Heat Transfer Characterization of Secondary Cooling in the Horizontal Direct Chill Casting Process for Aluminum Alloy Re–Melt Ingot

by

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Abstract

Horizontal direct chill (HDC) casting is a continuous process used to produce extrusion billet and re-melt aluminum ingot. As in vertical DC casting, secondary cooling, where water directly impinges on the cast surface, is an important process that can affect cast quality and production rates. During HDC casting, secondary cooling is further complicated by horizontal water flow and the water spray conditions. Characterizing the heat transfer during the secondary cooling process is necessary for improved understanding of the process. Since the accessibility of the HDC casting machine is limited and the direct measurement of heat transfer in secondary cooling are difficult, numerical modeling thus becomes a good approach for process development.

In this research, the heat transfer occurring in secondary cooling of an HDC ingot has been studied. Water spray conditions on three different casting surface were simulated separately by quenching the blocks of HDC cast A356 aluminum alloy which was cut from a T-ingot. The temperature history during the cooling within the blocks was recorded by sub-surface thermocouples. An inverse heat transfer model was developed and used to calculate the heat fluxes on the casting surfaces using measured temperature data. The heat fluxes were characterized via boiling curves, which are the functions of surface temperatures, in each spray configuration.

The effects of operational parameters, including the casting speed and the water cooling rate, were investigated by comparing the characteristic features of the calculated boiling curves. The spray configuration effect was also studied with the calculated results from the
stationary tests in a qualitative fashion.

Then a fitting technique was developed to idealize the calculated boiling curves. The idealized boiling curves were summarized into the functions, which provide practical database for application of the results in this research.

All in all, the simulation apparatus and the IHC model provide the ability of characterizing the heat transfer occurring in secondary cooling region of HDC casting with lab–scale experiments. Consequently, the expensive and risky plant trials can be avoided.
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Chapter 1

Introduction

As the most abundant metal and the third most abundant element in the Earth’s crust, aluminum is the world’s second most used metal following steel. Most of aluminum’s applications are in its alloys with properties tailored by adding other metals. These additions include copper (2000 series), manganese (3000 series), silicon (4000 series), magnesium (5000 series), zinc (7000 series), and others[1]. With various desirable properties, these aluminum alloys are now used widely in applications including products for household use to aerospace components. This widespread usage has driven the world’s production to dramatically increase to 33.7 million metric tonnes per year, of which about 9% is produced in Canada[2].

Approximately 70% of the aluminum produced around the world is first cast into ingot prior to further processing into semi-finished and finished products. Figure [1.1] shows the growth in worldwide production of primary aluminum over the last 60 years. The aluminum ingot manufacturing industry is now moving toward larger ingot cross sections, higher casting speeds, and an increasing array of mold technologies to increase overall productivity and satisfy demand. The direct chill (DC) casting process is the most widely used aluminum manufacturing processes around the world.
1.1 Direct Chill Casting

In the DC casting process, cooling which occurs in the mold is called primary cooling and is augmented by the water ejected out of the mold which is called secondary cooling. The majority of the heat extraction in DC casting occurs during the secondary cooling process, resulting in a shallower sump depth and flatter solidification profile compared to the mold casting without the direct chill. Also, compared to the traditional permanent mold casting, less macrosegregation, structure inhomogeneity and residual stresses occur in the DC cast product[3]. Based on the different casting orientations, DC casting can be classified into two types, vertical DC (VDC) casting and horizontal DC (HDC) casting.

1.1.1 Vertical DC Casting

A schematic picture of a VDC casting system is shown in Figure 1.2. At start up, a starter block is partially inserted into the mold and liquid metal is poured onto this while filling the mold at a prescribed rate. The liquid metal is first cooled by the primary cooling. The cross-section of the ingot or billet is determined by the size/shape of the mold. Once
the metal reaches a target height on the mold, the cast ingot is withdrawn from the mold at a preset velocity by lowering the starter block. The ingot has a thin solid shell upon emerging from the mold. During the start-up, heat transfer occurs between the solid shell and bottom block. As it exits the mold, secondary cooling occurs, the water jets ejected from mold and impinge on the cast ingot. The bulk of the heat (> 80%) is removed from the ingot during this secondary cooling process. The ingot is lowered into the pit until it reaches the length limit of the pit (∼ 10m). Then, the casting process is stopped, the mold system is moved, and a crane is used to remove the ingot from the pit. Thus, the VDC casting is a semi-continuous casting process, the process has to be stopped and reset for a new start.

![Diagram of Vertical DC casting](image)

**Figure 1.2:** Schematic picture of Vertical DC casting

Compared to permanent mold casting, better surface quality can be achieved via VDC and a more homogeneous structure with reduced centerline segregation is produced resulting in quality improvements. With regards to the production costs, it has been reported that there was a 50% reduction in workforce for DC casting operations compared with permanent
mold casting[4], leading to significant savings.

Despite the advantages of VDC casting, some drawbacks remain:

- Higher capital costs and significant plant requirements, including an overhead crane and deep pits.

- Semi-continuous operations lead to lower productivity compared to fully-continuous casting operations. The workforce is required in set-up work and removing the ingot from the pit.

1.1.2 Horizontal DC Casting

The HDC casting system is shown schematically in Figure 1.3. Compared with VDC casting, the ingot in HDC casting is horizontally withdrawn out of the mold, into a water spray. The cast ingot is withdrawn by a conveyor set to move at the casting speed. A flying saw is used to cut ingot to the required length. The secondary cooling process is still the dominant cooling stage.

![Schematic picture of Horizontal DC casting](image)

Figure 1.3: Schematic picture of Horizontal DC casting

Compared to VDC casting, HDC casting has the following advantages[5]:

4
• Lower capital costs since there is no crane or pit required in the HDC system.

• High productivity with a continuous casting operation, which can run up to 20 days without being stopped.

• High automation, only one worker is needed for T-ingot production.

• The length of ingot is not limited by the depth of pit, and within a predetermined range, the flying saw can cut the ingot into prescribed lengths without stopping the operation.

In modern aluminum casthouses, HDC casting is employed to produce re-melt ingot, T-bar/busbar and extrusion billet. Its use has expanded due to the production efficiency gained by continuous operation.

In order to improve HDC casting productivity, lower operating cost and improving product quality, the understanding and characterizing of fundamental process behavior and operation parameters is essential. In particular, the heat transfer occurring in the secondary cooling process, where most of the heat is removed, is one of the most important aspects of the process, which needs to be thoroughly studied.

Since the accessibility of an HDC casting machine is limited and plant trials in an industrial facility are difficult and costly, numerical modeling becomes an effective and efficient means to understand the intrinsic features of the operation process, which leads to process improvement and product development, especially by building on the experience and successful work in the area of VDC casting[6–11]. But compared with VDC casting, characterization of secondary cooling in HDC casting is challenging due to the distinct flow conditions on the three casting surface orientations, namely, top, side and bottom, as well as the unsteady heat transfer induced by boiling heat transfer.

As a follow–on project to a collaborative research and development program between Rio Tinto Alcan and The University of British Columbia, current work is focused on char-
acterizing heat transfer taking place in the secondary cooling process with lab-scale experimental results.
Chapter 2

Literature Review

2.1 Water Flows during Secondary Cooling

Since the ingot is directly cooled by impinged water during secondary cooling, the water flow has a significant impact on the heat transfer between the ingot surface and the cooling water. In this section, the water flow types in secondary cooling of both VDC and HDC casting process are discussed. The emphasis is put on the different flow types associated with different water spray configurations in HDC casting process.

The water spray that contacts the ingot is formed by water jets ejecting from the mold through an array of holes. A cross section of an HDC mould is shown in Figure 2.1. Hole sizes, orientation, and spacing are fixed for each mold and define the features of the water jets, such as jet diameter and impingement angle.

Based on the flow characteristics of the cooling water, it is common that secondary cooling is divided into two zones (Figure 2.2) in VDC casting[9, 12]:

- Impingement zone: where the water hits the casting surfaces with momentum normal to cast surface.

- Free falling zone: where the water falls down along the casting surfaces, accelerated by gravity.
In HDC casting, the casting surfaces have the three distinct configurations in terms of cooling water flow:

- Top: where the normal of the exterior surface points in the opposite direction to gravity.
- Side: where the normal of exterior surface is perpendicular to the direction of gravity.
- Bottom: where the normal of the exterior surface points in the same direction as gravity.

Thus, during the secondary cooling process of an HDC casting, flows of water along the cast
surfaces are further complicated, see Figure 2.3. Taking a square cross-sectioned ingot for example, the impingement zone (IZ) occurs on all the casting surfaces. However, a forced flow zone (FFZ), depending on the normal outward of the ingot of casting surfaces, occurs in HDC as compared to the free falling zone in VDC casting.

![Figure 2.3: Flow on three casting surfaces](image)

During secondary cooling of VDC casting, the water impinges on the ingot and flows down along the all casting surfaces. Whereas in HDC casting, three different types of water flow are expected owing to the different surface normals outward of the ingot. For top spray, in Figure 2.3a, the water jets flow along the casting direction and may fall off from the side surfaces. For bottom spray shown in Figure 2.3b, the water will fall off from the sample after creeping along the bottom surface for a certain distance. As to the side spray shown in Figure 2.3c, the parabolic-curve flow will form due to gravity.

Considering the distinct flow patterns expected for the three different surface normals during secondary cooling in HDC casting, different heat transfer regimes will exist, especially in the FFZ. For each water spray condition, the flow pattern is controlled or affected by many operational parameters, such as impingement angle, water velocity and casting speed.

### 2.2 Boiling Water Heat Transfer

The water used for cooling during HDC casting is below its saturation temperature $T_{sat}$ (boiling point for water, 100°C). When the water contacts the hot surface of a cast in-
got which is initially around 500°C, vapor bubbles form and may condense in the cold liquid[13]. This type of boiling is classified as subcooled boiling and provides the potential for extremely high peak heat fluxes.

**Figure 2.4:** Nukiyama’s boiling curve[14]

Nukiyama[15] was the first investigator to characterize the boiling water heat transfer phenomenon using the boiling curve, also called Nukiyama’s curve, depicted in Figure 2.4. In the boiling curve, the heat flux extracted from the hot surface by the cooling water is usually plotted versus wall temperature or wall superheat, $\Delta T_{\text{sup}} = T_s - T_i$. 
Figure 2.4 shows a typical pool boiling curve, which describes the heat flux resulting from a heated surface submerged in a quiescent liquid\cite{16}. Four distinct regimes can be identified, natural convection (A–B), nucleate boiling (B–D), transition boiling (D–E) and film boiling (E–F). In secondary cooling zone of HDC casting, natural convection is replaced by forced convection due to the fact that the cooling water on the casting surfaces is moving; however, the boiling curves with natural or forced convection are similar in shape\cite{13}. Since secondary cooling is a cooling process, the features of the boiling curve will be discussed in the order that they may be experienced (i.e. with decreasing temperature).

In film boiling (region EF in Figure 2.4), a vapor blanket, depicted schematically in Figure 2.5, forms on the hot surface and prevents direct contact between the water and the hot surface. Heat transfer rate is relatively low at the beginning of film boiling (point E) because of the low conductivity of the vapor blanket and increases to high values, due to radiation at high temperature ($\geq 1000^\circ C$).

\begin{figure}[h]
\centering
\includegraphics[width=0.5\textwidth]{vapour_blanket.png}
\caption{Vapour blanket in film boiling\cite{16}}
\end{figure}

Gunther\cite{17} studied film boiling using degassed subcooled water flowing through a horizontal rectangular glass duct with a heater on the central axis of the duct. By high–speed photography, he filmed bubble formation and detachment. The detached bubbles had a lifetime of only 220 microseconds before disappearing due to condensation in the subcooled liquid. Thus since only a small amount of water vapor escapes, heat losses by vapor escaping can be neglected in film boiling.

Besides conduction through the stable film layer, heat is also transferred by radiation in film boiling. However, based on a study of film boiling, Bromley\cite{18} found that the effects of radiation heat transfer were substantial only when the superheat $\Delta T_{sup}$ was greater than $1000^\circ C$. For aluminum casting, the temperature of the ingot entering the secondary cooling
is below 555°C, therefore radiation heat transfer in the film boiling can be neglected.

As wall superheat decreases, heat flux decreases until a minimum heat flux is achieved (point E in Figure 2.4) marking the start of the transition boiling regime. In 1756, Leidenfrost[19] was the first person to report the existence of this point, and it was subsequently named the Leidenfrost point.

Transition boiling, an intermediate regime between the nucleate boiling and the film boiling regimes, was described as a “vacillating” process by Bejan[14]. The vapor film is unstable, and broken into several segments, where vapor bubbles may detach from the hot surface, as shown in Figure 2.6. Hence film boiling may coexist next to nucleate boiling or at the same location on the surface[14].

![Figure 2.6: Transition boiling][16]

With the decreasing of the surface temperature, a peak heat flux (point D) is reached. This maximum heat flux is known as critical heat flux (CHF), shown in Figure 2.7a and marks the end of the transition boiling. The entire hot surface is covered by the active bubbles shown in Figure 2.7b. In the nucleate boiling, the heat flux decreases dramatically over small temperature drop. Point B is called the onset of nucleate boiling (ONB), which indicates that the nucleation stops and the bubbles disappear for a cooling process, shown in Figure 2.7c. The nucleate boiling ends at ONB and the forced convection takes place when the temperature keeps dropping. Since a large range of heat fluxes occur within a small range of surface superheats, nucleate boiling plays the most important role in boiling heat transfer[13, 14, 16].

Forced convective heat transfer in secondary cooling is induced by the turbulent jets coming out from the mold. Figure 2.8 shows a schematic diagram of one jet’s evolution after impacting on the casting surface. As the water emerges from the mold, the wall friction
is lost and then replaced as the jet impacts on the casting surface. When the water impinges on the casting surface, the water jets decelerate in the direction normal to surface and in the casting direction. Pressure gradients evolve on the surface in this region as a function of distance from the mold. In the forced flow zone, fully developed turbulent flow occurs after a boundary layer (laminar) region and transition region (laminar to turbulent)[18]. In engineering practice, the transition region is usually ignored and it is assumed that the laminar to turbulent transition occurs at $Re_{x_{cr}} = 5 \times 10^5$ [18].

2.3 Modeling of Secondary Cooling

Little previous work on modeling the secondary cooling heat transfer phenomena occurring during HDC casting has been reported. Thus quantitative research on the effects of process parameters that affects the heat flux during HDC operation is not available.

However, a considerable number of studies have been published on heat transfer during secondary cooling in the VDC casting process. Considering the different flow and operational conditions that exist in HDC and VDC casting, most of the correlations developed for
VDC casting can not be directly applied to HDC casting. Nevertheless, from a quantitative point of view, the results of these correlations provide a reference for the magnitude of heat flux and effects of operational parameters that may be expected in secondary cooling during HDC casting.

Considering two distinct heat transfer region in secondary cooling process, IZ and FFZ, detail review of these literatures is given separately for four different boiling heat transfer regimes.

### 2.3.1 Film Boiling in Secondary Cooling

Kohler et al. [20] modeled the heat flux occurring in the film boiling regime as an average heat flux over the length of vapour blanket, rather than the local heat flux as a function of local surface temperature. A good agreement was found with the results from an experiment with a water film flowing along a vertically oriented nickel sample. The average heat flux is expressed by Equation [2.1]. This was used by Caron[12] to model film boiling during secondary cooling of VDC cast AA5182.

\[
\Phi_{AVG,FB} = \frac{2}{\sqrt{3\pi}} \frac{k_f}{L} \sqrt{\frac{(Pe^2 + 2Gn)^{3/2} - Pe^3}{Gn}} \left( T_{sat} - T_f \right) \tag{2.1}
\]

where \( Pe \) is the dimensionless Peclet number and \( Gn \) is the dimensionless gravity number, which are given by

\[
Pe = \frac{V_0L}{\alpha_f} \tag{2.2}
\]
\[
Gn = \frac{gL^3}{\alpha_f^2} \tag{2.3}
\]

In Equation [2.2] and [2.3], \( k_f \) is the thermal conductivity of the cooling water \((W/(m \cdot K))\), \( \alpha_f \) is the thermal diffusivity of the cooling water \((m^2/s)\), and \( L \) is the length of the vapour blanket \((m)\).

In the impingement zone, the effect of the gravity force is small compared to the large
momentum when jets hit on casting surface. Therefore, $Gn$ in Equation 2.1 tends to be zero\cite{12} in the impingement zone and the heat flux can be expressed by:

$$\Phi_{AVG,FB} = \frac{2}{\sqrt{\pi}} \frac{k_f}{L} \sqrt{Pe} (T_{sat} - T_f)$$  \hspace{1cm} (2.4)$$

The vapour film is continually broken down in the impingement zone making the characterization of $L$ difficult. For aluminum VDC casting, the vapour film length $L$ was assumed to be the impingement zone height, which was characterized by Caron through rewetting experiments. The resulting equation (Equation 2.5) includes the inlet flow rate, $Q$, ($L/min \cdot m$) to give the length of impingement zone ($mm$). It is expected that this equation will be valid for HDC casting.

$$H_{IZ} = 6.5 + 0.11Q$$  \hspace{1cm} (2.5)$$

In the free falling zone, Equation 2.1 includes the gravity force because the water is accelerated by gravity during the secondary cooling process in VDC. In HDC casting, different conditions must be considered because of the different orientations of the cast surface. On the top and bottom surfaces, gravity has a negligible effect on changing the magnitude of flow rate along the casting direction; however, on the side surfaces, gravity not only changes the magnitude of flow rate but also the direction of flow. No correlations has been reported for the heat flux in FFZ of HDC casting.

Additionally, it should be noted that the correlations related to VDC conditions were developed from water flowing on a stationary surface. In terms of cooling in the actual casting process, the casting surface is always cooled by jet impingement first, which means, when it comes to the FFZ, film boiling is unlikely to occur because the surface temperature will already be below the Leidenfrost point.
2.3.2 Leidenfrost Point and Transition Boiling in Secondary Cooling

Accurate knowledge of the Leidenfrost point is critical as it marks the beginning of transition boiling. It is effectively the point on the casting surface delineating wetting due to the breakdown of the vapour blanket. The heat flux changes greatly at this point because of the direct contact between casting surface and cooling water.

Previous research with low velocity jets showed that the Leidenfrost point is proportional to the square root\textsuperscript{[21]} or the cubic root\textsuperscript{[22]} of the water flow rate. For high velocity jets similar to the DC casting processes, finding the Leidenfrost point is challenging because of the unstable vapour film. The temperature when the vapour blanket is broken down and the surface wets was found to be linear related to the thermal effusivity of the cast material for a given set of cooling conditions\textsuperscript{[21]}.

Similar slopes for the transition boiling regime were observed for different casting speeds with the same casting material during VDC casting. The average value of the slope in IZ was slightly higher than that reported for the FFZ\textsuperscript{[12]}. Comparing the boiling curves\textsuperscript{[23, 24]} for water jet impingement on different materials suggests that the higher a material’s thermal conductivity, the higher the slope of the transition boiling regime. This suggests that the propagation of wet spots is controlled by heat conduction below the quenched surface rather than the hydrodynamics of water jets\textsuperscript{[12]}.

2.3.3 Nucleate Boiling in Secondary Cooling

Rohsenow\textsuperscript{[25]} recommended that the local heat flux for nucleate boiling be obtained by simply adding the heat flux of forced convection (discussed in next section) and pooling boiling, where nucleate pooling boiling is estimated by the semi-empirical equation:

\[
\frac{C_{p,f} \cdot \Delta T_{sup}}{i_{fg}} = C_f \cdot \sqrt{\frac{\Phi_b}{\mu_f \cdot i_{fg}}} \sqrt{\frac{\sigma_{fg}}{g (\rho_f - \rho_g) \left( \frac{C_{p,f} \cdot \mu_f}{k_f} \right)^{1.7}}} \tag{2.6}
\]

\[
\text{where:} \\
C_f: 	ext{film coefficient} \\
\Phi_b: \text{Nusselt number} \\
\sigma_{fg}: \text{surface tension} \\
l_f: \text{latent heat of vaporization} \\
\rho_f, \rho_g: \text{density of water and gas} \\
\mu_f: \text{dynamic viscosity of water} \\
k_f: \text{thermal conductivity of water} \\
\]
where $C_{p,f}$ is the specific heat of water ($J/kg \cdot K$), $i_{fg}$ is the latent heat of evaporation ($J/kg$), $\mu_f$ is the viscosity of water ($kg/m \cdot s$), $\sigma_{fg}$ is the surface tension between the vapour and liquid phase ($N/m$), and $\rho_f$ and $\rho_g$ are the densities of water in liquid and vapour states, respectively. In the equation, all properties must be evaluated at the temperature of water film boundary layer. The coefficient $C_f$ and $r$ were reported to be dependent on the cooling liquid properties and surface roughness[26].

Weckman[27] used Equation 2.6 to characterize nucleate boiling during secondary cooling of VDC casting. $C_f$ and $r$ were found to be equal to 0.016 and 3 respectively for aluminum AA6063 by comparison with experimental results. Caron[12] also used this equation for aluminum AA5182. $C_f$ and $r$ were quantified separately for the impingement and free falling zones in Caron’s work. The correlations for pooling boiling during AA5182 VDC casting were:

\[
\Phi_{PB,IZ} = 9.47(T_s - T_{sat})^{2.59}
\]

\[
\Phi_{PB,FFZ} = 33.0(T_s - T_{sat})^{2.33}
\]

(2.7)

(2.8)

The CHF in IZ has been found to be a quadratic function of water flow rate[12, 21, 28]. The relationship reported by Caron[12] is:

\[
\Phi_{CHF,IZ} = 1.0 \cdot 10^5 Q - 3.3 \cdot 10^2 (Q)^2
\]

(2.9)

In the free falling zone of VDC casting, the CHF decreases with the increasing distance from the IZ. A linear decrease of CHF in the first 12 inches from IZ was observed, followed by a constant value[29]. In the FFZ of HDC casting, the CHF is expected to be dependent on the distance along the casting direction and the orientation of the surface.
2.3.4 Forced Convection in Secondary Cooling

In the absence of boiling, i.e. when $T_s$ is below 100°C, heat transfer is dominated by forced convection regime. The relationship between the heat flux and surface superheat is governed by Newton’s law of cooling, $\Phi_{FC} = h_{FC} \cdot (T_s - T_f)$, where $h_{FC}$ is the heat transfer coefficient for forced convection regime ($W/m^2 \cdot K$).

The heat transfer coefficient in the free falling zone, $h_{FC}$ ($W/m^2 K$), during secondary cooling of VDC casting can be calculated by[13]:

$$h_{FC} = C_l k_f \left( \frac{\rho_f^2 g \mu_f^3 Q}{\pi D} \right)^{1/3}$$  \hspace{1cm} (2.10)

where thermal conductivity $k_f$ ($W/mK$), density $\rho_f$ ($kg/m^3$), specific heat $C_{p,f}$ ($J/kgK$) and viscosity $\mu_f$ ($kg/ms$) are the properties of water, $g$ is the gravitational acceleration in $m/s^2$, $Q$ is the volumetric cooling water flow rate in $m^3/s$ and $D$ is the billet diameter in $m$. The coefficient $C_l$ is dependent on the alloy, for example, $C_l = 0.0100$ for AA6063[27].

However, in the secondary cooling of HDC casting, the heat transfer coefficient ($h_{FC}$) in forced flow zone varies along the casting surfaces due to hydrodynamic variations in the streamwise direction and different surface orientations.

2.4 Experimental Methods

Since the correlations used to describe the heat transfer in computational model should be verified experimentally, various experimental methods were designed and applied for heat transfer measurement occurring in DC casting process.

Despite different methods, accurate temperature measurement is the unifying key factor to a good heat transfer measurement[30]. A direct measure of the energy transfer, i.e. heat flux, into or out of a surface is limited to the steady state conditions, normally two boundary conditions, constant heat flux and constant wall temperature. Thus the methods of direct heat flux measurement are seldom adopted for measuring heat flux in DC casting process.
Instead, by recording and interpreting the temperature versus time history, the heat flux boundary condition is calculated backwardly, which is in the form of an inverse heat transfer problem.

Many efforts have been reported to measure the temperature variation during actual VDC casting operations. One of the most common used methods is that an array of thermocouples are frozen into the solidifying metal. Usually the thermocouples were supported by a frame\cite{27, 31} to make sure their positions within ingot were precisely defined. However due to metal flow in the solidification process, the final thermocouple locations had to be determined by other methods, such as X–ray photography\cite{31} and ultrasound\cite{32}, after the operation. Instead of measuring temperature variation within the cast metal, Založnik\cite{33} developed a measurement technique for determining surface heat flux based on the measurement of the water film temperature during the secondary cooling of VDC casing operation. This method is a non–destructive technique and can be carried out in a real plant environment. The thermocouples were mounted by a specially designed support to a pivot which was welded to the casting machine’s frame. However, due to the volume of this measurement system, the thermocouple could hardly reach the impingement points, hence the essential information for highest surface heat flux was missed.

A majority of researchers\cite{12, 23, 34–37} studied the secondary cooling of VDC casting by quenching an instrumented sample with similar cooling conditions to those of the DC casting process. The beauty of this instrumented sample method is that it provides the flexibility of adjusting operational parameters (water flow rate, initial temperature, casting speed, etc.), thus making thorough investigation of heat transfer occurring in secondary cooling easier. The thermocouples are inserted into the samples at the sub-surfaces. The samples are then preheated to a desired initial temperature and submitted into a water jet or curtain cooling system. The measured temperature will be used as an input to an inverse heat conduction algorithm and the local heat flux will be evaluated at the boundaries. The heat flux is usually expressed via a boiling curve with the following variables:
• Heat flux in forced convection regime;

• Heat flux in nucleate boiling regime and critical heat flux (CHF);

• Heat flux in transition boiling regime and minimum heat flux at the Leidenfrost point;

• Heat flux in film boiling regime;

• Temperature for onset nucleate boiling;

• Temperature for CHF;

• Leidenfrost point (temperature for minimum heat flux)

Among these, the CHF and the temperature for CHF is the most critical parameters in the secondary cooling process, thus most interested and concerned. The magnitudes of the CHF and its temperature in the literature are summarized in Table 2.1.

### Table 2.1: Critical heat flux found for VDC casting

<table>
<thead>
<tr>
<th>Researchers</th>
<th>CHF (MW/m²)</th>
<th>Temperature of CHF(°C)</th>
<th>Alloy</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bakken[38]</td>
<td>5.0</td>
<td>130°C ~ 150°C</td>
<td>AA6063</td>
</tr>
<tr>
<td>Wiskel[39]</td>
<td>2.5 ~ 3.0</td>
<td>180°C</td>
<td>AA5182</td>
</tr>
<tr>
<td>Maenner[34]</td>
<td>4.4 ~ 6.6</td>
<td>130°C</td>
<td>AA5052</td>
</tr>
<tr>
<td>Caron[12]</td>
<td>4.5 ~ 6.0 (IZ)</td>
<td>300°C</td>
<td>AA5182</td>
</tr>
<tr>
<td></td>
<td>2.5 ~ 4.5 (FFZ)</td>
<td>200°C</td>
<td>AA5182</td>
</tr>
<tr>
<td>Wells[40]</td>
<td>5.0 (IZ)</td>
<td>300°C</td>
<td>AA5182</td>
</tr>
<tr>
<td></td>
<td>4.0 (FFZ)</td>
<td>200°C ~ 250°C</td>
<td>AA5182</td>
</tr>
</tbody>
</table>

### 2.5 Effect of Parameters in Secondary Cooling

Many parameters influence boiling water heat transfer occurring in secondary cooling process. However, so far little research has been reported on the heat transfer conditions in HDC casting, compared to many studies published on VDC casting[11, 20, 22, 27, 41, 42].
The important VDC parameters include, but are not limited to, water flow rate, casting speed, casting surface condition, cooling water temperature, water jet impingement angle and the cast material’s thermophysical properties. The influence of one given parameter on heat transfer occurring in secondary cooling of VDC casting is always reported in two aspects. One is the effect of changing the magnitude of the heat flux or the heat transfer coefficient in different boiling heat transfer regimes; the other is to influence the transition between the different regimes by increasing or decreasing the transition temperature. For example, a higher cooling water temperature results in lower heat flux in transition boiling regime[43] and an earlier start of film boiling by lowering the temperature, Leidenfrost point[44]. Most parameters’ effects are reported in a qualitative fashion by comparing the key features of the boiling curves in the literature. It is worthy to point out that the effects on heat flux of a given parameter are not straightforward in all boiling regimes. For instance, a higher water cooling rate helps the convection heat transfer, resulting in larger heat fluxes in forced convection regime but has few effects in nucleate boiling regime, where the heat is mainly transferred by bubble motion and nucleation.
Chapter 3

Objectives

As part of larger project on modeling HDC casting for T-ingot, this research focuses on understanding the heat transfer occurring during secondary cooling in HDC casting. A particular emphasis was put on characterizing the heat flux with relationship to the surface temperature for each water spray condition. The calculated results will provide the database which could be implemented as the boundary conditions in the mathematical process model for HDC casting.

Since it is almost impossible to make direct measurements in the confined HDC casting process, especially in secondary cooling region with its high velocity water jets, an apparatus has been designed and built to simulate the cooling conditions during the secondary cooling on three different surface normals, namely, top, bottom and side, respectively. With the temperature history measured by the embedded sub-surface thermocouples, the surface heat flux was calculated by an inverse heat conduction (IHC) model coupled with a 2-D finite element model (FEM). The accuracy of the IHC model was assessed and validated by reproducing the known applied heat flux, which had the similar peak values and variations as expected in the secondary cooling. The effects of model parameters, including the number of future time steps and the regularization parameter, were investigated with the validation trials.
The second objective of this research is to investigate and evaluate the roles of operational parameters on boiling water heat transfer occurring in secondary cooling. In order to preserve the same cooling conditions as those in HDC casting, all the experiments in this research were carried out with a set of operation conditions, i.e. the AA356 cast block was directly cut from a cast T-ingot from industry, the cooling water was used from laboratory’s water supply with an average temperature around 18°C, etc. Only two operational parameters, the cooling water flow rate and the casting speed, was investigated in this research. The effects were evaluated in the qualitative fashions by comparing the key features of the boiling curves.

Finally, a fitting technique was developed to idealize the calculated boiling curves into different functions of the surface temperatures. Those functions were summarized into a table and can be easily applied as boundary conditions to characterize the boiling heat transfer occurring in secondary cooling of HDC casting.
Chapter 4

Methodology

To accomplish the objectives of this research, a number of analysis methods and experimental techniques were applied and developed, respectively. These include an HDC secondary cooling simulator, inverse heat conduction analysis and a finite element model for direct heat conduction analysis. The simulator was designed to provide cooling conditions similar to those experienced by an HDC cast ingot during secondary cooling. The temperature history, measured during each cooling test, was later used as the input to an inverse heat conduction algorithm coupled with a finite element model to calculate the surface heat fluxes. The calculated heat fluxes were expressed as functions of the surface temperature and idealized to determine the boiling curves.

4.1 Experimental

4.1.1 Apparatus

An apparatus was designed to simulate the water spray cooling occurring in HDC casting process. An overview of the configuration of the system is shown in Figure 4.1. The sample, which is instrumented with thermocouples (TCs), is attached to a rod. That is supported by a roller system. Initially, the sample is heated in the furnace after rotating the sample to
the side. The data acquisition system is used to monitor the TCs during heating. After the desired temperature is reached, the sample is rotated back into place and the rod is mounted to a linear slide system and actuated by a motor. Once mounted, the water spray is activated and the sample is moved through the water spray ejected from the mold system for moving tests, or the sample is placed at the location where the jets can impinge at the edge of the sample and then the water spray is activated for stationary tests. The temperature history is recorded until the sample reaches \( \sim 20^\circ \) and test is finished. Compared with using the industrial HDC casting process, the lab–scale simulator is cost effective because production does not need to be interrupted, it provides more flexibility in terms of the range of operational parameters that can be investigated, and it is more accessible from the standpoint of instrumentation. By separating the secondary cooling process from the casting process, the approach used in this study avoids the handling issues and dangers associated with liquid metal.

\[ \text{Figure 4.1: Overview of the HDC simulator configuration} \]

To maintain the features of the industrial HDC water jet system, including a centrally fed array of circular water jet holes with a specific impingement angle, a section of HDC mold was supplied by Rio Tinto Alcan. With one side blocked and water fed from the other
side, water is ejected from the water jet holes whose diameter is 2.8 mm. The impingement angle is $\sim 60^\circ$, and is virtually independent of flow velocity. A picture of mold is shown in Figure 4.2.

![Figure 4.2: Mold used in experiment](image)

Three test samples were cut from an HDC cast, A356 (Al; 7\%wt Si; 0.3\%wt Mg) T-ingot. The as–cast surface of the samples, shown in Figure 4.3, exhibits a lapped texture. In this manner, the contact conditions (i.e. surface structure) were similar to the industrial HDC process conditions. During the tests, the as–cast surface was exposed to the water spray, while the other surfaces of the sample were insulated with a ceramic coating (Boron Nitride Aerosol Lubricoat from ZYP Coatings Inc.) to limit heat loss. The heat in the sample was conducted to the as–cast surface and removed by the cooling water similar to secondary cooling.

The dimensions of the samples were determined based on the following considerations:

1. In the direction corresponding to the HDC casting direction (x–direction), the length must be long enough to obtain the fully developed cooling water flow conditions;

2. The thickness of the sample in the main heat conduction direction (y–direction) should not limit the heat conducted to the cooling surface;
3. The sample must be wide enough to ensure 2–D heat transfer conditions along the mid-plane.

Additionally, the overall weight of the sample had to be limited to allow manual manipulation of the sample. Following preliminary heat transfer analysis, the test sample dimensions were defined as $150 \times 150 \times 100 \text{mm}$ in the $x$, $y$ and $z$ directions, respectively.

During a test, the sample is fixed to the rod by a plate with two bolts through it. Thin fiber insulation was sandwiched between the sample and the rod mounting plate to prevent heat loss from the sample. Mounting the sample on the rod enabled easy–handling of the hot sample. The sample, controlled by the motor, can be moved at constant speeds ranging $0 \sim 150 \text{mm/min}$.

The mold was mounted on a frame in front of the lower support roller. The distance between the as–cast surface of the sample and the surface of the mold section was adjusted to $0.5 \text{mm}$ by adjusting the roller setup. A gap was imposed to avoid heat loss through contact between the sample and the mold before entering the water spray. When disconnected from

![Figure 4.3: Sample surface condition; yellow arrow: casting direction](image)
the linear slide, the rod can pivot on the lower support roller, so that the sample could be placed into the furnace set up on the side of the apparatus. The system configuration can be adjusted to test cooling in the different spray configurations. To accommodate the configuration changes, the mold can be mounted on the top, side, or bottom of the frame to simulate water spray in different directions and the sample can be turned by 90 or 180 degrees associating with the spray configuration. The final set-up of the whole system in the top spray configuration is shown in Figure 4.4.

![Figure 4.4: The final set–up experimental system](image)

Cooling water was supplied to the apparatus from the building municipal water supply at the University of British Columbia. A valve was used to adjust the flow to a desired flow rate (up to 15 gallon per minute) and pressure (up to 40 psi). The flow rate and pressure during each test were measured with a flow meter (FLMW–1015BR from Omega Engineering Inc.) and pressure gauge (PGUF–15B–100PSI/7BAR from Omega Engineering Inc.). The cooling water quality (e.g. dissolved solids, gas content and contamination by oil) was not evaluated. As the water was not recycled and was drawn from the building supply for
each test, water quality was assumed to be constant for all tests. The water temperature was measured during commissioning of the apparatus and was found to be constant at 18°C. A chamber was used to contain the water sprayed onto the sample and direct it into a drain.

4.1.2 Instrumentation

To instrument the sample with TCs, a series of 1.6mm diameter holes to were drilled 40mm deep from the sides of the samples parallel to the cast surface and perpendicular to the casting direction. In total, 16 holes (the red and blue dots) were placed in each sample based on the drawing shown in Figure 4.5. The holes for TC 3 to TC 11 were spaced 5mm apart while the others were 10mm. The holes shown as blue dots in Figure 4.5 were drilled from the opposite side of the sample to prevent interference between the adjacent (red) thermocouples. The holes were drilled with flat bottoms to ensure that the tip of the thermocouple touched the end of the hole. The distance from the as–cast surface to the holes needed to be small to reduce the time delay in thermocouple response after surface heat flux changes, but large enough to ensure the holes did not intersect with the lapped surface structure. In this work, a distance of 5mm was used.

Figure 4.5: TCs’ location
Type E thermocouples (Model number: 304–E–MO–062 from OMEGA Engineering Inc.) were selected for use in this work because of their measurement range (\(-200^\circ C \sim 900^\circ C\)). The TCs were inserted to the ends of the holes and “locked” inside by a wire fastening system on the side of the sample shown in Figure 4.6. The wire fastening system consisted of several pairs of copper wires affixed to the side of the sample by a series of screws in line with the TC holes. The TC wires were bent parallel to the side surface at a point 5\(mm\) outside the hole. The copper wires were twisted together to pull the TC wires to the surface of the sample, which forced the TCs’ tip to touch the end of the holes. Another group of screws were used along the mid portion of the sample to bind the TCs to the sample and eliminate any strain on the TC tips. In this manner, the TCs were stationary relatively to the sample during the tests.

Figure 4.6: A glance of the wire fastening system

A DAQ device (USB-2416) from Measurement Computing was used during each test. The data from the 16 thermocouples were recorded at a rate of 5\(Hz\). At this rate, the manufacturer specifies a maximum signal noise of 0.2\(^\circ C\)[45]. Since the temperatures were used for inverse heat conduction analysis which is quite sensitive to noise, it was necessary to filter and smooth the temperature data. To maintain the features and trends of the original data, a Savitzky–Golay smoothing filter was used[46]. The measured and smoothed tem-
perature data for one example test are shown in Figure 4.7. The amplitude and bandwidth features of the curves were well preserved.

Figure 4.7: Effect of data smooth method

4.1.3 Experimental Measurements

Two types of cooling tests were performed for each spray configuration in this research: i) stationary, and ii) moving sample. The target initial temperature of the sample was 500°C. The operational parameters, water flow rate and moving speed, were scaled from the parameters used by the industry based on the sample size relative to the regular T–ingot product. In the stationary tests, the water jet impingement point was positioned at TC 16. The water flowed on the as–cast surface along the casting direction toward the TC 1 position and then flowed off the sample. The effect of cooling water flow rate was studied with these tests by applying flow rates of 30, 40 and 55 liter per minute. 40LPM is close to the flow rate
used in industry. In the moving sample tests, samples were positioned manually just before contact with the water jets. The sample was then advanced into the jets by the slide with initial water contacting the sample nearest TC 1. Different casting speeds were simulated by adopting three different linear slide speeds: 59, 122, 150mm/min, where 122mm/min is close to extraction speed employed in industry. A summary of the experimental parameters for each spray configuration is shown in the Table 4.1

<table>
<thead>
<tr>
<th>Spray configuration</th>
<th>Test type</th>
<th>Flow rate (LPM)</th>
<th>Casting speed (mm/min)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Top spray</td>
<td>Moving</td>
<td>40</td>
<td>59</td>
</tr>
<tr>
<td>or</td>
<td>Moving</td>
<td>40</td>
<td>122</td>
</tr>
<tr>
<td>Side spray</td>
<td>Moving</td>
<td>40</td>
<td>150</td>
</tr>
<tr>
<td>or</td>
<td>Stationary</td>
<td>30</td>
<td>0</td>
</tr>
<tr>
<td>Bottom spray</td>
<td>Stationary</td>
<td>40</td>
<td>0</td>
</tr>
<tr>
<td></td>
<td>Stationary</td>
<td>50</td>
<td>0</td>
</tr>
</tbody>
</table>

4.2 Inverse Heat Conduction Analysis

One of the main difficulties in developing mathematical models of solidification processes is the lack of thermophysical properties and/or boundary conditions (BCs)[47]. Given temperature information, it is possible to back–calculate this/these information. These kinds of problems are considered to be ill-posed[48]. The approach, called inverse heat conduction (IHC) analysis, can be used to calculate time and location dependent heat fluxes with careful formulation of the problem. The IHC technique numerically calculates a sensitivity matrix based on a series of runs with a heat conduction model and uses this to estimate the parameter of interest. In this work, an in–house finite element (FE) based heat condition model was used to predict the temperature evolution of a sample. This section provides a basic description of the inverse heat conduction analysis, following by validation exercises performed to ensure its accuracy.
4.2.1 Forward Heat Conduction Model

The forward heat conduction model is used to calculate the temperature field within a given domain by solving the heat conduction equation. The domain is subject to known boundary conditions and initial conditions. The general form of the heat conduction equation can be written as:

\[
\rho c_p \frac{\partial T}{\partial t} = \nabla \cdot k\nabla T + Q
\]  

(4.1)

where \(\rho\) is the density of the domain (kg/m\(^3\)), \(c_p\) is the specific heat of the domain (J/kg·K), \(k\) is the thermal conductivity of the domain (W/(m·K)), \(T\) is the nodal temperature within the domain (°C) and \(Q\) is the heat generation per unit volume or “internal heat source”. The term at LHS of Equation 4.1 represents the rate of increase of internal energy per unit volume. The first term at RHS is the heat conducted in the system per unit volume. The last term on the RHS of Equation 4.1 is set to zero. As shown in Figure 4.5, heat was assumed to be conducted in the x–direction along the length of the sample and in the y–direction toward the cooled surface. Heat flow in z–direction was considered to be negligible because of the expected even distribution of the water flow along the width of the sample during the cooling tests and the insulation of the sides of the samples. The geometry used to define the domain was a the cross–section of the sample in the x and y directions which is a square with dimensions of 150 × 150 mm. The domain was meshed with 4–node linear elements as shown in Figure 4.8. The red and blue points in Figure 4.8 are TC locations. Overall, there are 2596 nodes (yellow points in Figure 4.8) and 2488 elements in the mesh.

In this work, the as-cast surface is subjected to a heat flux varying with time and location due to the water cooling. The other surfaces of the sample are insulated by the ceramic coating. There is no volumetric heat source or sink because the sample does not undergo a phase change during the tests performed for this study, hence the last term on the RHS of Equation 4.1 is set to zero. As shown in Figure 4.5, heat was assumed to be conducted in the x–direction along the length of the sample and in the y–direction toward the cooled surface. Heat flow in z–direction was considered to be negligible because of the expected even distribution of the water flow along the width of the sample during the cooling tests and the insulation of the sides of the samples. The geometry used to define the domain was a the cross–section of the sample in the x and y directions which is a square with dimensions of 150 × 150 mm. The domain was meshed with 4–node linear elements as shown in Figure 4.8. The red and blue points in Figure 4.8 are TC locations. Overall, there are 2596 nodes (yellow points in Figure 4.8) and 2488 elements in the mesh.
thus the edges representing these surfaces were assumed to be insulated, i.e. \( q = 0 \). The thermophysical properties of A356 (the sample material) were taken from the ASM Handbooks Online[49]. These properties, shown in Table 4.2, were assumed to be independent of temperature in the analysis.

**Table 4.2:** Domain’s thermophysical properties for validation problem

<table>
<thead>
<tr>
<th>Property</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Thermal conductivity ((W/mK))</td>
<td>( k )</td>
<td>159</td>
</tr>
<tr>
<td>Heat capacity ((J/kgK))</td>
<td>( C_p )</td>
<td>963</td>
</tr>
<tr>
<td>Density ((kg/m^3))</td>
<td>( \rho )</td>
<td>2685</td>
</tr>
<tr>
<td>Thermal diffusivity ((m^2/s))</td>
<td>( \alpha )</td>
<td>( 6.15 \times 10^{-5} )</td>
</tr>
</tbody>
</table>

A program was written in Fortran to solve the 2-D transient heat conduction problem. A number of numerical methods could be used to solve this problem. In this research, the finite element method (FEM) was used. A brief outline of the equations for solving the

![Figure 4.8: Nodes within the domain](image)
transient heat conduction problem using the FEM is given in Appendix A (p.93). A detailed evolution of these formulations can be found in [50].

### 4.2.2 Inverse Heat Conduction Method

Many numerical approaches have been proposed to perform reliable IHC analysis. These methods include the sequential function specification method[48], the boundary element technique[51], the space marching technique[52] and the Monte Carlo method[53]. In this research, Beck’s sequential function specification method was adopted because it is easy to couple to the existing heat conduction code and simple to understand.

Consider the case of a flat plate subjected to a time varying heat flux at \( x = 0 \) while the remaining surfaces are insulated, i.e. \( q = 0 \), as shown in Figure 4.9. A thermocouple is embedded at \( x = E \) to record the temperature history at a time interval of \( \triangle \theta \). The aim is to calculated the heat flux, \( q(t) \), using the measured temperature data with known initial conditions and the domain’s thermophysical properties.

![Figure 4.9: Simple inverse heat conduction problem](image)

In Beck’s method, the heat flux, \( q(t) \), is discretized into a series of \( q^i \) over a measurement
interval $\Delta \theta$. The objective is to find the value of $q^i$ in each time interval, which minimizes the difference between the measured and calculated temperature. At time $t = i$, an initial guess heat flux is employed if $i = 0$ or is set to the calculated heat flux from previous time step, $q^{i-1}$, if $i > 0$. Temperature response at the sub–surface location of the thermocouple will lag or be delayed following a change in heat flux at the surface. To compensate, the heat flux guess is kept constant within a period of time, $r\Delta \theta$, where $r$ is a number of future time steps, usually from 3 to 10. The careful selection of $r$ can help stabilize the estimation procedure. Solved as a forward heat conduction problem, the predicted temperature based on the applied heat flux guess $q^i$ is obtained at $t = i + r\Delta \theta$. The sum of the squared difference between the measured and predicted temperatures is then calculated by equation:

$$ F(q^i) = \sum_{n=1}^{L} (T^{i+r\Delta \theta} - Y^{i+r\Delta \theta})^2 \quad (4.2) $$

where $Y^{i+r\Delta \theta}$ is the measured temperature at time $i + r\Delta \theta$, $T^{i+r\Delta \theta}$ is the calculated temperature at time $i + r\Delta \theta$ based on $q^i$, $L$ is the number of the heat flux to be predicted, in this simple case, $L = 1$. The function $F$ is called the objective function and is minimized through iterations. In an iteration, the heat flux is increased by $\Delta q$, which is calculated by:

$$ \Delta q = S^{-1} \Delta T^{i+r\Delta \theta} \quad (4.3) $$

where $S$ is the sensitivity coefficient and defined as the temperature response at the measurement point with respect to a unit change in heat flux. For this simple case of one thermocouple and one unknown heat flux, the temperature used to calculate the sensitivity coefficient is defined using the initial heat flux guess to determine the “base” temperature. Then the heat flux guess is increased by multiplying by $1 + \varepsilon$, where $\varepsilon$ is a small value, i.e. 0.001. The forward heat conduction model is used again to obtain the predicted temperature for
this increased heat flux. The sensitivity coefficient, \( S \), can then be calculated by:

\[
S = \frac{\partial T}{\partial q} = \frac{T(q + \varepsilon q) - T(q)}{\varepsilon q}
\]  

(4.4)

Once the difference between the measured and predicted temperatures satisfies the tolerance criteria, i.e. \( |\Delta T_i + r\Delta \theta| \leq T_{cr} \), an estimate of \( q^i \) has been obtained. At this point, time is increased by \( \Delta \theta \), \( q^i \) is used as the initial heat flux guess for the next time step and the process is repeated for \( q^{i+1} \), yielding the full history of \( q(t) \).

It is important to note that there are two time intervals used in the IHC analysis: one is the time step, \( \Delta t \), for the heat conduction model; and the other is the measurement time step, \( \Delta \theta \). A small \( \Delta t \) is necessary to improve the accuracy of heat transfer calculation. But if too small, it makes inverse calculation unstable[48]. \( \Delta \theta \) is limited by the DAQ device, and increasing it may result in insufficient data to capture the dynamics of the temperature change. Usually, \( \Delta \theta = k\Delta t \), where \( k \) is an integer, such as 20.

For the more general case of interest in this work, the unknown heat flux is not only a function of time but also a function of space, i.e. \( q(x, t) \). \( J \) components of heat flux in space can be determined with measurements at \( L \) locations if \( J \leq L \). In our case, \( J \) was equal to \( L \). The objective function, \( F \) in Equation 4.5, becomes the sum of squared differences between the thermocouple measurements and the model predictions at all thermocouple locations.

\[
F(q^i) = \sum_{n=1}^{L} (T_n^i - Y_n^i)^2
\]  

(4.5)

A perturbation method[54] was used to calculate the sensitivity matrix. First, a guess heat flux, \( q^i \), is applied across the entire surface and the FE model is used to calculate a baseline temperature at all thermocouple locations, \( T_{0n}^i \), where \( n = 1 \ldots L \). Similar calculation are performed with the heat flux increased by multiplying by \( (1 + \varepsilon) \) at each measurement point, i.e. \( T_{1n}^i \), where \( n = 1 \ldots L \) calculated based on the heat flux increment at thermocouple
location 1. The heat flux was assumed to vary linearly between measurement points as shown in Figure 4.10. The sensitivity coefficients, expressed in Equation 4.6, are the ratio of the temperature differences (perturbated–base line) at each measurement point and the increment of the heat flux. Figure 4.10 shows the sequence of model runs needed to build up the sensitivity matrix.

\[
S_{n \times n} = \begin{pmatrix}
\frac{T_{11} - T_{01}}{\varepsilon q_1} & \frac{T_{12} - T_{02}}{\varepsilon q_1} & \cdots & \frac{T_{1n} - T_{0n}}{\varepsilon q_1} \\
\frac{T_{21} - T_{01}}{\varepsilon q_2} & \frac{T_{22} - T_{02}}{\varepsilon q_2} & \cdots & \frac{T_{2n} - T_{0n}}{\varepsilon q_2} \\
\cdots & \cdots & \ddots & \cdots \\
\frac{T_{n1} - T_{01}}{\varepsilon q_n} & \frac{T_{n2} - T_{02}}{\varepsilon q_n} & \cdots & \frac{T_{nn} - T_{0n}}{\varepsilon q_n}
\end{pmatrix}
\]  

(4.6)

\[
\text{Figure 4.10: Perturbation method for building up sensitivity matrix}
\]

A zeroth–order Tikhonov regularization method was used to reduce oscillations and further stabilize the calculated heat fluxes. A detailed discussion of this regularization method can be found in [48]. Using this method, the increment of \( q^i \) is calculated by:

\[
\Delta q^i = A^{-1} \Delta \vec{T}^{i+r \Delta \theta}
\]  

(4.7)

\[
A = S_{n \times n} + \alpha I
\]  

(4.8)

where \( \alpha \) is the Tikhonov regularization parameter, which is typically set to a very small number with a magnitude of \( 10^{-8} \) or less. For a related analysis on the primary cooling of
HDC casting of A356, $2 \times 10^{-9}$ was used[55]. $I$ is the $n \times n$ identity matrix.

The updated heat flux guess for the next iteration can be obtained by adding $\Delta q$ to the initial guess of heat flux. Iterations are performed until the objective function is minimized at each thermocouple location. Equation 4.9 is temperature criterion used to determine whether the calculated heat fluxes are accurate, where $T_{cr}$ is called the criterion temperature. The use of a large value of a criterion temperature may reduce the accuracy of the calculation, but if it is too small, long calculation times are incurred without appreciably improving the accuracy. In this work, $T_{cr}$ was set to 0.05°C.

$$\frac{1}{L} \sum_{n=1}^{L} (T_{n}^{i+r\theta} - Y_{n}^{i+r\theta})^2 < T_{cr}$$ (4.9)

The inverse calculation procedure described in this section is summarized in the flow chart shown in Figure 4.11.

4.3 Validation

4.3.1 Validation of Heat Conduction Model

The FE model was validated by comparing predicted temperatures to those calculated with an analytical solution to a 1–D transient heat conduction problem. Since the program written for this study solves 2–D problems, a small rectangular domain ($10mm \times 1mm$) was defined and meshed with $1mm$ elements. The domain was subjected to a constant heat flux, $q$, on one narrow edge. The model parameters and thermophysical properties used are provided in Table 4.3 and 4.2.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Surface heat flux ($W/m^2$)</td>
<td>$q$</td>
<td>$-1 \times 10^6$</td>
</tr>
<tr>
<td>Initial Temperature ($^\circ C$)</td>
<td>$T_0$</td>
<td>580</td>
</tr>
<tr>
<td>Time step (s)</td>
<td>$dt$</td>
<td>0.01</td>
</tr>
</tbody>
</table>

Table 4.3: FEM model parameters for validation problem
The analytical solution for the temperature in a semi–infinite solid subjected to a constant surface heat flux is:

$$T(x,t) = T_0 + \frac{2q(\alpha t/\pi)^{1/2}}{k} \exp\left(-\frac{x^2}{4\alpha t}\right) - \frac{qx}{k} \text{erfc}\left(\frac{x}{2\sqrt{\alpha t}}\right)$$  \hspace{1cm} (4.10)

where $x$ is the depth from the edge where the heat flux is applied (m), $t$ is the time (s) and $\alpha$ is the thermal diffusivity of the domain ($m^2/s$). The comparison between the analytical solution and the numerical predictions at 1mm and 5mm from the heated edge is shown in Figure 4.11.
Figure 4.12: The excellent agreement confirms the accuracy of FE model.

Figure 4.12: Comparison of the analytical and FEM solutions for a 1–D heat conduction problem

4.3.2 Validation of IHC Implementation

The IHC implementation was validated in two stages: In the first stage, the heat conduction model for a specified domain with known heat fluxes was employed to generate temperature data; This temperature history was then used in the second stage as measured temperatures to calculate the applied heat fluxes. The known and calculated heat fluxes were then compared to assess the accuracy and stability of the IHC algorithm.

To make this validation exercise relevant to the HDC cooling tests, the cross–sectional geometry of the sample used in the cooling experiments was used. The thermophysical properties summarized in Table 4.2 were input to the model. The edge representing the as–cast surface was exposed to a known heat flux defined by Equation 4.11,

\[ q(x,t) = -5 \cdot 10^6 \times \sin^2 \left( \frac{3\pi}{2} \cdot \frac{t}{5} \right)(0.7 + 5x - 20x^2) \]  

(4.11)
where \( t \) is the time \((s)\), and \( x \) is the location on the as–cast surface \((m)\). The other surfaces were insulated. A time step equal to 0.05s was used for both the forward and inverse calculations. The initial temperature of the domain was set to 500°C.

Initially, the temperature field was calculated with the model. Temperatures at the TC locations were used in the inverse calculation, assuming that there were no measurement errors. The comparison between applied and calculated heat fluxes is shown in Figure 4.13 for three TC locations (TC 1, TC 8, TC 16). The calculated heat fluxes accurately reproduce the applied ones. The number of future time steps and the regularization parameter employed in this calculation have no effect on the results, and were assumed to be zero[12].

![Figure 4.13: Comparison of predicted and applied heat fluxes with no measurement errors, FTS= 0 and \( \alpha = 0 \)](image)

However, noise cannot be ignored even if it is small, especially for inverse analysis, which is quite sensitive to measurement errors. To study the effects of temperature history mea-
measurement errors on the inverse calculation, the temperature calculated with the forward version of the model were augmented with random errors. Figure 4.14 gives an example of the input temperature with different levels of noise at \( x = 0.135m \). Three ranges, ±1°C, ±3°C and ±6°C, were applied. For the maximum range (±6°C) example, the relative error represents ±1.2% at higher temperatures (~500°C) to ±6% at lower temperatures (~100°C).

![Input temperature with different noise at x=0.135m](image)

**Figure 4.14:** Input temperature with different noise at \( x = 0.135m \)

With noise imposed on the input temperatures, convergence could only be achieved by enabling the use of future time steps. The number of the future time steps was determined by trial and error. Figure 4.15 shows the effect of measurement noise on the calculated heat flux. As the measurement noise increases, fluctuations at the beginning of the calculation become larger and more iterations are needed for convergence. The trend of the heat flux variation was adequately captured but the use of future time step results in a time shift in the curve and an under prediction ~7% of the peak heat flux. These problems were also observed by Xu[57] and are attributed to the use of future time steps.
Heat Flux (MW/m²)

Input heat flux at x=0.135m
Calculated heat flux at x=0.135m, no noise
Calculated heat flux at x=0.135m, ±1°C noise
Calculated heat flux at x=0.135m, ±3°C noise
Calculated heat flux at x=0.135m, ±6°C noise

Figure 4.15: The effect of measurement error, FTS= 3 and α = 0

Figure 4.16 shows the effect of using different numbers of future time steps on the calculated heat flux with a fixed noise level (±1°C). It is evident that increased numbers of future time steps reduce fluctuation at the beginning of the calculation, especially when the number of future time steps increases from 3 to 5. When the number of future steps is even higher, 7 or 10, less effect on initial fluctuations is observed and the underestimation of the peak heat flux are more obvious. It was concluded that using 5 future time steps yields adequate results with limited initial fluctuations.

The regularization parameter (α) affects the underestimation of the calculated heat flux. The calculated heat flux for different α are shown in Figure 4.17. The most accurate peak heat fluxes were obtained when α = 10⁻¹¹, however, the calculated heat flux at lower magnitudes was over estimated. In this work, the peak heat flux is a critical parameter in the cooling process. As a result, α = 10⁻¹¹ was chosen for this work. Figure 4.18 shows the comparison of the input or direct calculated temperature and the temperature calculated by re–applying the calculated heat fluxes (FTS= 5 and α = 10⁻¹¹). It is observed that, using

44
the calculated heat flux, the temperature at the discrete points is accurately reproduced.

![Figure 4.16:](image1)  
**Figure 4.16:** Calculated heat fluxes for different numbers of future time steps (noise = ±1°C and $\alpha = 0$)

![Figure 4.17:](image2)  
**Figure 4.17:** Calculated heat fluxes for different regularization parameters (noise = ±3°C and $FTS = 5$)
Figure 4.18: The temperature calculated by applying the predicted heat fluxes ($FTS = 5$ and $\alpha = 10^{-11}$) vs. the input temperature

4.3.3 Summary

Example calculations were performed to show that the proposed modeling approaches (forward and inverse) are accurate and capable of calculating the heat flux from a measured temperature field. An appropriate number of future time steps is needed to dampen the effect of measurement errors. With 5 future time steps, the errors in the calculated heat fluxes can be controlled to within 7% when the noise level is $\pm 1^\circ$C. A regularization parameter is necessary to capture the peak value of the heat flux. An adequate result can be obtained with $\alpha = 10^{-11}$, while matching the trend of the heat flux variation.
Chapter 5

Results and Discussion

The temperature measured during the experiments to simulate secondary cooling during HDC casting will be presented first in this chapter. This will be followed by the calculated heat fluxes, which are then idealized into representative boiling curves. Throughout the chapter, the results are discussed in terms of their similarities and differences related to the relevant process parameters investigated in this study (cooling water flow rate, withdrawal rate and water spray configuration).

5.1 Test Results

5.1.1 Water Flows

During the tests, different flows formed on the surface of the samples and were observed to depend on the flow rate. However, once the cold water touched the hot sample during the tests, a large amount of steam was generated. Considering the safety issues, the tests needed to be carried out in the chamber. This fact makes the direct observation of the flow profiles during the tests difficult. As a result, pictures of the flow phenomena were taken on cold stationary samples to document conditions.

Prior to hitting the as–cast surface, each jet can be identified. The diameter of each jet
enlarges with increasing distance from mold. As the jet impinges on the as–cast surface, the change in momentum results in an even larger diameter causing overlap between the jets. The jets either coalesce into a water curtain at lower flow rates or bounce off (eject) and interact with adjacent jets at higher flow rates. Pictures of these two conditions for the top spray configuration are shown in Figure 5.1.

Figure 5.1: Top spray with different flow conditions for different flow rates

Figure 5.2 shows a side view of the cooling water striking the top surface of a sample. The sharp change in momentum direction results in a hydraulic ejection immediately after the impingement point. For low velocity jets (Figure 5.2a), the water is then pulled back to the surface by gravity, resulting in a water curtain along the sample surface. The water then falls off from the sample at its edge. When the flow rate is increased to 30 LPM, the ejection phenomenon becomes more obvious. A portion of the cooling water flows along the sample surface in the casting direction at a very high velocity and the remainder is completely ejected.

For the bottom spray configuration, shown in Figure 5.3, the jets hit the as–cast surface and maintain contact along the surface in the casting direction due to surface tension effects and momentum. Eventually, the water reaches the end of the sample and drops off. No water ejection was observed at a flow rate of 10 LPM. When the flow rate increases to 30 LPM, ejection occurs immediately after the water impinges on the surface. Some water travels along the sample in the casting direction and some falls off the as–cast surface.
In the side spray configuration, shown in Figure 5.4, after hitting the as-cast surface, the water flows in the casting direction, but eventually changes direction due to gravity. The water travels along a parabolic path (Figure 5.4c). For the low flow rate conditions, this results in no water contact at different distances from mold, based on the height along the sample surface. From the top down view, shown in Figure 5.4a, no ejection is observed for low flow rate conditions (10 LPM). When the flow rate increases to 30 LPM, the cooling water flows along the entire surface of the sample in the casting direction as shown in Figure 5.4d. Compared with the low flow rate, the ejection is obvious with flow rate of 30 LPM (Figure 5.4b).
Figure 5.4: Side spray flow profiles for different flow rates

5.1.2 Temperature

An example plot of the original measured temperatures for a stationary test in the top spray configuration is given in Figure 5.5. Since this study is concerned with boiling water heat transfer, only the data between two vertical dashed lines were considered in the IHC analysis. Before performing IHC analysis, the data was smoothed with a Savitsky–Golay filter as described in the section 4.1.3. An example plot of the temperature data after limiting the time and smoothing is shown in Figure 5.6. By limiting the time considered, the temperature variation at each TC location can now be assessed. The effect of cooling water flow rate, motion and water spray configuration on temperature can now be compared and discussed.
Figure 5.5: Example of the measured temperature data before smoothed for the top spray configuration with stationary sample and $Q = 40LPM$

Figure 5.6: Example of the measured temperature data for top spray configuration with stationary sample and $Q = 40LPM$ after smoothing and limiting the time.
5.1.3 Effect of Oxidization

The repeated heating of each sample resulted in progressive oxidation. The amount of oxidation taking place during each test was not quantified. In order to assess the effect of oxidation on boiling water heat transfer occurring in tests, repeat tests with the same spray configuration (stationary sample, top spray, 40LPM flow rate) were performed during the testing campaign. The 2nd and 12th tests were repeats of the 1st test. The cooling rates and temperature variation from each of these tests, shown in Figure 5.7, were selected to study the effect of the sample oxidation. This analysis also provides a means of assessing the repeatability of this testing technique.

![Figure 5.7: Cooling rates comparison from the stationary sample tests (Test 1, 2 and 12) with top spray and 40LPM flow rate](image)

In Figure 5.7, it is evident that similar cooling rates are observed for all three tests. In Test 1, where less oxidation was expected, a higher peak cooling rate occurs, however, the
subsequent tests are within 5%. The cooling rates and the temperature variations from Test 2 and 12 are similar, which indicate that the oxidization effect on heat transfer after first test is not significant. As a result, the effect of the sample oxidization on boiling water heat transfer occurring during the tests was neglected in this work. This also ensures that the testing technique is repeatable and the new instrumented sample is not required for every single test.

5.1.4 Effect of Flow Rate

The effect of cooling water flow rate was assessed by comparing the temperature data from the stationary tests. In these tests, the water jets first contact the sample surface at a location nearest to TC 16. Water flows along the surface passing the other thermocouple locations in descending order toward TC 1. For convenience, the sample was divided into regions based on the observed thermal response. The regions and corresponding TC locations are:

- Impingement zone (IZ): TC 15 and TC 16
- Forced flow zone (FFZ): TC 1 to TC 14

Since the temperature curves in each region have similar features, only temperature data from one TC in each region was selected for comparison in the following discussion. In this section, the TC’s selected for comparison are: TC 16 (IZ) with red lines; TC 8 (FFZ) with green lines. It is important to point out that, due to end effects, the temperature response of TC 1 and TC 2 is slightly different from those TCs located in the middle of the sample in the FFZ. The discussion regarding to this difference are included in the following sections.

Top Spray

The temperature data for the top spray configuration is shown in Figure 5.8. In the IZ, identical cooling curves were observed for all flow rates until temperatures below \( \sim 300^\circ\text{C} \).
At $\sim 300^\circ$C, the cooling rate for the 10LPM flow rate decreases. After 15 seconds, the sample temperature in the IZ has cooled to $\sim 60^\circ$C for the 30LPM, 40LPM, and 50LPM flow rate tests, whereas the temperature for the 10LPM has reached $\sim 125^\circ$C.

![Figure 5.8: Measured temperatures for top spray configuration at TC locations 8 & 16 and flow rates of 10, 30, 40 & 50LPM](image)

In the FFZ locations shown in Figure 5.8, similar patterns of cooling were observed. For the tests with flow rates of 10LPM and 30LPM, prior to a high cooling rate with water spray cooling, an intermediate cooling rate is observed initially. The formation of a vapor film on the as-cast surface has been speculated to be the cause of this intermediate cooling rate. This vapor film condenses on the as-cast surface and prevents the surface from the cooling water. The heat has to be conducted through the vapor film before removed by the cooling water. Alternatively, the tests with flow rates of 40LPM and 50LPM exhibit a high cooling rate after a delay which is caused by the ejection. It is interesting to note that the highest cooling rate in the FFZ occurs in the tests with the highest flow rate (50LPM) but that cooling is also the most delayed in this test. This can be explained by the intense
ejection expected for the test with a flow rate of 50LPM. Consistent with this trend, the test with the lowest flow rate cools to temperatures below 200°C in the FFZ in the shortest time.

**Side Spray**

The temperature data measured in the tests with the side spray configuration at the locations previous identified for different flow rates is shown in Figure 5.9. For the TC location corresponding in the IZ, the measured temperatures for the tests with flow rates of 40LPM and 50LPM show similar high cooling rates. The slowest cooling rate occurs for a flow rate of 30LPM, while the 10LPM flow rate exhibits a high initial cooling rate which decreases when the temperature drops below 250°C. Within 15 seconds, the temperature reaches ~ 80°C for the tests with flow rates of 40LPM and 50LPM, but is 175°C and 125°C for the flow rates of 10LPM and 20LPM respectively.

![Figure 5.9: Measured temperatures for side spray configuration at TC locations 8 & 16 and flow rates of 10, 30, 40 & 50LPM](image)

Similar cooling curves were observed in FFZ locations for flow rates of 30LPM, 40LPM,
and 50LPM. The test with 10LPM flow rate exhibited a significantly lower cooling rate because of decreased water contact with the as-cast surface. In Figure 5.4c, no water contact was observed for the tests with 10LPM close to the end of the sample. Correspondingly, the cooling rate with 10LPM flow rate is extremely low at locations of TC 1 and TC 2 shown in Figure 5.10. The surface was cooled mainly by the natural convection to air instead of boiling water heat transfer.

![Temperature vs Time Graph](image)

**Figure 5.10:** Measured temperatures for side spray configuration at all TC locations and flow rates of 10LPM

**Bottom Spray**

The measured temperature data for the bottom spray configuration at the two TC locations for different flow rates is shown in Figure 5.11. In the IZ, similar cooling curves were observed for all four flow rates until the temperature cools below 200°C. After 20 seconds, the highest flow rate (50LPM) cools the sample to the lowest temperature (∼30°C).
Figure 5.11: Measured temperatures for bottom spray configuration at TC locations 8 & 16 and flow rates of 10, 30, 40 & 50LPM

Similar cooling curves were observed at FFZ locations for flow rates of 30LPM, 40LPM, and 50LPM. The cooling rates for each of these flow rates are nearly identical. In the tests with 10LPM flow rate, less water remains in contact with the as-cast surface with increasing distance from the mold, thus this test exhibits a lower cooling rate compared with the other three flow rates.

5.1.5 Effect of Sample Speed

In this section, the temperatures measured during moving sample tests are compared in each water spray configuration. In these tests, the IZ and FFZ zones were not stationary relative to the sample during the tests. The surface of the sample closest to TC 1 was the first region to be cooled as the sample moved into the water spray. Cooling occurred on the surface near the other TCs progressively as the sample moved into the spray. The blue, green, and red lines represent moving speeds of 150mm/min, 122mm/min, and 59mm/min.
respectively. The plots for each spray configuration are shown in Figures 5.12 to 5.14.

**Figure 5.12:** Measured temperatures for moving sample tests in the top spray configuration

**Figure 5.13:** Measured temperatures for moving sample tests in the side spray configuration
Similar features are observed in the cooling curves for all three spray configurations. As speed increases, the temperature drop occurs earlier with a higher cooling rate. In Figure 5.14, at TC 8, the cooling curves for moving speeds of 122 mm/min and 150 mm/min overlap, which suggest that, when increased from 122 mm/min to 150 mm/min, the moving speed has little effect on the heat transfer.

![Figure 5.14: Measured temperatures for moving sample tests in the bottom spray configuration](image)

**5.1.6 Effect of Spray Configuration**

The effect of the water spray configuration was evaluated with the data from the stationary tests with different flow rates. Figure 5.15 to 5.17 compare the measured temperature in each spray configuration for different flow rates. The red, green, and blue lines represent top, side and bottom spray configuration, respectively.
For the 50LPM flow rate (Figure 5.15), there is little difference in the cooling curves for all spray configurations in the IZ (TC 16) until the temperature is below ~200°C. At this temperature, the cooling rate decreases for the top and side spray configuration tests. In FFZ (TC 8), the cooling curves for all spray configurations are similar. The sharp temperature decrease occurs the earliest for the side spray configuration followed closely by the bottom spray configuration. However, the top spray configuration was delayed by almost 20 seconds.
Figure 5.16: Measured temperatures for stationary tests at two TC locations for each spray configuration ($Q = 40\text{LPM}$)

Figure 5.17: Measured temperatures for stationary tests at two TC locations for each spray configuration ($Q = 10\text{LPM}$)
There does not appear to be an effect of spray configuration for the 40\textit{LPM} flow rate. When the flow rate is decreased to 10\textit{LPM}, a significant difference is observed in both the IZ and FFZ measured temperature data. In the IZ, the cooling curves diverge when the temperature is below 300$^\circ$C. This is a higher temperature than the divergence temperature for 50\textit{LPM}. After the temperatures diverge, the cooling rate is the slowest for the side spray configuration and highest for the bottom spray configuration. In the FFZ, the sharp temperature drop occurs first for the top spray configuration for 10\textit{LPM} flow rate compared to the last for 50\textit{LPM} flow rate. This can be explained by that, with flow rate of 10\textit{LPM}, no water ejection occurred during the cooling, all sprayed water flowed along the top surface; with flow rate increased to 50\textit{LPM}, a huge amount of water was ejected and less water remained in touch with the top surface, resulting the delay in the temperature dropping.

5.2 Calculated Heat Flux and Boiling Curves

The IHC analysis technique described in Chapter 4 was applied to calculate heat fluxes from the measured temperature data. The set of temperature data from the top spray configuration stationary test with flow rate of 40\textit{LPM} (refer to Figure 5.6), was used for an example calculation. Following the example, the calculated heat fluxes are expressed as boiling curves, and the key features in the different regimes of boiling water heat transfer are discussed and used to highlight the effects of flow rate, moving speed and spray configuration.

5.2.1 Example Calculation

Based on the validation cases presented in Chapter 4, the IHC parameters used for the heat flux calculations are shown in Table 5.1. These parameters were used for all tests calculations.
Table 5.1: Calculation parameters

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Symbol</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Number of future time steps</td>
<td>FTS</td>
<td>5</td>
</tr>
<tr>
<td>Measurement time interval (s)</td>
<td>$\Delta \theta$</td>
<td>0.2</td>
</tr>
<tr>
<td>Model time step (s)</td>
<td>$dt$</td>
<td>0.01</td>
</tr>
<tr>
<td>Regularization parameter</td>
<td>$\alpha$</td>
<td>$1 \times 10^{-11}$</td>
</tr>
</tbody>
</table>

Initially, heat flux was calculated considering the temperature data from all 16 TCs together. Figure 5.18 shows the heat fluxes (vs. time) calculated using the input temperature data plotted in Figure 5.6. The calculated heat fluxes were not the results expected because of the presence of “positive” heat fluxes at some TC locations. This indicates that, in order to satisfy the numerical method, the sample was heated at various times at the different locations. In reality, the sample was not heated at any time and should only exhibit cooling due to the spray water. Upon closer inspection, the heat flux at specific downstream locations becomes positive and mirrors the upstream location to offset the rapid temperature decrease when cooling water accelerates the cooling, i.e. TC 8 & 9 and TC 5 & 6. Considering the magnitudes of the calculated heat flux, the peak heat flux at location of TC 9 is $\sim 3.8 \times 10^7$, which is 5 to 10 times higher than previous results for VDC casting listed in Table 2.1. It is important to point out that the heat fluxes calculated with the IHC technique reproduce the measured temperature history within $0.001^\circ$C. The fact that the temperature history is accurately reproduced by a clearly erroneous heat flux prediction demonstrates the ill–posed nature of the inverse heat conduction problem: in IHC problem, the answers that satisfy the numerical method are not unique.

In order to reduce the numerical “interference” between the 16 TCs, the distance between thermocouples was increased. This was done by separating the 16 TCs into four groups, summarized in Table 5.2. The data from TC 5, 7, 9 and 11 were ignored because it is difficult to separate them into the desired distance and their adjacent TCs can provide similar information.
Table 5.2: TC calculation groups

<table>
<thead>
<tr>
<th>Group number</th>
<th>TC number</th>
<th>Minimum distance</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>1, 6, 13</td>
<td>0.05m</td>
</tr>
<tr>
<td>2</td>
<td>2, 8, 14</td>
<td>0.05m</td>
</tr>
<tr>
<td>3</td>
<td>3, 10, 15</td>
<td>0.04m</td>
</tr>
<tr>
<td>4</td>
<td>4, 12, 16</td>
<td>0.04m</td>
</tr>
</tbody>
</table>

The heat flux calculation was performed with the measured temperature data for each TC group separately. The calculation parameters, initial conditions and exterior boundary conditions were the same for each group. The calculated heat fluxes at each TC location were combined and are shown in Figure 5.19.

Figure 5.18: Calculated heat fluxes based on IHC analysis all 16 TCs (temperature data shown in Figure 5.6)
Figure 5.19: Calculated heat fluxes with grouped TCs for the top spray configuration with stationary sample and $Q = 40\text{LPM}$

Figure 5.20: Modified heat fluxes with grouped TCs for the top spray configuration with stationary sample and $Q = 40\text{LPM}$
The calculated heat fluxes in Figure 5.19 show peak values in line with expectations, but continue to exhibit “positive” heat fluxes at high temperatures prior to the water spray cooling effect, which should not happen in reality. However, compared to the previous results shown in Figure 5.18, these positive heat fluxes are small with peak values up to $0.5 \times 10^6$ and last less than 10 seconds. The positive heat flux events occur prior to water spray cooling at each location. In order to eliminate these unreasonable values, only the negative (or cooling) portion of the heat flux curves were considered. This was done by giving constraint to the calculated heat fluxes in the IHC model. The modified result is shown in Figure 5.20.

The accuracy and impact of calculating the heat fluxes in group and modifying the calculated values was assessed by applying the final modified heat flux curves at all TC locations in a forward heat conduction model to calculate the temperature at each TC location. The comparison of the calculated temperatures and the measured temperatures at each TC location is shown in Figure 5.21. Good agreement is observed considering the calculation technique. The modified heat fluxes applied to the model adequately reproduce the temperature variation, indicating a satisfactory calculation method.

With the developed approach, the surface temperatures at locations corresponding to the nearest TC locations can be obtained. The absolute values of the calculated heat fluxes were plotted as functions of surface temperatures at each TC location to determine the boiling curve. An example of the calculated boiling curves is shown in Figure 5.22.
Figure 5.21: The comparison of the measured and calculated temperatures with the modified heat flux for top spray configuration with stationary sample and $Q = 40LPM$

Figure 5.22: Calculated boiling curves for top spray configuration with stationary sample and $Q = 40LPM$
5.2.2 Effect of Flow Rate

The calculation method developed with the example data set was used to calculate boiling curves for each test. The boiling curves developed with the temperature data from the stationary tests were used to assess the effect of flow rate in each spray configuration. The temperatures at two TC locations, representing the IZ (TC 16–red lines) and FFZ (TC 8–green lines), are plotted for comparison based on spray configuration.

Top Spray

The boiling curves for the top spray configuration at different flow rates are shown in Figure 5.23. Film boiling did not occur in the IZ for all four flow rates because water jets disturb the evolving vapor and do not allow a stable vapor blanket to form. Transition boiling occurs immediately and the heat flux increases with decreasing surface temperature to the CHF. For the 50LPM flow rate, the CHF exceeds 5.5MW/m² at a surface temperature of ~150°C whereas the CHF is ~4MW/m² at temperature ~200°C for a flow rate of 10LPM. After reaching its peak, the heat flux gradually decreases until the surface temperature drops below 50°C. The heat flux rebounds a little bit at this temperature before decreasing to lower values. The transition from nucleate boiling to forced convection occurs during this rebound.

In the FFZ region, film boiling is observed for the low (10LPM and 30LPM) flow rate tests. The Leidenfrost points are around 370°C and 420°C at 10LPM and 30LPM flow rates, respectively. The heat flux reaches a local minimum before increasing to the CHF. The maximum CHFs occurred in the 40LPM flow rate test. Overall, the CHFs in FFZ are lower than the one in IZ. The transition from nucleate boiling to forced convection occurs at temperature near 100°C in the FFZ, which is higher than the transition temperatures in the IZ.
Side Spray

The boiling curves for the side spray configuration at different flow rates are shown in Figure 5.24. As with the top spray configuration, film boiling does not occur in the IZ. Similar boiling curves are observed for flow rates of 40LPM and 50LPM in the IZ. The CHF is \( \sim 0.3 \text{MW/m}^2 \) higher for the test with a flow rate of 50LPM (\( \sim 5.1 \text{MW/m}^2 \)) versus the 40LPM test (\( \sim 4.9 \text{MW/m}^2 \)). For a flow rate of 30LPM, the CHF is reached at 300°C, which is the highest temperature among the four curves. After CHF is reached, the heat flux decreases as the temperature drops below 100°C. Little heat flux rebound is observed at low temperatures, which is distinct from the top spray configuration.

No film boiling occurs for all flow rates in the FFZ. Similar to the temperature curves in Figure 5.9, the boiling curves for tests with flow rates of 30LPM, 40LPM and 50LPM are quite similar in the FFZ (TC 8). With a flow rate of 10LPM, no water contact from location at TC 4 was observed in the test, shown in Figure 5.4, resulting in no boiling water heat.
transfer at these locations (TC 1 to TC 4 in Figure 5.25). Thus the CHFs, $2 \sim 2.5 MW/m^2$, is much lower than the other three results for higher flow rates at TC 8.

**Figure 5.24:** Boiling curves for side spray configuration with different flow rates

**Figure 5.25:** Boiling curves for side spray configuration with a flow rate of 10LPM
Bottom Spray

In Figure 5.26, the boiling curves for bottom spray configuration at different flow rates are shown. No film boiling is observed in the IZ. Nearly identical boiling curves in the IZ are observed for the tests with flow rates of 30LPM and 40LPM when the surface temperature is above 100°C. The highest CHF (∼5MW/m²) occurs with the highest flow rate (50LPM). Consistently, with the lowest flow rate (10LPM), the CHF is just above 3.5MW/m². The temperatures for CHFs (∼200°C) are quite similar for flow rates of 30LPM, 40LPM and 50LPM, whereas, for a 10LPM flow rate, the CHF occurs at a slightly lower temperature (∼170°C).

Figure 5.26: Boiling curves for bottom spray configuration with different flow rates

In the FFZ, no film boiling is observed for all four flow rates. As the flow rate decreases from 50LPM to 30LPM, the heat flux decreases but the boiling curves are quite similar. However, when the flow rate is as low as 10LPM, the heat flux drops significantly (CHF ∼2MW/m²). This can be explained by the flow conditions shown in Figure 5.3a. For a 10LPM flow rate, a significant amount of cooling water falls off the sample before reaching
the location at the end of the sample (i.e. TC 1). Thus less water is available for the boiling water heat transfer resulting in less cooling in the nucleate boiling regime.

Summary

In all spray configurations and flow rates, film boiling does not occur in the IZ. In the FFZ, film boiling only occurs in the top spray configuration at low flow rates ($10LPM$ $\sim$ $30LPM$). The minimum heat flux at Leidenfrost point with a flow rate of $10LPM$ is between $0.5MW/m^2$ and $0.8MW/m^2$, which is comparable to the range of $0.42 \sim 0.71MW/m^2$ reported by Caron[12] for VDC casting.

The average value of the CHFs in the IZ and FFZ for different flow rates and spray configurations are summarized in Table 5.3. Compared with the data reported for the secondary cooling of VDC casting (Table 2.1), the calculated results are in a comparable range.

Table 5.3: Summary of CHFs in the IZ and FFZ with stationary tests ($MW/m^2$)

<table>
<thead>
<tr>
<th>Configuration</th>
<th>Flow rate</th>
</tr>
</thead>
<tbody>
<tr>
<td></td>
<td>10LPM</td>
</tr>
<tr>
<td></td>
<td>IZ</td>
</tr>
<tr>
<td>Top spray</td>
<td>4.2</td>
</tr>
<tr>
<td>Side spray</td>
<td>3.5</td>
</tr>
<tr>
<td>Bottom spray</td>
<td>3.6</td>
</tr>
</tbody>
</table>

5.2.3 Effect of Moving Speed

The boiling curves developed with temperature data from the moving sample tests are compared for each spray configuration in Figures 5.27, 5.28, and 5.29. Similar boiling curves are observed at either locations of TC 1 or TC 8 for each spray configuration.

For TC 1 which enters the water spray first, no film boiling is observed, transition boiling occurs immediately when the cooling water impinges on the as–cast surface. A higher moving speed leads to higher CHF for all three spray configurations. It is known that the
nucleate boiling regime is controlled by the dynamic of the bubble growth and the bubble detachment, a higher sample surface speed could help the bubble growth and improve the bubble motion, resulting in higher CHF. The CHFs range from $3 \sim 3.5 MW/m^2$ and are reached at surface temperatures of $\sim 200^\circ C$.

The surface of the samples near TC 8 are first cooled by the natural convection to air prior to entering the water spray. Thus, very low heat fluxes are observed in the beginning of each test. As the surface of the sample near TC 8 enters the spray, transition boiling takes place without the occurrence of film boiling. Increased moving speed results in higher CHFs for all spray configurations. Because of the initial convective cooling, boiling water heat transfer starts at a lower temperatures ($\sim 450^\circ C$). The CHFs, ranging from $2 \sim 2.5 MW/m^2$, for all three spray configurations are lower than the data obtained from TC 1. The CHFs are reached at lower temperatures: $\sim 150^\circ C$ for the top and side spray configurations; and $\sim 100^\circ C$ for bottom spray configuration.

\[\text{Figure 5.27: Boiling curves for top spray configuration with different moving speeds}\]
Figure 5.28: Boiling curves for side spray configuration with different moving speeds

Figure 5.29: Boiling curves for bottom spray configuration with different moving speeds
5.2.4 Effect of Spray Configuration

The effect of spray configuration on the boiling curves may be assessed by considering the results from the stationary sample tests. The IZ and FFZ results are compared for each spray configuration in Figures 5.30, 5.31 and 5.32 for flow rates of 10LPM, 40LPM and 50LPM, respectively. The red, green and blue lines in the figures represent the top, side and bottom spray configuration results, respectively.

When the cooling water flow rate is 40LPM (Figure 5.30), the effect of spray configuration is minimized as similar boiling curves are observed in both the IZ and FFZ. The CHF for side spray configuration is slightly higher than the others, and the CHF temperature is also the lowest.

When the flow rate increases to 50LPM (Figure 5.31), the CHF for the top spray configuration is the highest in the IZ, but is the lowest in the FFZ. The CHF for the bottom spray configuration is similar to the one for the side spray in the IZ, but is lower than the side.
spray result in the FFZ. For the top spray configuration in the FFZ region, a very low heat flux occurs at the beginning of the test before transition boiling occurs. This occurs because of the ejection of the water jets. The surface experiences the natural convection before getting wet. This pre-cooling in the top spray configuration results in the sample beginning to be cooled by transition boiling regime at a lower surface temperature. Thus CHF for top spray in FFZ is the lowest, observed in Figure 5.31.

![Boiling curves in different spray configurations with flow rate at 50LPM](image)

**Figure 5.31:** Boiling curves in different spray configurations with flow rate at 50LPM

When the flow rate is decreased to 10LPM (Figure 5.32), film boiling occurs for the top spray configuration only, and the resulting CHF for this test is 1.5 times higher than the others in the FFZ. In the IZ, the bottom spray configuration has the highest CHF (∼ 5MW/m²), which is reached at the lowest surface temperature (∼ 150°C) compared to the others (∼ 240°C).
5.3 The Idealization of Boiling Curves

One of the goals of this research is to develop a database for industry of practical heat flux boundary conditions for describing the boiling water heat transfer occurring in the secondary cooling of HDC casting. In order to easily apply the heat flux results, it would be helpful if the calculated boiling curves could be idealized to be a standardized function of surface temperature. To fulfill this and yield the ultimate results of this research, the boiling curve data were fit to a standard function using Gnuplot, which is an open-source package (current version: 4.4.0). In the following sections, the fitting method will be introduced and assessed. Then, the idealized boiling curves will be presented and summarized.

5.3.1 Fitting Method

The example boiling curves, shown in Figure 5.22, were used to develop the fitting methodology. The heat fluxes at high temperatures are the most important for characterizing...
the cooling process, thus special attention was paid to these portions of the curves. Similar features may be observed in the boiling curves: i) if film boiling occurs, the heat flux varies linearly until the temperature decreases to $\sim 480^\circ C$; ii) if no film boiling occurs, the heat flux rises almost linearly as the surface temperature decreases directly from $500^\circ C$; iii) a quadratic transition occurs through the CHF until the temperature drops below $\sim 100^\circ C$; iv) the heat flux drops rapidly in a nearly linear or logarithmic fashion in the convective zone.

The general equations used for fitting are summarized in Table 5.4.

<table>
<thead>
<tr>
<th>Fitting function</th>
<th>Temperature range</th>
<th>Fitting parameters</th>
</tr>
</thead>
<tbody>
<tr>
<td>$q_0(T_s) = a_0 \times T_s + b_0$</td>
<td>$T_s &gt; 480^\circ C$</td>
<td>$a_0, b_0$</td>
</tr>
<tr>
<td>$q_1(T_s) = a_1 \times T_s + b_1$</td>
<td>$T_s &gt; 400^\circ C$</td>
<td>$a_1, b_1$</td>
</tr>
<tr>
<td>$q_2(T_s) = a_2 \times T_s^2 + b_2 \times T_s + c_2$</td>
<td>$400^\circ C &gt; T_s &gt; 100^\circ C$</td>
<td>$a_2, b_2, c_2$</td>
</tr>
<tr>
<td>$q_3(T_s) = a_3 \times T_s + b_3$</td>
<td>$T_s &lt; 100^\circ C$</td>
<td>$a_3, b_3$</td>
</tr>
<tr>
<td>or $q_3(T_s) = a_3 \times \log_{10} T_s + b_3$</td>
<td>$T_s &lt; 100^\circ C$</td>
<td>$a_3, b_3$</td>
</tr>
</tbody>
</table>

Once the fitting parameters were calculated, the transition temperatures from $q_0(T_s)$ to $q_1(T_s)$, $q_1(T_s)$ to $q_2(T_s)$ or $q_2(T_s)$ to $q_3(T_s)$ were determined by the intersection of the functions. For the boiling curves in Figure 5.22, the fitting functions are summarized in Table 5.5. The boiling curves at TC 16 was chosen to represent the IZ, and TC 8 & 2 were chosen to represent the FFZ at the middle and end of the sample, respectively. The idealized and calculated boiling curves are compared in Figure 5.33 for the 3 TC locations.

The boiling curves in the IZ and FFZ near the edge of the sample have comparable heat fluxes (TC 1 & TC 2 and TC 15 & TC 16) in Figure 5.22, especially the high peak values. Based on this, the boiling curve at TC 16 will also be used for TC 15 and the same assumption is also made for TC 1 and TC 2. The boiling curves in rest of the FFZ show a variation with distance from the mold. Since the CHF plays such an important role in the cooling process, a method was sought to include this variation into the fit. The boiling
curves at the other TC locations in the FFZ were scaled by a factor $\beta_l$, based on the CHFs at each TC locations relative to TC 8:

$$\beta_l = \frac{CHF_{TCn}}{CHF_{TC8}}$$

(5.1)

where n is the TC number, from 3 to 14. The boiling curve at TC 8, which is located in the center of the sample, exhibits all the features of the boiling curves in FFZ. Thus it was chosen as the standard boiling curve for the fitting method. The results would be further improved if $\beta_l$ could be expressed as a standard function of distance from the mold. Figure 5.34 shows $\beta_l$ vs. distance from the mold for the top spray configuration with different flow rates. Unfortunately, the random distribution of $\beta_l$ along the surface makes the standardization impossible.

<table>
<thead>
<tr>
<th>TC number</th>
<th>Temperature range</th>
<th>Fitting functions</th>
</tr>
</thead>
<tbody>
<tr>
<td>2</td>
<td>$T_s \geq 355.8^\circ C$</td>
<td>$q = -9090.9T_s + 4417550$</td>
</tr>
<tr>
<td></td>
<td>$355.8^\circ C &gt; T_s &gt; 76.6^\circ C$</td>
<td>$q = -72.8T_s^2 + 29101.4T_s + 49763.2$</td>
</tr>
<tr>
<td></td>
<td>$T_s \leq 76.6^\circ C$</td>
<td>$q = 19102.3T_s + 388320$</td>
</tr>
<tr>
<td></td>
<td>$T_s \geq 484.9^\circ C$</td>
<td>$q = -432.8T_s + 288306$</td>
</tr>
<tr>
<td>8</td>
<td>$484.9^\circ C &gt; T_s \geq 404.1^\circ C$</td>
<td>$q = -18298.6T_s + 8951420$</td>
</tr>
<tr>
<td></td>
<td>$404.1^\circ C &gt; T_s \geq 94.5^\circ C$</td>
<td>$q = -46.9T_s^2 + 19634.9T_s + 1230110$</td>
</tr>
<tr>
<td></td>
<td>$T_s &lt; 94.5^\circ C$</td>
<td>$q = 28153.8T_s + 6179.8$</td>
</tr>
<tr>
<td></td>
<td>$T_s \geq 382.2^\circ C$</td>
<td>$q = -31707.8T_s + 15937300$</td>
</tr>
<tr>
<td>16</td>
<td>$812.2^\circ C &gt; T_s \geq 53.8^\circ C$</td>
<td>$q = -36.6T_s^2 + 16683.3T_s + 2793370$</td>
</tr>
<tr>
<td></td>
<td>$T_s &lt; 53.8^\circ C$</td>
<td>$q = 60127.4T_s + 350828$</td>
</tr>
</tbody>
</table>

The idealized boiling curves are shown in Figure 5.35. The accuracy of these idealized boiling curves was tested by applying these heat fluxes in the forward heat conduction model to predict the temperatures in the example test. The temperatures calculated by the application of the idealized heat fluxes are compared to the measured temperature in Figure 5.36.
Figure 5.33: Idealized and calculated boiling curves at TC 2, TC 8, and TC 16 with top spray configuration at 40LPM flow rate with stationary sample.

Figure 5.34: $\beta_l$ distribution along the distance from the mold for top spray configuration with different flow rates.

The comparison shows a good agreement between measured and predicted temperatures. Compared with previous results from VDC casting, the idealized heat flux is of the same
order of magnitude, and its distribution shows similar behavior. Additionally, the measured temperatures are compared to the temperatures calculated by scaling the idealized boiling curves by ±10% and ±20%. The comparison of these temperatures at TC 8 is shown in Figure 5.37. Similar temperature responses are observed for each condition. The original idealized boiling curves yield the best result and decreasing or increasing the idealized boiling curves causes a time shift (delay or advance) in the temperature decrease. The time decreases by 1.0s and 3.0s when the idealized boiling curves are increased by 10% and 20%, and increases by 0.6s and 2.4s when the idealized boiling curves are decreased by 10% and 20%. Thus, despite the combined effect of the compromise introduced by considering thermocouples in groups of 4 and the loss of resolution caused by idealizing the boiling curves, the calculated boiling curves are accurate to within 10%. This suggests that the idealized boiling curves can be used to reproduce the temperature variation during the cooling process and thus can be used as the boundary conditions to characterize the heat transfer occurring during secondary cooling of HDC casting.

![Idealized boiling curves](image)

**Figure 5.35:** The idealized boiling curves for top spray configuration with stationary test and $Q = 40LPM$
Figure 5.36: Test of the idealized boiling curves for top spray configuration with stationary test and $Q = 40LPM$

Figure 5.37: Comparison of measured and calculated temperature using different scaled idealized boiling curves with stationary test and $Q = 40LPM$
5.3.2 Summary

The functions of the idealized boiling curves for stationary sample tests with flow rate at 40LPM and moving sample tests with speed at 122mm/min are summarized in Table 5.6 and 5.7. These parameters were chosen because they are close to the operational parameters used in the industry. With the methodology developed by this work, the functions for the other conditions could be obtained using the corresponding measured data.
Table 5.6: Summary of the fitting functions of idealized boiling curves

<table>
<thead>
<tr>
<th>Operational parameters</th>
<th>Spray configuration</th>
<th>TC number</th>
<th>Temperature range</th>
<th>Fitting functions</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Stationary sample</strong></td>
<td><strong>Side spray</strong></td>
<td>TC 15</td>
<td>$T_s \geq 382.6^\circ$C</td>
<td>$q = -31714.7T_s + 16015400$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$382.6^\circ$C $\geq T_s \geq 45.7^\circ$C</td>
<td>$q = -36.9T_s^2 + 15966.1T_s + 3166940$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$T_s &lt; 45.7^\circ$C</td>
<td>$q = 104971T_s - 977711$</td>
</tr>
<tr>
<td><strong>Stationary sample</strong></td>
<td><strong>Side spray</strong></td>
<td>TC 8</td>
<td>$T_s \geq 486.1^\circ$C</td>
<td>$q = -1805T_s + 818763$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$486.1^\circ$C $&gt; T_s \geq 396.0^\circ$C</td>
<td>$q = -17939T_s + 8805750$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$396.0^\circ$C $&gt; T_s \geq 86.3^\circ$C</td>
<td>$q = -45.6T_s^2 + 18140.3T_s + 1624570$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$T_s &lt; 86.3^\circ$C</td>
<td>$q = 32846.5T_s + 17155.8$</td>
</tr>
<tr>
<td><strong>Stationary sample</strong></td>
<td><strong>Bottom spray</strong></td>
<td>TC 2</td>
<td>$T_s \geq 284.4^\circ$C</td>
<td>$q = -13654.5T_s + 6493300$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$284.4^\circ$C $&gt; T_s \geq 99.7^\circ$C</td>
<td>$q = -63.8T_s^2 + 26548.1T_s + 216150$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$T_s &lt; 99.7^\circ$C</td>
<td>$q = 22136.6T_s + 22275.5$</td>
</tr>
<tr>
<td><strong>Stationary sample</strong></td>
<td><strong>Bottom spray</strong></td>
<td>TC 15</td>
<td>$T_s \geq 381.6^\circ$C</td>
<td>$q = -30578T_s + 15131000$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$381.6^\circ$C $&gt; T_s \geq 49.9^\circ$C</td>
<td>$q = -46.5T_s^2 + 19675.2T_s + 2729210$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$T_s &lt; 49.9^\circ$C</td>
<td>$q = 138053T_s - 3290400$</td>
</tr>
<tr>
<td><strong>Stationary sample</strong></td>
<td><strong>Bottom spray</strong></td>
<td>TC 8</td>
<td>$T_s \geq 483.1^\circ$C</td>
<td>$q = -1506.7T_s + 818763$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$483.1^\circ$C $&gt; T_s \geq 407.1^\circ$C</td>
<td>$q = -17729.7T_s + 8656130$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$407.1^\circ$C $&gt; T_s \geq 84.4^\circ$C</td>
<td>$q = -43.5T_s^2 + 17663.8T_s + 1395710$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$T_s &lt; 84.4^\circ$C</td>
<td>$q = 39088.1T_s - 720834$</td>
</tr>
<tr>
<td><strong>Stationary sample</strong></td>
<td><strong>Bottom spray</strong></td>
<td>TC 2</td>
<td>$T_s \geq 467.9^\circ$C</td>
<td>$q = 64.7T_s + 43.0$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$467.9^\circ$C $&gt; T_s \geq 315.1^\circ$C</td>
<td>$q = -10703T_s + 5038110$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$315.1^\circ$C $&gt; T_s \geq 75.0^\circ$C</td>
<td>$q = -95.7T_s^2 + 38814.6T_s - 1060750$</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>$T_s &lt; 75.0^\circ$C</td>
<td>$q = 28918T_s - 856811$</td>
</tr>
<tr>
<td>Operational parameters</td>
<td>Spray configuration</td>
<td>TC number</td>
<td>Temperature range</td>
<td>Fitting functions</td>
</tr>
<tr>
<td>------------------------</td>
<td>--------------------</td>
<td>-----------</td>
<td>-----------------------------------</td>
<td>------------------------------------------------</td>
</tr>
<tr>
<td></td>
<td>Moving sample</td>
<td>v = 122mm/min</td>
<td>Top spray</td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td>40LPM</td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td></td>
<td></td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>TC 1 420.8°C ≥ T_s ≥ 420.8°C</td>
<td>( q = -24359.4T_s + 12447900 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>TC 8 420.8°C &gt; T_s ≥ 39.1°C</td>
<td>( q = -24.0T_s^2 + 9908.1T_s + 2268770 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>T_s &lt; 39.1°C</td>
<td>( q = 48784.7T_s + 711018 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>TC 16 362.6°C ≥ T_s ≥ 160.9°C</td>
<td>( q = -300.8T_s + 146155 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>160.9°C ≥ T_s ≥ 58.0°C</td>
<td>( q = -98.5T_s + 1843570 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>T_s &lt; 58.0°C</td>
<td>( q = -379.1T_s + 2672550 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>TC 1 419.4°C ≥ T_s ≥ 115.1°C</td>
<td>( q = -13.8T_s^2 - 379.1T_s + 2672550 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>T_s &lt; 115.1°C</td>
<td>( q = 2053160\log_{10}T_s - 1786690 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>TC 8 419.4°C &gt; T_s ≥ 115.1°C</td>
<td>( q = -13.8T_s^2 - 379.1T_s + 2672550 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>T_s &lt; 115.1°C</td>
<td>( q = 2053160\log_{10}T_s - 1786690 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>TC 1 422.3°C ≥ T_s ≥ 35.5°C</td>
<td>( q = -26.9T_s^2 + 109901T_s + 2012630 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>T_s &lt; 35.5°C</td>
<td>( q = -78135.3T_s + 5145500 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>TC 8 422.3°C &gt; T_s ≥ 35.5°C</td>
<td>( q = -26.9T_s^2 + 109901T_s + 2012630 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>T_s &lt; 35.5°C</td>
<td>( q = -78135.3T_s + 5145500 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>TC 1 423.8°C ≥ T_s ≥ 128.2°C</td>
<td>( q = 10.9T_s - 385.3 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>T_s &lt; 128.2°C</td>
<td>( q = 4521.3T_s + 1920200 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>TC 8 423.8°C &gt; T_s ≥ 128.2°C</td>
<td>( q = 10.9T_s - 385.3 )</td>
</tr>
<tr>
<td></td>
<td></td>
<td></td>
<td>T_s &lt; 128.2°C</td>
<td>( q = 4521.3T_s + 1920200 )</td>
</tr>
</tbody>
</table>
Chapter 6

Conclusions and Future Work

6.1 Summary and Conclusions

An experimental apparatus was developed to simulate the secondary cooling process that occurs during HDC casting. Compared to performing these experiments on the operating process, this approach is cost effective, safe, and more flexible.

An IHC technique was used to calculate the heat fluxes that occurred during the simulated secondary cooling process. A 2 stage validation procedure was employed to verify that this technique can accurately predict the surface heat flux in this problem.

Overall, film boiling does not occur for most spray configurations because it is hard to form stable vapor blanket with the turbulent water jets. The calculated heat fluxes in the IZ are higher than in the FFZ. The CHF in the IZ can be as high as $5.5\text{MW/m}^2$ (stationary test, top spray and 50LPM), compared to $4\text{MW/m}^2$ in the FFZ (stationary test, top spray and 50LPM).

The cooling water flow rate plays an important role in many aspects of the cooling process. For the top spray configuration, high flow rates (40LPM and 50LPM) result in transition boiling occurring immediately without film boiling. For the side spray configuration, when the flow rate decreases to 10LPM, boiling heat transfer did not occur at locations near
the top of the sample and far away from the mold because the water failed to reach these locations. In the industrial process, these areas would be cooled by water flowing over the side from the top surface.

Higher moving speed leads to a higher CHF because the motion of the sample relative to the mold enhance bubble growth and detachment, accelerating bubble motion in the nucleate boiling regime.

An technique to idealize the boiling curves for a given set of operational parameters was developed. The resulting heat fluxes were used to adequately reproduce the temperature history data from the experimental apparatus.

### 6.2 Future Work

Based on the results of this research work, the following future work is recommended:

(I) Resize the sample dimensions to match the ingot produced in industry and develop an easy–to–operate and adjustable system for handling different sizes of the ingot. In this way, the effect of sample size on the heat transfer in the secondary cooling region can be investigated.

(II) Since the initial temperature plays an important role in boiling water heat transfer, a temperature monitor and control system is necessary to ensure the boiling water heat transfer starts accurately at the desired temperature and also the effect of the initial temperature on the heat transfer needs to be studied.

(III) Different alloys can be used in the tests to study the effect of the thermophysical properties on the boiling water heat transfer.
Bibliography


Appendix A

FEM Algorithm for 2–D heat conduction problem

The governing equation for 2–D transient heat conduction problem is:

\[
\frac{\partial}{\partial x} \left( k_x \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left( k_y \frac{\partial T}{\partial y} \right) = \rho C_p \frac{dT}{dt}
\]  \hspace{1cm} (A.1)

in which thermal conductivity \( k \) is independent of \( x \)- and \( y \)-direction if the material is isotropic.

In this research, the PDE is solved subject to the following boundary conditions and initial condition:

\[
k \left( \frac{\partial T}{\partial x} n_x + \frac{\partial T}{\partial y} n_y \right) + q(x,y,t) = 0 \hspace{1cm} (A.2)
\]

\[
k \left( \frac{\partial T}{\partial x} n_x + \frac{\partial T}{\partial y} n_y \right) + h(x,y,t) T = 0 \hspace{1cm} (A.3)
\]

\[
T = T_0(x,y) \big|_{t=0} \hspace{1cm} (A.4)
\]

Equation \( A.2 \) and \( A.3 \) are called Neumann boundary condition and Cauchy boundary condition respectively.

The finite element solution is based on one of weighted residual method called Galerkin’s
method, in which weight functions are set equal to shape functions $N_i$. So a weight function or so-called trial function for this PDE is assumed:

$$\tilde{T} = \sum N_i T_i$$  \hspace{1cm} (A.5)

in which $T_i$ is the nodal temperature. For 2-D 4-node linear element, the four shape functions are:

$$N_1(u,v) = \frac{(1-u)(1-v)}{4}$$  \hspace{1cm} (A.6)

$$N_2(u,v) = \frac{(1-u)(1+v)}{4}$$  \hspace{1cm} (A.7)

$$N_3(u,v) = \frac{(1+u)(1-v)}{4}$$  \hspace{1cm} (A.8)

$$N_4(u,v) = \frac{(1+u)(1+v)}{4}$$  \hspace{1cm} (A.9)

By substituting Equation A.5 into the PDE and requiring the residual to vanish, the following form of transient heat conduction equation can be achieved after a series of algebraic manipulations[50]:

$$[C] \frac{dT}{dt} + [K] \{T\} = \{f\}$$  \hspace{1cm} (A.10)

The conductivity term (first term of Equation A.10) is expressed by:

$$-[K] \{T\} = -\sum_{i=1}^{2} \sum_{j=1}^{2} [B]^T [k] [B] |J| \{T^e\}$$  \hspace{1cm} (A.11)

where

$$[B]_{2 \times 1} = \begin{bmatrix} \frac{\partial N_i}{\partial x} \\ \frac{\partial N_i}{\partial y} \end{bmatrix}$$  \hspace{1cm} (A.12)
and Jacobian matrix $[J]$ is defined by:

$$
[J] = \left[ \begin{array}{c}
\sum \frac{\partial N_i}{\partial u} x_i \\
\sum \frac{\partial N_i}{\partial v} x_i \\
\sum \frac{\partial N_i}{\partial u} y_i \\
\sum \frac{\partial N_i}{\partial v} y_i 
\end{array} \right]
$$

(A.13)

so the determinant of Jacobian matrix $|J|$ is calculated by:

$$
|J| = \left( \sum \frac{\partial N_i}{\partial u} x_i \right) \left( \sum \frac{\partial N_i}{\partial v} y_i \right) - \left( \sum \frac{\partial N_i}{\partial u} y_i \right) \left( \sum \frac{\partial N_i}{\partial v} x_i \right)
$$

(A.14)

Correspondingly, the heat capacity term in Equation (A.10) is defined by:

$$
-C \frac{dT}{dt} = - \left[ \sum_{i=1}^{2} \sum_{j=1}^{2} \{N\} \rho C_p \{N\}^T |J| \right] \frac{dT}{dt}
$$

(A.15)

and load vector at RHS of the Equation is given by:

for top and bottom faces

$$
 f_\phi = \sum_{i=1}^{2} \phi \{N\} \sqrt{J_{11}^2 + J_{12}^2}
$$

(A.16)

for sides faces

$$
 f_\phi = \sum_{i=1}^{2} \phi \{N\} \sqrt{J_{21}^2 + J_{22}^2}
$$

(A.17)

If any of the boundary conditions or material thermophysical properties vary as a function of temperature or time, the governing equation becomes non-linear. Newton–Raphson (NR) method is chosen in order to find the approximate roots, namely, nodal temperature, of this non-linear equation.

A suitable convergence criteria is required to estimate the accuracy of the iterative solution within each time increment. Because of unknow actual solution, the following tolerance criteria is chosen:

$$
Error = \sum_{i=1}^{n} \Delta T_i^2 < 1 \times 10^{-6}
$$

(A.18)

where $n$ is the number of nodes. Fig. (A.1) shows the flow chat of procedure of FEM model.
Figure A.1: Flow chart of calculation procedure for FEM model