situation: At the beginning, the cooling heat flux stays at a moderate level as long as the surface temperature is above the Leidenfrost temperature. As a result of the cooling heat flux, the surface temperature decreases. If the Leidenfrost temperature is reached, the cooling heat flux increases, causing the temperature to decreases even faster, which leads to an unstable self-amplifying situation. Although the surface temperature decreases to a very low (and maybe unrealistic) level of 330 °C, it increases quickly as heat is transferred from deeper layers of the shell (or probe) and reaches a level of 800 °C quite after the strand surface leaves the spray nozzle impingement zone.

2.2. Draining Spray Water Model

Another mathematical model is proposed for the formation of draining spray water retained by rolls. As shown in Figure 4, it is assumed that a roll consists of three parts. Between the roll parts, retained water can flow through the two gaps between bearing support and strand to the next lower roll gap. This amount is \(\approx 2/3\) of the draining water, whereas the remaining \(1/3\) leaves the roll gap at the strand’s narrow faces and does not reach the next lower gap. As a consequence, an equilibrium total water charge of each roll gap that is three times the output of the nozzles in each gap is reached after a few roll gaps with identical water charge. For symmetry reasons, one-sixth of the total roll width is considered, which is charged approximately by one-sixth of the total gap water \(V\), originating from the spray nozzles in the actual gap plus the water draining from the gap(s) above. The draining water flow rate depends on the retained water level. Under steady-state conditions, charged and draining water must be equal

$$V/6 = A(h) \cdot \dot{u}(h)$$

(11)

where \(V\) is the water flow rate charged over the whole gap, \(A(h)\) is the cross-sectional area of the retained water depending on the retained water level \(h\) (Figure 5), and \(\dot{u}(h)\) is the draining flow velocity \(\dot{u} = c\sqrt{2gh}\) where \(g\) is the gravitational acceleration and \(c = 0.4\) is a dimensionless factor with a value typical for water flowing over a dam.[19] This equation can be solved with respect to the retained water level \(h\) for each roll gap. Figure 5 shows a

Figure 4. Roll retaining draining spray water: front view (left) and side view (right) on a roll consisting of three parts in the vertical part of a strand casting machine.
few exemplary results for a conventional slab casting machine at different roll gaps. According to the model results, the retained water covers up to about 40% of the strand surface in a roll gap and will probably significantly contribute to heat transfer. The cooling HTC in these regions may be $500^{[20]} - 1500^{[3]}$ Wm$^{-2}$K$^{-1}$. As the forces of the spray nozzle jets are not considered, the level of retained water may be lower in reality, and the retained water levels calculated with the described model are somehow a “worst-case” scenario.

3. Experimental Facilities

3.1. Influence of Differences Between Real Situation and Laboratory Setup

Laboratory setups are generally a simplification of the real situation. The following effects can influence the heat flux in the real situation but are (partially) neglected in laboratory setups.

3.1.1. Effect 1: Hot Strand Core Acting as permanent Heat Source

Under real casting conditions, there is a permanent heat flux through the strand shell originating from the liquid core. In all known laboratory setups, this effect is not considered as it would require a very strong heating source that must be active during the spray nozzle passage. In the laboratory setups, the probe is heated up before the trial to almost a homogeneous temperature either by an external (electrical heater$^{[13]}$ and furnace$^{[8]}$) or integrated (e.g., induction heating$^{[4]}$) heat source. As a consequence, the probe surface temperature is permanently decreasing in the laboratory setup in comparison with the real situation (Figure 3).

3.1.2. Effect 2: Moving Strand Surface

Under real casting conditions, the hot strand surfaces moves with casting speed in relation to the spray nozzle. Thus an asymmetric surface temperature profile is established, as shown in Figure 3, and the process can be seen as steady and, if several subsequent nozzle rows in the casting direction are considered, even quasiperiodic. The simulation results in Figure 6 show how an increasing casting speed reduces the surface temperature drop during the cooling process and therefore reduces the cooling heat flux. Therefore, different velocities between hot probe and spray nozzle should be considered in experimental facilities to obtain cooling heat fluxes for different casting speeds. In some test facilities nozzle and hot surface do not move with respect to each other, changing the process from a steady to a transient cool-down procedure$^{[8,10]}$. For symmetric nozzle spray densities, symmetric temperature profiles are established that are always significantly different to the realistic steady profile, no matter at what point in time the transient process is looked at (Figure 7). So this kind of experiment is only useful if there is no influence of the surface temperature on the heat transfer. As a further drawback, a high density of sensors is required to resolve typical flat spray nozzles satisfyingly.

3.1.3. Effect 3: Heat Transfer outside Nozzle Centre

A further decrease in similarity to the real casting situation occurs in the case of no motion between probe and nozzle if only the surface heat transfer at the nozzle center line is considered and furthermore if a different nozzle type is used in the laboratory setup (e.g., round spray nozzle$^{[8]}$) than in the real situation (e.g., flat spray nozzle). It does not seem appropriate to apply such experimental results to an arbitrary nozzle spray profile.
This could be accomplished by considering the HTC measured at the nozzle center \( \alpha_1 \left( \dot{m}_W \right) \) for different spray densities \( \dot{m}_W \) at the nozzle center to determine the local heat transfer \( \alpha_2(x, z) \) at an arbitrary position \( x, z \) of the desired nozzle according to its local spray intensity \( \dot{m}_W(x, z) \): \( \alpha_2(x, z) = \alpha_1(\dot{m}_W(x, z)) \), as it is done in the aforementioned mathematical model using \( \alpha_1 \) from a previous study\(^8\), as no other data are available. This neglects convective film effects as well as different spray impingement angles and inhomogeneous surface temperatures.

### 3.1.4. Effect 4: Overlapping Spray Cones

Similarly, the heat transfer of the overlapping nozzle regions of flat spray nozzles used in slab casting should be considered in the laboratory setup; especially in these regions, spray water density and heat transfer do not correlate as known from measurements.\(^4,5\)

### 3.1.5. Effect 5: Probe Size

The water cooling effect generally depends on the surface temperature distribution if cooling water flows parallel to the hot surface. This is especially neglected in laboratory setups if the hot surface is too small: if the heat transfer in the outer regions of the spray cone is measured, a cooling water film can flow from the nozzle center toward the measurement position. The measured heat transfer will be different if the water film is only partially over the hot surface (Figure 8). Therefore, the hot surface should be larger than the size of the spray impact area,\(^14\) nevertheless, smaller probe sizes are often used.\(^4,7,8,11\)

### 3.1.6. Effect 6: Impact of Roll Geometry

The roll surfaces in combination with the strand surface form a nearly closed cavity (Figure 1b). The roll surfaces can reflect splash water and retain draining water, as shown in Figure 5. Both effects can contribute to the strand surface cooling and should be considered in an experimental setup like, e.g., in the study by Raudensky and Horsky.\(^14\)

### 3.1.7. Effect 7: Orientation of Strand Surface

The orientation of the strand surface and spray nozzles with respect to gravity changes in a bow-type caster gradually from a vertical to a horizontal strand surface with asymmetric conditions on the inner and outer surfaces in the strand bow: Draining

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**Figure 6.** a) Surface temperature during the passage of a spray nozzle under real conditions at the beginning of a conventional slab casting secondary cooling zone (solid line: local temperature, dashed line: average temperature) with similar starting temperatures and equal cooling nozzle conditions, but with different casting speeds calculated with the 1D transient thermal model. b) Surface heat fluxes (solid line: local heat flux, dotted line: average heat flux); dashed black line: Gaussian spray density distribution.

**Figure 7.** Surface temperature due to spray nozzle cooling under real casting conditions (dashed line), under laboratory conditions without permanent heating (red solid lines) with moving probe (or nozzle, left, steady) and with fixed probe and nozzle (right, transient, different points in time).

**Figure 8.** Effect of a small heated surface in laboratory setup (right) in comparison with a large heated surface (left).
water is easier retained at the inner side of the bow, spray water reflected by the strand surface will rather stick the surface in the inner bow. All known existing test facilities seem to have a fixed orientation: the probe surface is either facing upward and sprayed from top\cite{8} or facing downward and sprayed from below.\cite{14,15}

3.1.8. Effect 8: Scaling

As a part of the continuous casting process, scale forms on the strand surface, which is known to influence the heat transfer. It is hard to ensure similar scale formation in a laboratory setup, the scale may vary during experiments as an unidentifiable influence parameter. Some groups measured the influence of scaled surfaces.\cite{9,10} In most setups, a different material than in the casting process is used without scale formation like stainless steel\cite{8,13} or nickel.\cite{9}

3.1.9. Effect 9: Different Probe Materials

Basically the local heat transfer only depends on the surface temperature and the surface roughness, whereas the thermal properties of the probe material do not have a direct influence. Nevertheless, the thermal properties of the probe material influence the surface temperature change during the passage of the strand under a nozzle and therefore indirectly influence the heat transfer by the surface temperature. This effect can be compensated by adopting the probe or nozzle speed accordingly, as described later. The temperature-dependent thermal probe material properties have to be known for the mathematical determination of the heat flux from measured temperatures as described in the following paragraphs. Especially, phase transitions make the heat flux calculations difficult.\cite{9}

3.1.10. Conclusions and Compensation Proposals

All the mentioned criteria should be either considered in a laboratory setup or it should be proven that they are not relevant, or if both is not possible, relevant but neglected effects should be somehow compensated. In the study by Raudensky and Horsky,\cite{14} it is reported that stationary tests with no relative motion between the nozzle and the cooled surface did not provide results suitable for precise cooling models and that using a large cold plate with a small heated element produced conditions far from those in the casting plant.

All known facilities neglect the effect of the permanent heat flux (effect 1). A possible compensation could be to measure the heat flux for different starting temperatures, as shown in Figure 3. Using the model in Equation (7), the surface heat flux in the real casting situation can be calculated by interpolating between the measured heat flux curves according to the instantaneous surface temperature. This procedure neglects convective effects in combination with inhomogeneous surface temperatures but is probably the best possible compensation.

A further proposed compensation concerns the use of probe material that is different than the real material (effect 9). According to the surface temperature curve arising due to a sudden change in the heat flux,\cite{17,21} the local surface temperature depends with $1/b \cdot \sqrt{x/v}$ on the position $x$. If a probe material has the thermal effusivity $b \neq b$, the surface temperature curve is equal if a scaled feeding velocity $v_s = v \cdot (b/b)_s^2$ is used in the experiment.

All other effects should be considered in an experimental setup: a heated and large-enough probe (effect 5) moves with (scaled) casting speed through a realistic nozzle arrangement (effects 3 and 4), surrounded by emulated roll surfaces (effect 6) and with the angular orientation of the probe surface equal to the real situation (effect 7).

3.2. Heat Flux Measurement

The next challenge is to measure the surface heat flux and the corresponding surface temperature in the laboratory setup. The cooling usually effects only a thin layer beyond the surface, with high temperature gradients and fast temperature changes near the surface. Therefore, a direct measurement of the surface heat flux is not possible without influencing the cooling process. As a consequence, the heat flux is usually measured with temperature sensors placed inside the hot probe close to its surface. The surface heat flux and temperature are then reconstructed using inverse modeling. Inverse modeling of thermal problems is characterized as an “ill-posed problem,” which results in unwanted large artificial oscillations in the solution. The mathematical algorithm of regularization can help damp these oscillations but always with the potential drawback that relevant oscillations are also damped. Therefore, it is necessary to make the problem as little as possible “ill posed.” This can be achieved by adopting the sensor depth and the sampling rate to the cooling conditions and the sensor noise.\cite{18}

The following figures show results from the simulations of a cooling process corresponding to an average heat flux density of $\approx 500 \text{ kW m}^{-2}$. In Figure 9a, it is shown that how the measurement error can be reduced by placing the sensor closer to the surface and reducing the sensor noise level. For example, for a typical sensor depth of 2 mm, 8% error can be achieved with a noise level of 0.68 K; reducing the sensor depth to 1 or even 0.5 mm could reduce the error to 5% or 3%, respectively. Especially for higher cooling rates as required in thin-slab casting, the sensor depth should be even lower than 0.5 mm to achieve an acceptable accuracy.

According to Figure 9b, a sampling frequency of 1 kHz is at least recommended, higher frequencies help to further increase the measurement accuracy.

There will be limitations for the thickness of the remaining probe material between surface and the end of the drill hole. The thermocouple should be as close as possible to the end of the drill hole, and it should have a good thermal connection to the probe. The ceramic insulation of the thermocouple increases the effective distance of the thermocouple and finally, the measuring position of a thermocouple is at its lower end\cite{18} (red dot in Figure 10). As an alternative, a ceramic probe material could be used in combination with uninsulated thermocouples, as shown in Figure 10. Even in this case, a distance of 0.5 mm is already hard to achieve. For a good accuracy of the results, the effective measuring position (depending on the actual sensor depth) and the thermal probe and thermocouple material
properties have to be known accurately. A calibration procedure for both quantities is proposed in the study by Javurek et al.\cite{22}

Even for an optimized setup, regularization will be necessary to solve the inverse problem for the determination of the surface heat flux and temperature based on the measured thermocouple temperature. Regularization algorithms usually introduce a parameter that determines how smooth the solution will be. There are different proposals of how to estimate an optimum regularization parameter based on the measured data like the L-curve criterion,\cite{18} but simulations showed that the so-determined parameters could be quite far away from the optimum parameter value.\cite{18} Unfortunately, such optimum values can only be determined in simulations, where the surface heat flux is known exactly, in contrast to measured data, where the corresponding surface heat flux is unknown.

Finally, despite all described challenges, if a measurement of the surface heat flux and temperature during a nozzle passage can be carried out, there are still two questions open: which parameter variations should be considered in the experiment and how can measured data be applied in (real-time) simulations of the casting process?

Figure 9. Heat flux measurement error using an immersed temperature sensor: a) with different noise signal levels as a function of the sensor depth, b) as a function of the sampling frequency.\cite{18} \( n \) denotes the number of samples used for the heat flux determination.

As the spray nozzle cooling heat flux generally depends on the local surface temperature, and the surface temperature decrease during a nozzle passage depends on the casting speed, heat flux measurements should be carried out for different initial surface temperatures and for different feeding velocities. As a result, the local heat flux density and surface temperature is obtained as a function of time, which can be transformed to the surface position relative to the nozzle.

For the control and for simulations of the casting process, an averaged heat flux based on an average surface temperature is often required. These data could simply be obtained from the measured time signal by averaging. The simulations showed that the average surface temperatures of the experiment and casting process are approximately equal if the average heat flux is equal (Figure 3). For a more accurate approach, the proposed mathematical model (Equation (4)–(9)) could help to calculate local surface temperatures based on averaged temperatures with little effort: Using local surface temperatures, the local heat transfer can be calculated by the interpolation of the measured data for the heat flux \( \dot{q}_N \) (Equation (9)) for the actual local surface temperature and for the actual casting speed. Finally, the so-calculated heat flux curve can be averaged and used for the casting simulation as a boundary condition.

4. Conclusions
Measuring the spray cooling heat transfer in laboratory experiments is essential for tuning the cooling conditions in the secondary cooling of continuous steel casting. The functional dependence of the surface averaged heat transfer from the surface-averaged spray density is often confused and mixed up with the dependence of the local heat transfer from the local spray density but have to be clearly distinguished due to underlying nonlinear relations. While the influence of the surface temperature on the cooling heat flux can be neglected for smaller water spray densities, the influence becomes relevant for higher spray densities as the Leidenfrost temperature increases with increasing spray densities and therefore the surface temperature can drop below the Leidenfrost temperature during the nozzle passage. Especially for the high-performance cooling required
in thin-slab casting, this can result in a drastic and unstable increase in the cooling effect, as could be shown exemplary a mathematical simulation model. A list of criterions is proposed for experimental setups, followed by simulation-based findings on the accuracy of the inverse modeling process, which is required to calculate the surface heat flux from the measured data of immersed temperature sensors. The non-negligible influence of the surface temperature requires to extend the parameter space for measurements: experiments for a certain nozzle arrangement have to be conducted for different initial surface temperatures and probe feeding speeds, corresponding to different casting speeds. Finally, a procedure is proposed to apply the locally varying measured data to (real-time) casting simulations using averaged temperature and heat flux data.

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Conflict of Interest

The authors declare no conflict of interest.

Keywords

continuous casting, heat transfer measurements, inverse modeling, secondary cooling, water spray nozzles

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