STUDY OF BOILING HEAT TRANSFER ON A STATIONARY DOWNWARD FACING HOT STEEL PLATE COOLED BY A CIRCULAR WATER JET

By

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Abstract

The controlled cooling of steel on the runout table is crucial to the desired microstructural evolution in the manufacturing process and desired mechanical properties of the final product.

The purpose of this study is to improve our fundamental knowledge of heat transfer on the runout table by investigating transient heat transfer on a downward facing hot surface cooled by a highly subcooled single circular free surface jet of water. Both surface and interior temperatures of the plate were collected at conditions as close as possible to industrial conditions. A two-dimensional inverse heat conduction model including the effect of progression of the rewetting front on heat transfer was developed, validated and employed to estimate heat fluxes on the bottom of plate. The influence of jet subcooling and flow rate on heat transfer was discussed. Boiling heat transfer correlations were tested with data from the current study and the cooling efficiency of bottom jets was compared with corresponding top jet cooling data from the literature.
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Nomenclature

\( c_p \)  \quad \text{Specific heat (J/kg°C)}

\( d_n \)  \quad \text{Circular jet diameter at exit of the nozzle diameter (nozzle diameter) (mm)}

\( d_j \)  \quad \text{Jet diameter at impingement (mm)}

\( D \)  \quad \text{Diameter of test cell (mm)}

\( F \)  \quad \text{Liquid solid contact area fraction (%)}

\( g \)  \quad \text{Gravitational acceleration (m/s}^2\text{)}

\( h \)  \quad \text{Heat transfer coefficient (W/m}^2\text{°C)}

\( H \)  \quad \text{Distance between the nozzle and the heated plate (mm)}

\( h_{lg} \)  \quad \text{Latent heat of evaporation from liquid to vapour (J/kg)}

\( J_a. \)  \quad \text{Jakob number (} \left( \frac{\rho_l}{\rho_g} \right) \left( c_p \cdot \Delta T_{sub} / h_{lg} \right) \)

\( k \)  \quad \text{Thermal conductivity (W/m°C)}

\( L \)  \quad \text{Characteristic length (mm)}

\( p \)  \quad \text{Pressure (Pa)}

\( Pr \)  \quad \text{Prandtl number (} \mu c_p/k \text{)}

\( q'' \)  \quad \text{Heat flux (W/m}^2\text{)}

\( r \)  \quad \text{Radial coordinate (mm)}

\( Re \)  \quad \text{Reynolds number (vL/v)}

\( t \)  \quad \text{Time (second)}

\( th \)  \quad \text{Thickness (mm)}

\( T \)  \quad \text{Temperature (°C)}

\( T_{sat} \)  \quad \text{Saturation temperature of fluid (°C)}

\( v \)  \quad \text{Velocity (m/s)}

\( v_n \)  \quad \text{Jet velocity at exit of the nozzle (m/s)}

\( v_j \)  \quad \text{Jet velocity at the impingement point (m/s)}

\( W \)  \quad \text{Width (mm)}
Greek symbols

\( \alpha \)  
Regularization parameter

\( \Delta T_{\text{sat}} \)  
Wall superheat, the difference between surface temperature and fluid saturation temperature (°C)

\( \Delta T_{\text{sub}} \)  
Subcooling, the difference between fluid saturation temperature and fluid temperature (°C)

\( \mu \)  
Dynamic or absolute viscosity (Pa s)

\( \nu \)  
Kinematic viscosity (\( \mu/\rho \)) (m²/s)

\( \rho \)  
Density (kg/m³)

\( \sigma \)  
Surface tension (N/m)

\( \sigma_{\text{SB}} \)  
Stefan-Boltzmann radiation constant (5.67 \times 10^{-8} \text{ W/(m}^2\text{°C}^4))

Subscripts

\( B \)  
Black region

\( \text{CHF} \)  
Critical heat flux

\( \text{FB} \)  
Film boiling

\( \text{FNB} \)  
Fully developed nucleate boiling

\( g \)  
Vapour

\( i \)  
Impingement

\( \text{ini} \)  
Initial

\( j \)  
Jet

\( l \)  
Liquid

\( \text{MHF} \)  
Minimum heat flux

\( n \)  
Nozzle

\( \text{NB} \)  
Nucleate boiling

\( \text{rew} \)  
Rewetting

\( \text{sat} \)  
Saturation

\( \text{sub} \)  
Subcooling
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<td>Transition boiling</td>
</tr>
<tr>
<td>w</td>
<td>Wall</td>
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1. Introduction

Following the finishing rolling mill, runout table cooling is employed to control the temperature profiles in the strip. Detailed knowledge of this process is required to control the quality of the final product, such as the microstructure and mechanical properties as well as flatness of the strip. Studying heat transfer on a fully scaled runout table is difficult, expensive and generally unrealistic. As an alternative method, researchers often perform small laboratory experiments, which include either one jet or array of jets with round or rectangular nozzles, impinging on stationary or slowly moving test specimens.

In order to obtain cooling data under the real runout table cooling conditions, an industrial pilot runout table facility had been previously built at the Centre for Metallurgical Process Engineering (CMPE) of the University of British Columbia allowing both stationary and moving plate tests on various types of steel samples. Temperature measurements were made in the plate at different distances from the jet. The effects of various parameters such as jet flow rate, subcooling, and initial temperature of plate on the cooling have been investigated.

So far these investigations had been restricted to top jet impingement heat transfer. However, the investigations on bottom jet impingement, whether on hydrodynamic or thermodynamic aspects, are rare. Thus, the general purpose of the present study is to obtain fundamental knowledge on heat transfer during bottom jet impingement cooling by experimentally studying the transient heat transfer between a stationary steel plate in the “as rolled” condition and an upward impinging water jet.
The tests for this study were performed on UBC's pilot-scale runout table which has been upgraded for bottom jet experiments. Data were collected at conditions as close as possible to industrial conditions. The heat flux on the bottom surface of the test plate was estimated using an inverse heat conduction model and measured internal temperatures. The influence of the water temperature and jet flow rate on heat transfer was investigated. Various boiling heat transfer models were tested.
2. Literature review

Controlled runout table cooling plays an important role in obtaining optimum mechanical properties of final products by providing desired microstructures. Thus, it has attracted significant research interest over the past several decades. In this chapter, following a brief introduction to the runout table layout, the heat transfer characteristics during runout table cooling are reviewed. The emphasis is put on both theoretical and experimental work on boiling heat transfer.

2.1 Introduction to runout table cooling

As shown in Figure 2-1, the runout table typically consists of a set of cooling banks mounted at the top and the bottom of the strip, motorized rolls and a down coiler. The strip exits the finishing mill and is subjected to cooling when moving on rolls through the jet lines to the down coiler where it is coiled for storage or further processing. Each bank has several headers and each header consists of one or two rows of jet lines. One jet line is an array of nozzles in the transverse direction of the strip. Three types of nozzles, i.e. spray nozzles, round tube nozzles, and slot-type nozzles are employed for spray cooling, laminar flow cooling and water curtain systems, respectively, as illustrated by Figure 2-2.

The water flow is generated by gravity when the storage tanks are located above the runout table level or by pump when storage tanks are located below the runout table level. The level of water is maintained constant in the storage tanks to obtain stable water flow. The water in storage tanks is usually supplied by water towers.

As shown in Figure 2-3, Tacke et al. (1985) compared the cooling capacity of three cooling systems, and found that water curtain and laminar flow cooling had higher cooling
capacity than spray cooling because they could penetrate the vapour film on the cooled wall. Besides, the water curtain could provide more uniform cooling rate across the width of strip since it spanned the entire width of the strip (Kohring, 1985).

Figure 2-1 Schematics of a runout table

Figure 2-2 Three cooling systems (Chen et al., 1992)
To determine the strip temperature along the runout table, pyrometers are placed at the exit of the finishing mill and before the down coiler (sometimes also between the finishing mill and the down coiler).

### 2.2 Heat transfer mechanisms on the runout table

It is very important to give a realistic description of the different heat transfer mechanisms on the runout table, especially for computer modelling purposes. However, the results from different research groups show that there are still great differences in their opinion on the major heat transfer mechanisms during runout table cooling.

Figure 2-3 Cooling rates as a function of strip gauge under different cooling systems

(Tacke et al., 1985)
According to Colás (1987,1994), Evans et al. (1993), and Prieto et al. (2001), the heat transfer on the top wall of the hot strip impinged by a water curtain or a circular water jet, as demonstrated by Figure 2-4, can be classified into three cooling zones:

1) The impingement zone, which is just below the active nozzles and the immediately neighbouring regions, and the major cooling mechanisms are forced convection and radiation
2) The parallel flow zone which is beside the impingement zone and the heat transfer is dominated by stable film boiling

3) The radiation zone, where the strip is assumed dry and the predominant cooling mechanism is radiation

For bottom jet cooling there is no parallel flow zone since gravity will pull the water out of contact from the strip.

Viskanta et al. (1992) assumed five heat transfer regimes on a stationary plate impinged by a single jet. Besides forced convection, film boiling and radiation suggested by Prieto et al. (2001), the transition/nucleate boiling regime took place in a comparatively narrow region separating the force convection region from the film boiling region. Another regime was agglomerated pool cooling, which was due to wall tension effects. This regime was located between the film boiling regime and the air-cooling regime.

Based on experimental results of Ishigai et al. (1978), and Takeda et al. (1986), Hernandez (1995,1999) proposed that transition boiling rather than nucleate boiling was the predominant cooling mechanism in both impingement zone when the surface temperature was over 300 °C, and parallel flow zone when the temperature was over 250 °C.

The heat transfer mechanism may be further complicated by its dependence on various operating parameters such as the local surface temperature, strip velocity, water temperature and flow rate, wall conditions such as roughness and oxidation. Other important parameters may include thermal properties of the material and the coolant and the geometry of the cooling system such as the nozzle shape, dimension, height and angle, and the jet arrangement along the table.
However, a direct study of heat transfer on a fully scaled runout table is difficult and unrealistic. As an alternative method, researchers often perform small laboratory experiments, which include one jet or an array of jets of round or rectangular nozzles, impinging on stationary or slowly moving test specimens.

2.3 **Experimental investigation**

In an effort to improve the knowledge of heat transfer on the runout table, experiments were performed under both steady-state and transient cooling conditions, as summarized in Table 2-1. In this section, these two experimental methods will be introduced and compared.

Employing steady-state methods a specimen is heated by means of temperature controlled or heat flux controlled systems and cooled at the same time until steady-state is achieved (Robidou et al. 2002, 2003; Fry et al., 1997). The heat flux removed by the cooling system can be estimated directly from the heat input from the heat source and the heat transfer coefficient can be calculated by solving Fourier’s law of heat conduction equation and Newton’s law of cooling. In contrast, transient tests are usually performed under cooling conditions (Hauksson, 2001; Meng, 2002). With this approach, the test plate is heated up prior to the test temperature and then cooled under jet impingement. During this process, temperature-time history of the plate is recorded. Mathematical methods such as the inverse heat conduction model are employed to determine the heat flux and/or heat transfer coefficient.

2.3.1 **Steady-state cooling test**

Miyasaka et al. (1980) investigated the high heat flux mechanism at stagnation with a two-dimensional water jet impinging upward on a small heat transfer wall, which
<table>
<thead>
<tr>
<th>Author</th>
<th>Initial temperature (°C), Subcooling (°C), Superheat (°C), Flow rate (l/min) or velocity (m/s)</th>
<th>Experimental parameters</th>
<th>Data acquisition</th>
<th>Heat flux or Heat transfer coefficient (maximum)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Ishigai et al. 1978</td>
<td>Transient, $T_{ini}=1000$ °C, top jet $\Delta T_{sub}=5-55$ °C, $\Delta T_{sat}=900$ °C $v_n=1.1-3.17$ m/s</td>
<td>Rectangular $W=6.2$, $L=56.2$, $H=15$</td>
<td>Stainless steel, 50×12×2mm</td>
<td>Up to 12 MW/m²</td>
</tr>
<tr>
<td>Miyasaka et al. 1980</td>
<td>Steady, $T_{ini}=900$ °C, bottom jet $\Delta T_{sub}=80$ °C, $\Delta T_{sat}=10-800$ °C $v_n=1.5-15.3$ m/s</td>
<td>Rectangular $W=10$, $H=15$</td>
<td>Copper with platinum foil surface, D=1.5mm</td>
<td>Up to 46.7 MW/m²</td>
</tr>
<tr>
<td>Hatta et al. 1983</td>
<td>Transient, $T_{ini}=900$ °C, top jet $\Delta T_{sub}=30-85$ °C, $\Delta T_{sat}=800$ °C $v_n=0.1-7$ l/min</td>
<td>Circular $d_n=10$, $H=50-600$</td>
<td>18-8 stainless steel, 200×200×10mm</td>
<td>Type-K TC 0.65mm</td>
</tr>
<tr>
<td>Ochi et al. 1984</td>
<td>Transient, $T_{ini}=1100$ °C, top jet $\Delta T_{sub}=5-80$ °C, $\Delta T_{sat}=1000$ °C $v_n=2-7$ m/s</td>
<td>Circular $d_n=5$, $10$, $20$, $H=25$</td>
<td>Stainless steel, 210×50×2mm</td>
<td>Up to 10 MW/m²</td>
</tr>
<tr>
<td>Kumagai et al. 1995a</td>
<td>Transient, $T_{ini}=390$ °C, top jet $\Delta T_{sub}=0-50$ °C, $\Delta T_{sat}=0-290$ °C $v_n=1.5$, 2.5, and 3.5 m/s</td>
<td>Rectangular $W=1$, $L=28$, $H=N/A$</td>
<td>Copper 150×120×20mm</td>
<td>Up to 17 MW/m²</td>
</tr>
<tr>
<td>Kumagai et al. 1995b</td>
<td>Transient, $T_{ini}=390$ °C, top jet $\Delta T_{sub}=14-50$ °C, $\Delta T_{sat}=0-300$ °C, $v_n=3.5$ m/s</td>
<td>Rectangular $W=1$, $L=28$, $H=N/A$</td>
<td>Copper 150×120×20mm</td>
<td>Up to 27 MW/m²</td>
</tr>
<tr>
<td>Mitsutake et al. 2001</td>
<td>Transient, $T_{ini}=250$ °C, bottom jet $\Delta T_{sub}=20-80$ °C, $v_n=5-15$ m/s</td>
<td>Circular $d_n=2$, $H=N/A$</td>
<td>Copper, brass and carbon steel, D=94mm, $th=60$ mm</td>
<td>Copper: 14 MW/m²; Steel: 2 MW/m² at $\Delta T_{sub}=50$ °C, $v_n=5$ m/s</td>
</tr>
</tbody>
</table>

Note: $d_n$ = diameter of nozzle, $D$ = diameter of test cell, $H$ = distance between nozzle and cooled surface, $L$ and $W$ = length and width of slot nozzle, $th$ = thickness, $TC$ = thermocouple, $N/A$ = not available
### Table 2-1 Jet impingement test conditions - continued

<table>
<thead>
<tr>
<th>Author</th>
<th>Initial temperature (°C), Subcooling (°C), Superheat (°C), Flow rate (l/min) or velocity (m/s)</th>
<th>Nozzle dimension, Nozzle-to-surface distance (mm)</th>
<th>Test cell material, and size (mm)</th>
<th>Data acquisition</th>
<th>Heat flux or Heat transfer coefficient</th>
</tr>
</thead>
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<tr>
<td>Chen et al. 1991</td>
<td>Transient, $T_{in}$ = 245 °C, top jet $\Delta T_{sub}$ = 75 °C $v_n$ = 2.66 m/s</td>
<td>Circular $d_n$ = 4.76 H=90</td>
<td>Stainless steel, 355x254x6.35mm</td>
<td>Type-K TC, 12.7μm in diameter, Sampling interval 1 ms</td>
<td>Up to 22 MW/m²</td>
</tr>
<tr>
<td>Hall et al. 2001a</td>
<td>Transient, $T_{in}$ = 500, 650, 800 °C, top jet $\Delta T_{sub}$ = 75 °C, Deionized water, $v_n$ = 2-4 m/s</td>
<td>Circular $d_n$ = 5.1 H=100</td>
<td>Copper, D=112, t=25.4</td>
<td>Type-k TC, Sampling interval 0.25 s</td>
<td>Up to 45 MW/m²</td>
</tr>
<tr>
<td>Filipovic et al. 1995a,b,c</td>
<td>Transient, $T_{in}$ = over 700 °C, top jet $\Delta T_{sub}$ = 45-75 °C, $v_n$ = 2-4 m/s</td>
<td>N/A</td>
<td>508 x 38.1 x 25.2 mm, Copper</td>
<td>Type-k TC, Sampling interval 0.5 s</td>
<td>Up to 6.0 MW/m²</td>
</tr>
<tr>
<td>Robidou et al. 2002,2003</td>
<td>Steady-state, $T_{in}$ = 500 °C, top jet $\Delta T_{sub}$ = 5-15 °C, $\Delta T_{sat}$ = 0-400 °C $v_n$ = 0.5-1 m/s</td>
<td>Rectangular W=1, L=9 H=3, 6,10</td>
<td>8 Copper heaters, each 10x10x5mm</td>
<td>TC, 0.25mm in diameter, Sampling interval 5 ms</td>
<td>Up to 4.5 MW/m²</td>
</tr>
<tr>
<td>Hauksson 2001, Meng 2002</td>
<td>Transient, $T_{in}$ = 850 °C, top jet $\Delta T_{sub}$ = 30-95 °C, Flow rate = 15-45 l/min</td>
<td>Circular $d_n$ = 19 H=1500</td>
<td>Stainless steel, DQSK 280x280x7.62mm</td>
<td>Type-K TC, Sampling interval 0.01 s</td>
<td>SS: 20 MW/m² DQSK: 25 MW/m²</td>
</tr>
<tr>
<td>Lee et al. 2004</td>
<td>Transient, $T_{in}$ = 750 °C, top jet $\Delta T_{sub}$ = 70°C, Flow rate = 3 l/min</td>
<td>Circular $d_n$ = 6 H=0.04-0.45</td>
<td>SUS304 200x200x10mm</td>
<td>Type-k TC, 1.5 mm in diameter (sheath)</td>
<td>Up to 2.75 MW/m²</td>
</tr>
</tbody>
</table>

Note: $d_n$ = diameter of nozzle, D=diameter of test cell, H=distance between nozzle and cooled surface, L and W=length and width of slot nozzle, th=thickness, TC=thermocouple, N/A=not available
was made of platinum foil and attached to the bottom of a 1.5-millimetre diameter copper cylinder. The heat was generated by an electrical current controlled molybdenum heater and transferred to the impinged wall through a copper block. The test was performed under steady-state over the range of superheat 10-800 °C. The heat flux was obtained by solving Fourier’s equation for steady-state heat conduction using measured temperatures at location of 1.0 mm and 2.0mm above the wall. The effect of subcooling, jet velocity and stagnation pressure on nucleate boiling was discussed. They found that under this condition, the maximum heat flux could be as high as 46.7 MW/m².

Fry et al. (1997) performed steady-state heat transfer tests using a plate cooled by a single spray. The plate was heated by a propane gas burner, which was placed under the test plate. The tests were run from the low to the high temperature. Assuming a one dimensional heat conduction problem, the heat transfer coefficients were estimated from thermal gradients through the test plate using Newton’s law of cooling and Fourier’s law of heat conduction. The heat fluxes obtained were in ranges of 200 to 400 kW/m² in both nucleate and film boiling regions. The heat transfer coefficients fell between 250 and 500 kW/m²°C in the film boiling region and 2250 to 4250 kW/m²°C in the nucleate boiling region.

2.3.2 Transient cooling test

Hatta et al. (1983,1989) designed and performed the experiments of laminar jet impingement cooling on a 18-8 stainless steel plate heated up to an initial temperature of 900 °C in a gas furnace. The plate had a regular square shape with a dimension of 200 mm in side length and 10mm in thickness. One 10 mm U-shape hairpin nozzle was used in the experiments and the distance between the nozzle exit and test plate was in the range of 50
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to 600 mm. Temperatures were measured at five locations, which were at the bottom of a 2 mm deep hole made at the bottom of the plate. The heat transfer coefficients in the nucleate boiling region were estimated by comparing the experimental cooling curves with the calculated using a transient axisymmetric finite difference model.

Ochi et al. (1984) designed an apparatus to investigate boiling heat transfer in a hot stainless steel plate impinged by a circular water jet. The water pressure was provided by the head tank and its flow rate was measured by an orifice and a manometer. The plate with a size of 210×50×2 mm was heated by A.C. power up to 1100 °C. Temperature measurements were made at five places and thermocouples were spot-welded on the back-face of the plate. Three nozzle diameters of 5, 10, and 20 mm were used during the test. Subcooling and impinging jet velocities were varied from 5 to 80 °C and 2 to 7 m/s respectively.

Mitsutake et al. (2001) conducted a transient boiling heat transfer study with a single bottom water jet impinging upward on a hot cylindrical copper block. The block was heated by slot heaters to an initial temperature of 250 °C. The water with three different subcooling was fed to the nozzle by a pump. Sixteen thermocouples of 0.1 mm in diameter were used to measure the temperatures at eight radial locations. Each location included a pair of thermocouples, which were located at 1 mm and 5 mm below the wall respectively. The wall heat flux was evaluated with the measured temperatures at two different depths using a numerical model of two-dimensional heat conduction.

Using the top water jet impingement, Hall et al. (2001a, b) investigated the boiling heat transfer in the stagnation and radial flow region of the test cell (a copper disk). The test cell was heated in an electric furnace to approximately 10 °C above the desired initial
temperatures, which were at: 500 °C, 650 °C, and 800 °C. A thermocouple was placed at the bottom of the hole, which was drilled at 45 degree angle from the bottom wall of the test cell. The distance from the bottom of the hole to the top wall is within 1.3 mm. Temperature measurement was made at twelve locations at a frequency of 4 Hz, with nine thermocouples placed at nine discrete radial locations with an increment of 6.4 mm from the centreline and three other thermocouples placed at the 12.7 mm radial location with a space of 90 degree apart. Heat fluxes at thermocouple depth were estimated by solving the two-dimensional transient heat conduction equation with the control volume method. The wall heat fluxes and temperatures were extrapolated from those at thermocouple depth using an inverse heat conduction technique.

In order to study industrial runout table cooling, Liu (2001) designed and constructed a pilot-scale runout table facility in the High Head area of the Advanced Materials and Process Engineering Laboratory (AMPEL) at UBC. The schematic of the test rig is shown in Figure 2-5. Before the test, a steel plate with a size of 280×280×7.6 mm is heated in the furnace to a desired initial temperature and then cooled with water jets using industrial scale headers. One header can supply water for up to three pipe-type nozzles and the distance between nozzles can be adjusted between 50 to 90 mm. The height of the nozzles above the hot plate is also adjustable from 0.6 to 2 m. Heating units were installed in both the overhead water tank and the containment water tank to control the cooling water temperature, which was monitored by thermometers (not plotted in graph). The water in the containment tank was brought up to the overhead tank to keep the water level constant and hence the flow rate is constant.
With the facility described above, a number of experiments were conducted by Liu (2001), Hauksson (2001), and Meng (2002) for diverse purposes. Liu measured temperatures of the plate by embedding thermocouples about 1 mm below the surface in a line at various distances from the stagnation point. A two-dimensional inverse heat conduction model was developed to calculate heat fluxes and heat transfer coefficients at the surface. Hauksson and Meng measured both plate surface temperatures and temperatures at approximately 1 mm below the surface to quantify the heat flux at the plate surface using a direct method. Hauksson investigated water temperatures from 30 to 50 °C and Meng investigated higher cooling water temperatures from 60 to 95 °C, i.e. lower subcooling.

![Figure 2-5 Schematic of the pilot scale runout table](image-url)
Although it is more convenient to measure the surface temperature with surface mounted thermocouples, the fin effect due to heat transfer at the tip of the thermocouples to the water may enhance cooling in the immediate vicinity of the contact points (Park et al., 1991; Tszeng and Saraf, 2001; Li, 2003). Thus the surface mounted thermocouple "reads" a lower temperature than what it should be. Park performed a two-dimensional axisymmetric analysis and found an error of about 40 °C occurred within 0.1 s of immersion when quenching a 500 °C nickel cylinder into 20 °C for 1.7 s. Tszeng investigated the fin effect and suggested that its impact on the measured temperatures was greatly dependent on the wire diameter. The thinner the thermocouple wire, the more significant the influence.

Besides stationary plate tests described above, experimental techniques using moving plates have been developed by researchers such as Prodanovic et al. (2004), Chen et al. (1991), Han et al. (1991), Filipovic et al. (1992a and b, 1994), and Zumbrunnen et al. (1989).

2.4 Jet impingement heat transfer

Generally, the heat transfer behaviour in jet impingement cooling is described by a plot of the heat flux as a function of wall superheat, which is the difference between surface temperature and coolant saturation temperature. This kind of curve is commonly called the "boiling curve". Figure 2-6 is an illustration of a typical pool boiling curve. Initial cooling temperature (point A) is usually high enough to induce film boiling. When the cooling continues, the surface temperature drops to the Leidenfrost point (B), which corresponds to the minimum heat flux. At this point, the vapour film starts to collapse and liquid jets penetrate the vapour film and contact the metal wall. Further decrease of surface temperature causes an increase in the heat flux up to the second inflection point known as
the critical heat flux (point C, also maximum heat flux) due to more and more frequent water-solid contact. The region between points B and C is called the transition boiling region. Below the critical heat flux temperature is the nucleate boiling regime, in which the heat flux decreases sharply with the decrease of the surface temperature, since less and less bubbles are created and heat conduction to the liquid also decreases. At point D', the heat transfer mechanism changes from fully developed nucleate boiling to partially developed nucleate boiling. Below the bubble onset temperature (point D), bubble nucleation ceases and heat transfer is the predominant by single-phase convection.

![Figure 2-6 Schematic drawing of a typical pool boiling curve](image)

Similar boiling regions can be observed in jet impingement boiling curves, although the shape of the boiling curve may look quite different due to the effect of operating
parameters such as the local surface temperature, water temperature and flow rate, wall conditions such as roughness and oxidation, thermal properties of material and coolant, and geometry of cooling system. The experimental method also has a significant influence. Figures 2-7 and 2-8 show the boiling curves obtained under steady-state and transient experimental conditions respectively.

Figure 2-7 Boiling curves under steady-state cooling conditions: H=6 mm, $v_n=0.8$ m/s, $\Delta T_{sub}=16$ °C, x is the distance from the stagnation line (Robidou et al., 2003)

Figure 2-7 shows the boiling curves measured at three locations under steady-state planar jet impingement cooling condition by a temperature controlled system. Similar to the pool boiling curve illustrated in Figure 2-6, jet impingement boiling curves under steady-state condition can be divided into four distinct regimes with different mechanisms and degree of heat transfer. The boiling curves at stagnation were quite different from those obtained in the parallel flow region. Generally, the boiling curve at stagnation had a higher position than the boiling curve in the parallel flow region. At stagnation, transition boiling
took place for a wider range of wall superheat with heat fluxes almost as high as the critical heat flux. A local minimum heat flux close to the critical heat flux in the transition boiling regime was observed due to the microbubble emission boiling phenomenon (Robidou et al., 2002). In comparison, in the parallel flow region film boiling occupied a larger range of wall superheat and nucleate boiling initiates at a lower wall superheat.

![Diagram of boiling curves](image)

Figure 2-8 Local boiling curves obtained under transient cooling conditions: $v_n=3$ m/s, $\Delta T_{sub}=75$ °C, $T_{ini}=650$ °C, H=100 mm (Hall et al., 2001a)

Figure 2-8 shows the boiling curves obtained by a circular top jet with subcooling of 75 °C impinging from a nozzle of 100 mm to a hot copper block at a jet velocity of 3 m/s. Similar to the boiling curves shown in Figure 2-7 obtained at three locations under steady-state condition, larger heat fluxes were obtained closer to stagnation. The heat flux quickly
increased with the decrease of the surface temperature once the jet initially hit the test sample and then a local minimum appeared in the boiling curves in radial locations, where $r/d$ is over 1. In the impingement region ($r/d<1$) defined by the author, neither stable film boiling regime nor local minimum heat flux was observed for initial surface temperature of 650 °C. The “shoulder” of the boiling curve appearing at wall superheat of approximately 50 °C was attributed to the reheating of the surface during the transition of the boiling mechanism from nucleate boiling to single-phase convection by Hall et al. (2001a). The shoulder was also observed by Ishigai et al. (1978) in their transient experiments, while not reported in their steady-state experiments.

During jet cooling on the runout table, the strip surface is subjected to a wide range of temperatures, typically from 900 °C down to 300 °C. Therefore, it is expected that all different boiling mechanisms take place during this process. Thus, film boiling, transition boiling and nucleate boiling all need specific attention.

2.4.1 Film boiling

In the film boiling regime, from point B in Figure 2-6 and up, vapour film covers the solid surface and liquid-solid contact does not occur. The primary heat transfer mechanisms are forced convection within the vapour film and radiation through the vapour film. Several reviews of film boiling heat transfer such as the one by Kalinin et al. (1975) were published.

The occurrence and magnitude of film boiling heat transfer are mainly influenced by liquid subcooling and jet velocities.

Ishigai et al. (1978) obtained the entire boiling curve at stagnation by using a planar water jet to cool a steel plate from approximately 1000 °C. Data, as shown in Figure 2-9,
indicated that the heat flux increased with increasing subcooling and jet velocities. The effect of water subcooling on film boiling heat transfer was more apparent in the vicinity of the point of the minimum heat flux. Film boiling did not occur when subcooling was over 55 °C. Hall et al. (2001a) also did not observe the film boiling regime at stagnation with a circular water jet of 75 °C subcooling impacting on a copper surface at an initial temperature of 650 °C. However, it appeared in experiments with an initial surface temperature of 800 °C. Robidou et al. (2002) made similar conclusion by performing steady-state jet impingement experiments and reported that the effect of jet subcooling and velocities on film boiling heat transfer became smaller with increasing distance from stagnation.

Figure 2-9 Boiling curves at stagnation obtained under different subcooling and jet velocities (Ishigai et al., 1978)
Filipovic et al. (1995a) investigated the transient boiling phenomenon by quenching an oxygen-free copper block (25.2 mm thick, 38.1 mm wide, and 508 mm long) from an initial temperature of over 700 °C with a parallel wall jet using water as coolant. Temperatures were measured at ten equally spaced locations which were set along with the longitudinal centreline of the test cell and approximately 0.2 mm below the surface. The control-volume approach was employed to solve the two-dimensional, transient conduction problem and the heat fluxes on the surface were tuned to reduce the difference of the measured and the calculated temperatures. The obtained boiling curves revealed that the heat flux in the film boiling regime for a particular longitudinal location increased slightly with decreasing wall superheat. And the minimum heat flux (Leidenfrost point) was not obvious. This result is inconsistent with the saturated pool boiling curve, since convection rather than radiation is the primary heat transfer mechanism for forced flow with subcooling (Filipovic et al., 1994).

Li (2003) investigated the influence of the initial sample temperatures in the range of 400 to 1000 °C on the boiling curves at both water spray zone and water flow zone. As shown in Figure 2-10, it was found that after undergoing the initial cooling stage, the boiling curves starting from different initial temperatures higher than the Leidenfrost temperature almost coincided in the film boiling region.
Liu and Wang (2001) performed experimental and theoretical analysis on the unsteady film boiling heat transfer of water jet impacting on high temperature flat plate at stagnation. Experiments were conducted on a thin stainless steel plate (12 × 12 × 2 mm), which was cooled by a 10 mm diameter nozzle from an initial temperature of 1000 °C. The effect of subcooling on film boiling was found to be stronger than jet velocity, which was similar to what was presented by Ishigai et al. (1978). The model they developed is given by Equation (2-1)

\[ q_{FM} = 1.414 \cdot \frac{Re_j^{1/2} \cdot Pr_i^{1/6} \cdot (k_i \cdot k_e \cdot \Delta T_{sub} \cdot \Delta T_{sat})^{1/2}}{d_n} \]  

(2-1)

where \( Re_j \) is the Reynolds number in the form of

\[ Re_j = \frac{v_j d_n}{v_i} \]  

(2-2)
Liu and Wang applied Equation (2-1) to predict their experimental data. As shown in Figure 2-11, the curves with symbols were experimental boiling curves, the dashed lines (in Figure 2-11(b)) were calculated heat fluxes for water subcooling lower than 15 °C, and the solid lines were for water subcooling higher than 15 °C. The calculated values generally agreed with the experimental data at lower liquid subcooling. However, at higher subcooling the experimental data were considerably higher than the predicted.

![Graphs showing experimental and predicted heat fluxes](image)

Figure 2-11 Comparison between experimental data to the predicted at stagnation:

(a) subcooling = 15 °C; (b) impinging velocity = 3 m/s (Liu and Wang, 2001)

2.4.2 Minimum film boiling

At the point of the minimum heat flux (MHF), film boiling terminates and the heat flux starts to increase sharply with decreasing wall superheat. At this point, liquid-solid contact occurs and a gas-liquid-solid triple interface is generated. In jet impingement
cooling, the minimum film boiling temperature is often called the rewetting temperature (Filipovic et al., 1995c).

Knowledge of the rewetting temperature and heat flux is important to characterize boiling regimes during the jet impingement cooling process. Filipovic et al. (1995c) interpreted the rewetting temperature as the maximum change of the cooling rate and proposed a procedure to obtain the rewetting temperatures from measured cooling curves, i.e. the temperature variation with time. As graphically represented in Figure 2-12, the cooling curve was idealized by a modified inverse cotangent function:

\[ T = M \arccot(x - x^*) \]  

(2-3a)

where dimensionless quantities \( M \) and \( x^* \) are 4.25 and 20, respectively, obtained by curve fitting of experimental data. The first derivative of the function \( y \) would represent the cooling rate, and the second derivative would be the change of cooling rate with time. The first, second, and third derivatives of the function \( y \) are shown below:

\[ T' = \frac{M}{1 + (x - x^*)^2} \]  

(2-3b)

\[ T'' = \frac{2 \cdot M (x - x^*)}{[1 + (x - x^*)^2]^2} \]  

(2-3c)

\[ T''' = \frac{2 \cdot M [1 - 3(x - x^*)^2]}{[1 + (x - x^*)^2]^3} \]  

(2-3d)

The rewetting temperature obtained from this method is represented by a minimum of the function \( T'' \) (in Figure 2-3c). The method developed by Filipovic et al. is simple although the inverse cotangent function is not always suitable for describing the cooling curves and sometimes it is difficult to match experimental data.
Filipovic et al. (1995a) applied this approach to their experimental cooling curves. The results indicated that the rewetting temperature decreased with increasing longitudinal distances from the leading edge due to the decrease in local subcooling. The velocity of the rewetting front increased as approaching the trailing edge of the specimen since the stored energy reduced due to the decrease of specimen temperature in the zone downstream the rewetting front. Increasing jet subcooling and velocity resulted in the increase of the velocity of the rewetting front due to an increase in longitudinal heat conduction from the dry (downstream) portion to the wet (upstream) portion of the test specimen and an increase in efficiency of film boiling in the precursory zone.

Figure 2-12 Characteristics of a cooling curve and its derivatives: (a) function; (b) first derivative; (c) second derivative; (d) third derivative (Filipovic et al., 1995c)
Photographs from the experiments performed by Hatta et al. (1983) reveal that a black zone in the proximity of the stagnation point appears upon jet impingement. Boiling occurs around the black zone and the vapour bubbles generated run away on the plate. Therefore, the gas-liquid-solid triple line, i.e., the rewetting front, might exist near the circumference of the black zone. Hatta et al. measured the black zone radius in their photographs and obtained a correlation between the black zone radius $r_b$ (mm) and the cooling time $t$ (second):

$$r_b = a\sqrt{t}$$  \hspace{1cm} (2-4)

where $a$ is a constant. The velocity of the rewetting front, which is the first derivative of Equation (2-4), decreases with the increase of time.

Mitsutake and Monde (2001) filmed bottom jet impingement cooling process with a high speed camera. They measured the progression of the rewetting front and correlated the position of the rewetting front $r_{\text{wet}}$ with a power function of time as given by Equation (2-5)

$$r_{\text{wet}} = a \cdot t^n$$  \hspace{1cm} (2-5)

where variables $a$ and $n$ were related to jet velocity, subcooling, and thermal inertia of the material ($\rho c_p k$) and determined using a least mean square method by plotting them against subcooling or jet velocity for different materials. The exponent of $n$ changed from 0.4 to 0.5 under experimental conditions. Thus, similar to what was found by Hatta et al., the velocity of the rewetting front also decreased with increasing time. The rewetting front expanded faster with increasing jet subcooling and velocity, which agreed with the results obtained by Filipovic et al. (1995c).

The boiling curves obtained by Ishigai et al. (1978) showed that the MHF shifted to higher wall superheat and heat fluxes with the increase of subcooling (Figure 2-9). For
lower subcooling, the heat flux increased with the increase of jet velocity, but the minimum film boiling temperature was insensitive to jet velocity. Ishigai et al. correlated their data using an empirical equation considering the effects of subcooling and jet velocity as:

$$q^*_{_{MHF}} = 5.4 \times 10^4 \cdot (1 + 0.527 \Delta T_{_{sub}})v_n^{0.607}$$  \hspace{1cm} (2-6)

Equation (2-6) is valid for jet velocities and water subcooling in the range of 0.65 m/s ≤\(v_n\)≤3.5 m/s and 5 °C ≤\(\Delta T_{_{sub}}\)≤55 °C, respectively.

Ochi et al. (1984) incorporated the effect of the nozzle diameter into their empirical equation for minimum heat flux. In a range of nozzle diameters (5 mm≤\(d_n\)≤20 mm), jet velocities (2 m/s≤\(v_n\)≤7 m/s), and subcooling (5 °C ≤\(\Delta T_{_{sub}}\)≤45 °C), their correlation at stagnation can be expressed by Equation (2-7)

$$q^*_{_{MHF}} = 3.18 \times 10^5 (1 + 0.383 \Delta T_{_{sub}})(\frac{v_n}{d_n})^{0.828}$$  \hspace{1cm} (2-7)

Under steady-state condition, Robidou et al. (2002) also investigated the effect of subcooling and jet velocity on the MHF and they reported similar findings to those of Ishigai et al. (1978).

2.4.3 Transition boiling

Transition boiling, defined in the range from points B to C (Figure 2-6), is characterized by a reduction in surface heat flux with an increase in wall superheat. Compared with other boiling modes, transition boiling is the least understood area due to the inherent complexity of this phenomenon and the difficulties encountered in experimental studies. Generally, the mechanistic study of transition boiling is based on the concept suggested by Rohsenow (1952) and then deduced by Berenson (1962) according to the measurement of the boiling curves with varying surface characteristics. They suggested
that transition boiling was a combination of nucleate boiling during liquid-solid contact and film boiling during vapour-solid contact on the heating surface. The variation of heat transfer with wall superheat reflected a change in the fraction of time each boiling regime dominant at a given location.

Several reviews on transition boiling heat transfer have been presented by Auracher (1990), Dhir (1991), and Kalinin (1987). Reports on transition boiling under jet impingement cooling condition are sparse and will be discussed in the current work.

Robidou et al. (2002, 2003) performed experiments under steady-state heating condition to study the boiling heat transfer from a copper heater to a planar water jet. Surface temperature control technique was used and the surface temperature of the heater was increased stepwise in 2 or 5 °C steps from the saturation temperature to the temperature of film boiling occurrence. They investigated the effects of subcooling, jet velocity, and the nozzle to plate spacing. As shown in Figure 2-13, the boiling curve for a higher subcooling (17 °C) has a transition boiling regime with a larger range of wall superheat. A local minimum appeared in the transition boiling regime for both subcooling conditions and it occurred at a lower superheat as subcooling increased, since the microbubble emission boiling started sooner (Robidou et al., 2002). Figure 2-14 shows the effect of jet velocity on the boiling curve at stagnation. It can be seen that jet velocity did not have obvious influence on transition boiling heat transfer probably due to its small variation. For a surface jet, the nozzle to plate spacing had similar effect on transition boiling to that of increasing jet velocity. However, for an immersed jet, heat fluxes for transition boiling increased with increasing the nozzle to plate spacing at stagnation.
Using jet impingement tests with a planar jet, Ishigai et al. (1978) measured transient and steady-state boiling at the stagnation point. In the meantime, they recorded electrical conductance between the water and the test plate to detect the liquid-solid contact. Figure 2-9 shows the boiling curves for various jet velocities and subcooling. The data reveal that the boiling curves are considerably influenced by water subcooling and jet velocity. For a jet velocity of 2.1 m/s, as shown in Figure 2-9 (a), the curve moves toward higher wall superheat and heat fluxes. Besides, the shape of the boiling curve is dependent on subcooling. At high subcooling ($\Delta T_{sub}=55 \, ^\circ C$), the heat flux increases to a local maximum, followed by a “shoulder” or region near constant heat flux, and then increases monotonically to the maximum heat flux (Critical heat flux). Based on their observation, intermittent liquid-solid contact occurred and transition boiling heat transfer was thought to
be predominant at this flat region. While at low subcooling ($\Delta T_{\text{sub}}=5^\circ\text{C}$ and $15^\circ\text{C}$) the boiling curve declines from the start of quenching to the minimum heat flux and then increases monotonically to the critical heat flux.

![Graph](image)

**Figure 2-14** Effect of jet velocity at stagnation

As shown in Figure 2-9 (b) the effect of jet velocity on transition boiling heat transfer is not strong. The general trend is that transition boiling heat transfer increases with increased jet velocity.

Similar trends have been reported by Ochi et al. (1984), who performed similar tests to Ishigai et al. (1978) and found a shoulder in the transition boiling regime. Moreover, they measured heat transfer at stagnation for various nozzle diameters and found that the heat transfer decreased with increasing nozzle diameter at the stagnation point due to the decrease of velocity gradient.
Basically there are two types of modelling approaches used to correlate experimental data in the transition boiling region: mechanistic and empirical models.

The mechanistic model is usually established on the basis of existing physical mechanisms of transition boiling. Based on transition boiling mechanism proposed by Berenson (1962), many researchers (Kalinin, 1987; Ragheb and Cheng, 1984; Kao and Weisman, 1985; Pan et al., 1989,1992) assumed a dual-phase boiling combination model to describe transition heat transfer mathematically

\[ q_{TB} = q_i \times F + q_g \times (1 - F) \]  \hspace{1cm} (2-8)

where \( q_i \), \( q_g \) are heat fluxes between solid-liquid and solid-vapour contact, respectively. \( F \) represents the solid-liquid area fraction.

Generally, liquid has much better thermal transport properties than its associated vapour, so that the heat flux to the contacting liquid, \( q_i \) (by transient conduction, convection, and evaporation), is believed to be significantly greater than the heat flux to the vapour \( q_g \) (by transient conduction and convection). Dhir (1991) replaced nucleate boiling and film boiling heat fluxes in Equation (2-8) with the maximum and minimum heat fluxes respectively to obtain:

\[ q_{TB} = q_{CHF} \times F + q_{MHF} \times (1 - F) \]  \hspace{1cm} (2-9)

The mechanism that controls the liquid-solid contact, and therefore the instantaneous liquid contact-area faction, \( F \), remains unclear in pool boiling, as well as in forced flow situations.

Hernandez et al. (1995) developed a simple model of the macrolayer evaporation mechanism in conjunction with a semi-empirical fitting procedure to obtain the fractional liquid-solid contact area, \( F \). The liquid-solid contact heat transfer \( q_i \) was calculated from
extrapolation of the nucleate boiling regime, \( q_{NB} \). The heat flux between vapour-solid, \( q_s \), was approximated by extrapolation of the film boiling curve, \( q_{FB} \).

Mechanistic models are preferred since they are more physically based and thus less restricted to particular experimental conditions. However, due to the complexity of heat transfer taking place during runout table cooling, to the best knowledge of the author, there is still no sound theoretical model. Researchers usually involve several tuneable parameters for different cases in their models and these parameters are to some extent empirical.

Empirical models are based on statistical analysis of experimental data. They have a simple form, thus they are easy to be incorporated into a runout table model. However, the models are usually obtained under specific experimental or industrial conditions and hence probably only applicable to a specific mill layout.

Nishio and Auracher (1999) recommended a method to predict boiling heat transfer by using two anchor points, namely, the CHF-point and the MHF-point. It was assumed that a linear relationship existed in a log/log-plot between the two limiting points

\[
\frac{\ln(q'' / q''_{MHF})}{\ln(q''_{CHF} / q''_{MHF})} = \frac{\ln(\Delta T_{MHF} / \Delta T_{sat})}{\ln(\Delta T_{MHF} / \Delta T_{CHF})}
\]

(2-10)

Due to the difficulty in determining the quantities at the MHF, it was proposed to anchor the transition boiling curve only in the CHF point using Equation (2-11)

\[
\frac{q''}{q''_{CHF}} = 2.6\left(\frac{\Delta T}{\Delta T_{CHF}}\right)^n
\]

(2-11)

where the exponent \( n \) depends on fluid properties, the mass flux and the ratio of \( \Delta T_{sat} / \Delta T_{CHF} \). Apparently, the concept according to Equation (2-11) is not a generally applicable method.
Kandlikar et al. (1999) correlated pool boiling data from the literature for $F$ and $q'$ using Equations (2-12) and (2-13) and assumed $q_g$ to be equal to the minimum heat flux, $q_{MHF}$, i.e.

$$F = e^{- \frac{q_{MHF} \cdot \Delta T_{CHF}}{2}}$$

$$q' = q_{MHF} \frac{1 - 0.18 \cdot \frac{q_{MHF}}{q_{CHF}}}{0.82 \cdot \frac{\Delta T_{sub}}{\Delta T_{CHF}}}$$

where $q_{MHF}$ and $\Delta T_{CHF}$ are the minimum heat flux and its corresponding wall superheat, respectively.

### 2.4.4 Critical heat flux

Point C in Figure 2-6, which marks the termination of transition boiling and start of nucleate boiling, represents the critical heat flux. CHF occurs when the vapour blankets isolating the heated surface break down and the entire heated surface becomes available for liquid contact.

It is widely agreed that subcooling, and jet velocity enhance the critical heat fluxes by both steady-state and transient jet impingement cooling experiments (Robidou et al., 2002; Ishigai et al., 1978). Kumagai et al. (1995 a, b) measured the transient boiling heat transfer rate on the whole surface of a 20×150 mm rectangle copper plate during cooling from 400 to 100 °C under a planar water jet with subcooling of 0 to 50 °C. It was found that the cooling rate of the copper plate was very sensitive to the jet impinging velocity and subcooling, and the CHF increased monotonically with the increase of jet velocity and liquid subcooling, which was the same as the result of Miyasaka and Inada (1980). Water
subcooling had more significant influence on the value of CHF than jet velocity. For a jet velocity of 3.5 m/s, the CHF increased from 6 MW/m² for saturated water to more than 16 MW/m² for a subcooling of 50 °C at the stagnation point.

Models for the critical heat flux developed by several Japanese researchers under impinging or wall jet steady-state cooling conditions can be expressed by a general form as follows:

\[ q_{\text{CHF}} = a \cdot \left( \frac{\mu_l}{\mu_g} \right)^b \cdot \left( \frac{\sigma}{\mu_l \cdot v_n^2 \cdot L} \right)^c \cdot f \left( \frac{\mu_l}{\mu_g}, \frac{D}{d_n}, \Delta T_{\text{sub}} \right) \] (2-14)

where \( L \) is the characteristic dimension of the heater such as the length of heated surface for a rectangular heater, and the diameter for a round heater (D). The influence of density ratio \((\mu_l/\mu_g)\), geometry ratio \((D/d_n)\) and liquid subcooling is considered by the function \( f \). A summary of experimental conditions is given in Table 2-2, and the function \( f \) and constants \( a, b, \) and \( c \) reported by various researchers are given in Table 2-3. In Table 2-3, \( Ja \) is the Jakob number and \( Ja = (\rho_l / \rho_g) \left( c_p \cdot \Delta T_{\text{sub}} / h_{tg} \right) \).

Table 2-2 Experimental conditions

<table>
<thead>
<tr>
<th>Researcher</th>
<th>Test liquid</th>
<th>L (mm)</th>
<th>( v_n ) (m/s)</th>
<th>( \Delta T_{\text{sub}} ) (K)</th>
<th>( P ) (MPa)</th>
<th>Heated surface</th>
</tr>
</thead>
<tbody>
<tr>
<td>Monde, 1980</td>
<td>Water</td>
<td>5-36.4</td>
<td>0.3-15</td>
<td>0</td>
<td>At.</td>
<td>Upward</td>
</tr>
<tr>
<td>Monde, 1984</td>
<td>Water</td>
<td>36.4-54.1</td>
<td>0.3-15</td>
<td>0</td>
<td>5.9-15.6</td>
<td>Upward</td>
</tr>
<tr>
<td>Monde et al., 1994</td>
<td>Water</td>
<td>40, 60</td>
<td>5-16</td>
<td>0-115</td>
<td>1.3</td>
<td>Downward</td>
</tr>
<tr>
<td>Katto and Ishii, 1978</td>
<td>Water,</td>
<td>10-20</td>
<td>1.5-15</td>
<td>0</td>
<td>At.</td>
<td>Downward</td>
</tr>
<tr>
<td>Wang and Monde, 1997</td>
<td>Water,</td>
<td>40,60,80</td>
<td>3-15</td>
<td>0-60</td>
<td>0.1</td>
<td>Upward</td>
</tr>
<tr>
<td>Furuya et al., 1995</td>
<td>Water</td>
<td>2-3.2</td>
<td>&lt;27.5</td>
<td>0-60</td>
<td>At.</td>
<td>Upward</td>
</tr>
</tbody>
</table>

Note: At-Atmospheric pressure
Table 2-3 Summary of function f and constants a, b, and c in Equation (2-14)

<table>
<thead>
<tr>
<th>Researcher</th>
<th>Jet type</th>
<th>Constants</th>
<th>Function f</th>
</tr>
</thead>
<tbody>
<tr>
<td>Monde, 1980</td>
<td>Circular impinging jet (Saturated)</td>
<td>a=0.0757, b=0.725, c=1/3</td>
<td>(\frac{1}{1+0.00113(D/d_n)^2})</td>
</tr>
<tr>
<td>Monde, 1984</td>
<td>Circular impinging jet (Saturated)</td>
<td>a=0.28, b=0.645, c=0.343</td>
<td>((1 + L/d_n)^{-0.364})</td>
</tr>
<tr>
<td>Monde et al., 1994</td>
<td>Circular impinging jet (Subcooled)</td>
<td>a=0.28, b=0.645, c=0.343</td>
<td>(\frac{1 + \sqrt{1 + 4C \cdot Ja}}{2} \cdot (1 + L/d_n)^{-0.364})</td>
</tr>
<tr>
<td>Katto and Ishii, 1978</td>
<td>A wall jet (Saturated)</td>
<td>a=0.0164, b=0.867, c=1/3</td>
<td>1</td>
</tr>
<tr>
<td>Haramura and Katto, 1983</td>
<td>A wall jet (Saturated)</td>
<td>a=0.175, b=0.533, c=0.1/3</td>
<td>((1 + \rho_g / \rho_i)^{1/3})</td>
</tr>
<tr>
<td>Wang and Monde, 1997</td>
<td>A wall jet (Subcooled)</td>
<td>a=0.193, b=0.533, c=1/3</td>
<td>1 + 0.35 ((\rho_g / \rho_i)^{0.46} \cdot Ja)</td>
</tr>
<tr>
<td>Furuya et al., 1995</td>
<td>A plane impinging jet (Subcooled)</td>
<td>a=7.67, b=0.327, c=-1</td>
<td>215 \cdot Ja + \rho_i / \rho_g</td>
</tr>
</tbody>
</table>

Ishigai and Mizuno (1974) experimentally investigated the influence of jet subcooling and velocity on the critical heat flux using circular jet impinging on an upward facing flat surface. Subcooling and jet velocities varied in the range of \((45^\circ C<\Delta T_{sub}<80^\circ C)\) and \((1.3m/s<v_n<9.0m/s)\), respectively. They suggested a correlation in dimensional form, expressed by Equation (2-15).

\[
q_{CHF}^{*} = 0.0142 \cdot \left(\frac{v_n}{d_n}\right)^{0.34} \cdot \Delta T_{sub}^{1.15} \cdot 10^6
\]  

(2-15)

Miyasaka et al. (1980) presented another CHF correlation at the stagnation point based on their experiments, in which a subcooled jet impinged on a small downward facing...
heated surface (copper material). The effect of jet velocity in a range of 1.5 m/s to 15.3 m/s and subcooling of 30 °C to 85 °C was included in their correlation described using Equations (2-16) through (2-18).

\[
q_{CHF}^* = q_{CHF,sub, pool}^* \cdot (1 + 0.86 \cdot v_n^{0.28}) 
\]  

(2-16)

where

\[
q_{CHF,sub, pool}^* = q_{CHF,sub=0, pool}^* \cdot (1 + 0.112 \left( \frac{\rho_i}{\rho_g} \right)^{0.8} \cdot \left( \frac{c_{pl} \cdot \Delta T_{sub}}{h_g} \right)^{1.13}) 
\]  

(2-17)

and

\[
q_{CHF,sub=0, pool}^* = 0.16 \cdot h_g \cdot \rho_g \cdot \left( \frac{\sigma \cdot g \cdot (\rho_i - \rho_g)}{\rho_g} \right)^{1/4}
\]  

(2-18)

where \( \rho_i \) and \( \rho_g \) are the densities of saturated water and water vapour, respectively, \( c_{pl} \) is the specific heat of saturated water, \( h_g \) is the latent heat of evaporation, \( \sigma \) is the surface tension and \( g \) is the gravitational acceleration.

### 2.4.5 Nucleate boiling

Nucleate boiling, defined in the range from point C to D in Figure 2-6, is characterized by a reduction in surface heat flux with decreasing wall superheat. In the nucleate boiling regime, heat is released mainly through liquid evaporation. The nucleate boiling regime can be divided into two sub-regimes based on its slope. The regime C-D' is defined as fully developed nucleate boiling (FNB) region, in which nucleation sites cover the whole surface and the predominant heat transfer mechanism is bubble evaporation and coalescence. The heat flux decreases significantly with the decrease of wall superheat. The boiling curve in regime D'-D, which has a smaller slope, is called partially developed nucleate boiling (PNB). It is the transition region between nucleate boiling and single-phase
convection. In this regime, heat is released by bubble evaporation and transient conduction into liquid adjacent to the wall (Dhir, 1998). Point D denotes the onset of nucleate boiling (ONB). The ONB was found to shift to higher heat flux and wall superheat with increasing jet velocity and subcooling (Miyasaka, 1980). A similar effect of subcooling was also found by Robidou et al. (2002).

Robidou et al. (2002) found that in the fully developed nucleate boiling regime, the boiling curves in different locations merged and the heat flux seemed independent of the distance from the stagnation line. This phenomenon was consistent with what has been reported by Wolf et al. (1996). However, under transient cooling condition, this conclusion was not supported by the data of Hall et al. (2001a), as shown in Figure 2-8.

The effect of jet subcooling and velocity on nucleate boiling heat transfer has received wide attention. It is generally agreed that in fully developed nucleate boiling the heat flux is almost independent of both subcooling and jet velocity under steady-state cooling condition (Copeland, 1970; Monde and Katto, 1978; Ishigai et al., 1978; Wolf et al., 1996; Robidou et al., 2002). Monde and Katto (1978) investigated the effect of subcooling at stagnation. Compared to their previous work (Katto and Monde, 1974), which was performed using saturated water jet, they found that subcooling influenced nucleate boiling heat transfer at low wall superheat. But at higher wall superheat, the results under different subcooling were the same.

Under transient cooling condition, the results seem different. Kumagai et al. (1995b) reported that in the nucleate boiling regime, the boiling curves for subcooling of 14 to 50 °C at stagnation coincided with each other. While results obtained by Ochi et al. (1984) showed that a difference was evident in the nucleate boiling regime. Moreover, shoulders
were constantly observed in nucleate boiling curves at the stagnation point (Ishigai et al., 1978; Ochi et al., 1984; Hall et al, 2001a), while they are not found under steady-state cooling conditions.

The effect of wall superheat on the heat flux under both steady-state and transient cooling conditions is significant. Their correlation can be expressed by a general form:

\[ q_{\text{m}} (W/m^2) = C[\Delta T_{\text{sat}} (^\circ C)]^n \]

where C and n are constants depending on the process. Wolf et al. (1993) tabulated available data of C and n in Table 2-4 for free-surface and submerged jets with circular, planar and wall jet configurations.

**Table 2-4 Constants in Equation (2-19) (Wolf et al., 1993)**

<table>
<thead>
<tr>
<th>Author</th>
<th>Fluid</th>
<th>Jet type</th>
<th>C</th>
<th>n</th>
<th>Range of $\Delta T_{\text{sat}}$ ($^\circ C$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Copeland</td>
<td>Water</td>
<td>Circular-free</td>
<td>740</td>
<td>2.3</td>
<td>8--31</td>
</tr>
<tr>
<td>Ishigai et al.</td>
<td>Water</td>
<td>Planar-free</td>
<td>42</td>
<td>3.2</td>
<td>26--47</td>
</tr>
<tr>
<td>Katto and Ishii</td>
<td>Water</td>
<td>Planar-wall</td>
<td>130</td>
<td>3.0</td>
<td>21--33</td>
</tr>
<tr>
<td>Katto and Kunhiro</td>
<td>Water</td>
<td>Circular-sub</td>
<td>340</td>
<td>2.7</td>
<td>18--38</td>
</tr>
<tr>
<td>Katto and Monde</td>
<td>Water</td>
<td>Circular-free</td>
<td>450</td>
<td>2.7</td>
<td>18--46</td>
</tr>
<tr>
<td>Miyasaka et al.</td>
<td>Water</td>
<td>Planar-wall</td>
<td>79</td>
<td>3.0</td>
<td>26--90</td>
</tr>
<tr>
<td>Toda and Uchida</td>
<td>Water</td>
<td>Planar-wall</td>
<td>6100</td>
<td>1.42</td>
<td>16--68</td>
</tr>
<tr>
<td>Katsuta and Kurose</td>
<td>R-113</td>
<td>Circular-free</td>
<td>2.93E-6</td>
<td>7.4</td>
<td>24--31</td>
</tr>
<tr>
<td>Ma and Bergles</td>
<td>R-113</td>
<td>Circular-sub</td>
<td>0.15</td>
<td>4.4</td>
<td>26--33</td>
</tr>
<tr>
<td>Monde and Katto</td>
<td>R-113</td>
<td>Circular-free</td>
<td>790</td>
<td>2.0</td>
<td>15--30</td>
</tr>
<tr>
<td>Ruch and Holman</td>
<td>R-113</td>
<td>Circular-free</td>
<td>467</td>
<td>1.95</td>
<td>17--44</td>
</tr>
</tbody>
</table>

Note: sub-submerged
3. Test rig and procedures

Over the past decades many jet impingement experiments have been done under steady-state heat transfer conditions and using test rig with samples and nozzles at much smaller scale than those of a real runout table. The results obtained through such tests may not be applicable to an industrial runout table. In order to simulate industrial runout table cooling, Liu (2001) initiated the design and construction of a pilot-scale runout table facility in the Advanced Materials and Process Engineering Laboratory (AMPEL) at UBC. Liu (2001), Hauksson (2001) and Meng (2002) performed tests with a single jet impinging on a stationary upward facing plate.

In the present study modifications to the existing facility were made and they include the design of the bottom nozzle, the plate carrier, and the development of experimental procedures.

3.1 Bottom jet cooling system

The bottom jet cooling system is an extension to what was built by Liu (2001) and Prodanovic et al. (2003). It consists of tubes, pipes, valves, flow meter, and tube, pipe fittings. As shown in Figure 3-1 (in comparison to Figure 2-5), the modification was made in the region outlined by the dashed line. The valve 1 was placed in between the pump and the overhead tank and used to control the flow to the overhead tank. The valve 2 is used to control the flow rate of the bottom jet, which can be measured by the flow meter downstream. The valve 3 downstream of the flow meter is employed to start and stop the experiment. The picture in Figure 3-2 shows the bottom jet cooling system. The facility
also includes an overhead tank (upper tank), a containment tank (lower tank), a heater in the overhead tank, a pump, and a furnace. The specifications are given in Table 3-1.

Figure 3-1 Schematic of the test facility
Table 3-1 Specifications of the facility

<table>
<thead>
<tr>
<th>Facility</th>
<th>Specification</th>
</tr>
</thead>
<tbody>
<tr>
<td>Electric heat furnace</td>
<td>Lindberg/Blue M. (240V, 25A, 5.8kW, 60Hz)</td>
</tr>
<tr>
<td></td>
<td>Maximum heating temperature: 1200±5°C</td>
</tr>
<tr>
<td></td>
<td>Heated area: 584x30x57mm</td>
</tr>
<tr>
<td>Water pump</td>
<td>11.2 kW, 1735 RPM, 60 Hz</td>
</tr>
<tr>
<td>Upper tank</td>
<td>1.5x1.5x1.0 m</td>
</tr>
<tr>
<td>Heater in upper tank</td>
<td>30 kW</td>
</tr>
<tr>
<td>Lower tank</td>
<td>2x0.7x1.2 m</td>
</tr>
</tbody>
</table>

The overhead tank with a capacity of 1350 l is located at the top of a 6.5 m tower and used to maintain the predetermined temperature of the water as high as 95 °C used for
the test (Meng, 2002). The containment tank supplies water for the bottom nozzle through the pump. The pump is also used to recycle the water between the overhead and containment tanks when necessary.

A carefully cut 7/8" commercial 304 stainless steel tube with inner diameter of 19mm and the length of 40cm was used as the bottom nozzle, which is connected with the pump through several tubes, pipes, and tube, pipe fittings. The distance between nozzle and test plate was adjustable and set to 88mm. The jet flow rate was measured by an OMEGA FTB-905 turbine flow meter, and controlled by a valve upstream from the flow meter, as shown in Figure 3-1. The valve downstream of the flow meter was used to start and stop the test.

A Lindberg/Blue M 5.8kW furnace was used to heat up the test plate to the required initial temperature. During operation, it was filled and pressurized by nitrogen gas to prevent oxidation of the steel plate in the furnace.

3.2 Design of the carrier

The lower carrier was mounted underneath the jet in the test section (shown in Figure 3-3) and supported the instrumented plate during the test.

The test plate was mounted on the upper carrier. The upper carrier was used to deliver the plate from the furnace to the test section and to protect the thermocouple connections during tests. Figure 3-3 shows a schematic view of the upper carrier. The plate was bolted on T shaped steel, which was welded to the steel handle. Thermocouples wires, connected to the data acquisition system on one end, were placed inside the handle, and spot welded on both the surface of the plate and the flat bottom of the holes in the plate. Insulation (ceramic fibre) was placed between the test plate and the T shaped steel to avoid
heat transfer by conduction to the carrier, which makes it valid to assume adiabatic boundary condition between the plate and the carrier. On both sides of T shaped steels, two steel bars (not plotted in Figure 3-3) were welded to strengthen the construction. Insulation (ceramic fibre) was also placed between the test plate and the steel bars.

3.3 Instrumentation of the test plate

DQSK (Drawing Quality Special Killed) sheets from Stelco Inc. were cut to plates (test samples) with the size of 280x250x7.6mm. The chemical composition of the DQSK steel (Hauksson, 2001) is listed in Table 3-2.

Table 3-2 Chemical composition of DQSK steel, wt%

<table>
<thead>
<tr>
<th>Steel</th>
<th>C</th>
<th>Mn</th>
<th>P</th>
<th>S</th>
<th>Si</th>
<th>Cr</th>
<th>Ni</th>
<th>Mo</th>
<th>Al</th>
<th>N</th>
</tr>
</thead>
<tbody>
<tr>
<td>DQSK</td>
<td>0.06</td>
<td>0.24</td>
<td>0.005</td>
<td>0.011</td>
<td>0.006</td>
<td></td>
<td></td>
<td>0.041</td>
<td>0.0035</td>
<td></td>
</tr>
</tbody>
</table>

Each plate was used only once in the as rolled condition as the plate might oxide, bend and warp during the cooling process. Since the bending was small and took place at the end of the cooling process, its effect on heat transfer and jet flow was neglected.
Type K thermocouples (Omega INC-K-Mo-1.6mm) were employed to measure the transient temperature response. The diameter of the thermocouple is approximately 1.6 mm. The temperature measurements were made at eight locations of the plate for every test. As shown in Figure 3-4, the first location was at the centre of the plate and the other seven locations were evenly distributed in different rays and at 15.9 mm increments radially outward from the centre. Such an arrangement of thermocouples was first suggested by Hauksson (2001) to minimize the influence of one thermocouple on the others downstream.

Each location included a pair of intrinsic thermocouples with one mounted on the bottom surface and the other mounted in the flat bottom hole around 1 mm above the bottom surface. The diameter of the holes was the same as that of the thermocouple (1.6 mm). Although collected during experiments, the surface temperature data were not used for the analysis due to apparent measurement errors associated with the fin effect. Details on the uncertainties of using thermocouples for the surface temperature measurement under high heat flux conditions were discussed elsewhere (Li, 2002; Tszeng et al., 2003). However, in order to validate experimental procedures and assumptions presented herein, a study on the effect of the surface thermocouple wire on the accuracy of internal temperature measurements has been conducted by the author and presented in Appendix D. It has been concluded that this effect was minor, i.e. within the error band of the internal temperature measurements. Hence, the inverse heat conduction method was employed to back calculate surface heat fluxes and temperatures.
Signals from sixteen thermocouples and from the flow meter were collected via two external data acquisition boards (I-Net100) and transferred to a PC by DASYLab 7.0 data acquisition software. In order to capture potentially rapid changes of the temperature curve, 1000 data per second were recorded.

Experimental errors were estimated according to the equipment specifications and are presented in Table 3-3.
Table 3-3 Accuracy of the measurement

<table>
<thead>
<tr>
<th>Quantity</th>
<th>Measurement error</th>
</tr>
</thead>
<tbody>
<tr>
<td>Temperature</td>
<td>±2 °C ( T &lt; 277 °C ) \hspace{1cm} ±0.75% ( T \geq 277 °C ) (Loveday, 1982)</td>
</tr>
<tr>
<td>Flow rate</td>
<td>±0.5% of the reading</td>
</tr>
<tr>
<td>Thermocouple depth</td>
<td>±0.003mm</td>
</tr>
<tr>
<td>Thermocouple location</td>
<td>±0.1mm (with respect to central point)</td>
</tr>
</tbody>
</table>

3.4 Experimental procedures

The plate was cleaned with methyl alcohol before it was instrumented. The thickness of the plate was measured at several places around each hole. The depths of all flat bottom holes were measured with a micrometer.

The plate was bolted to the upper carrier. The plate after instrumentation was put on the lower carrier to test the level the right location to put the upper carrier so that the jet will hit the centre of the plate. Then it was put into the furnace and heated up to about 920 °C (target temperature). This temperature is about 70 °C higher than the temperature to initiate the test (850 °C, initial temperature) since the plate will lose temperature due to air-cooling during its moving from the furnace to the cooling section.

The heating process required about 3 hours. The plate was kept in the furnace for about half an hour after the target temperature had been reached to ensure that the whole plate had a uniform temperature. Then the plate was taken out of the furnace and positioned in the test section.

When the plate was put in the right position and its temperature had dropped to the initial temperature (850 °C), the flow was turned on. Data collection started several seconds
before the flow was turned on and stopped when the internal temperature at location 8
dropped close to the water temperature. Each experiment took about 2 minutes, depending
on flow rate and subcooling.

The cooling process of the tests except flow rate of 60 l/min was filmed by a Canon
video camera. The video film was converted to a digital film and digital images were
extracted from the digital film for estimating the initial size of the impingement zone as
well as the progression of the rewetting front during cooling.
4. Data processing

With measured temperatures, heat fluxes can be calculated by two methods: direct method and inverse method. Hauksson (2001) developed a direct method based on the Fourier's law of heat conduction, which was also called two-thermocouple method. Temperatures at both surface (measured surface temperatures) and interior location very close to the surface were utilized. The interior temperatures were calculated by using a one-dimensional finite difference heat conduction model developed by Crank-Nicolson (1947). The accuracy of the model may be influenced by the following aspects. First, measured surface temperatures may be not accurate; second, the model is one dimensional, which neglects heat conduction in the radial direction. However, measured temperatures revealed that significant temperature gradients existed in the radial direction of the test plate during circular jet impingement cooling process.

Among several inverse heat conduction models, function specification and gradient/adjoint methods are the most widely used approaches (Dowding and Beck, 1999). The function specification method assumes a prescribed functional form for the heat flux within a future interval (Beck et al. 1985) and solves the problem in a sequential manner (Osman, et al. 1989, 1990, 1999). Gradient/adjoint methods require solutions of three equations: direct, adjoint and sensitivity for temperature, search direction and step size in the search direction on the whole time domain, respectively (Jarny et al. 1991). The sequential function specification method usually uses Tikhonov regulation to stabilize the solution. The gradient method employs Tikhonov or iterative regularization (Alifanov and

Beck et al. (1996) compared two inverse heat conduction methods introduced above using experimental data. They found that both methods gave accurate results. The advantage of the sequential function specification method (SFSM) is that it is conceptually simple and computationally efficient in single parameter estimations. In addition, the SFSM takes advantage of the sequential nature of the problem. But the requirement of a prescribed functional form restricts the application of the SFSM (Dowding and Beck, 1999). The gradient/adjoint methods (GAM) have more mathematical foundations (Dowding et al. 1998) and do not require assumption of functional forms. These methods have more computational efficiency for linear multidimensional problems. A sequential gradient method has been suggested for nonlinear problems and the new method is supposed to be more promising than the whole domain gradient method (Dowding and Beck, 1999).

The jet impingement cooling process is a highly transient problem and substantial thermal gradient exists in the plate under such cooling conditions. Due to the progression of the rewetting front on the surface of the plate, the distribution of the heat flux on the boundary varies with both time and space. All these factors make the problem more complicated. To date, the application of multidimensional inverse heat conduction model to analyze jet impingement cooling process is very limited.

In the current study, function specification and zeroth-order Tikhonov regularization method was combined to solve the inverse heat conduction problem with a sequential-in-time concept used to improve the computational efficiency. The future information was included in the sequential-in-time method to consider the lag of time and hence enhance the
accuracy. The temperature history was obtained by using a two-dimensional axisymmetric finite element method.

In this model, thermal properties were temperature dependent, which made the problem non-linear. However, since the problem was solved on very short time intervals, the assumption that the temperature-dependent material properties were constant during the sequential interval (R-future time steps) was valid. At each interval, thermal properties were estimated only at the initial time step for linearization. This quasi-linearization method has significantly reduced the computational time.

The sensitivity equations were obtained by differentiating equations for the direct problem. Due to the similarities of the equations for sensitivity problem and the direct problem, the conductance matrix, capacitance matrix and the solver used in the direct problem could be applied to the sensitivity problem with the modification of the initial and boundary conditions. Therefore, the required computations were reduced again.

4.1 Inverse heat conduction model (IHCM)

Solution of the inverse heat conduction problem consists of two steps: solving the direct problem and solving the inverse problem. These two steps are repeated until the differences between the calculated and the measured heat fluxes are smaller than a pre-set value (iteration process). When solving direct problem, the temperatures at measured locations are calculated using pre-assumed heat fluxes. In the second step, the calculated and the measured temperatures are compared and their differences are used for calculating the increments of the assumed heat fluxes. The procedure to solve the inverse heat conduction problem is as follows:
1) Input initial data: initial temperature profile, measured temperatures at different time steps \(Y\), assumed heat flux components \(q^0\).

2) Call direct problem solver for \(R\) times (Equation (4-4a)) for temperature \(T\).

3) Solve sensitivity equation (Equation (B-7)) \(M\) times (\(M\) is number of unknown surface heat flux components).

4) Calculate sensitivity matrix \(X\) (defined by Equation (B-5)) and its transpose \(X^T\).

5) Calculate \(\Delta q\), increments of assumed heat flux components, according to Equation (4-5) and update original heat fluxes according to \(q = q^0 + \Delta q\).

6) Repeat procedure 1-5 until the sum of square error between the computed and measured temperatures is smaller than a pre-set value (1e-4).

**4.1.1 Direct Problem**

The heat transfer in the plate during the bottom jet impingement cooling can be approximately described by a two-dimensional heat conduction equation

\[
\frac{1}{r} \frac{\partial}{\partial r} \left( kr \frac{\partial T}{\partial r} \right) + \frac{\partial}{\partial z} \left( k \frac{\partial T}{\partial z} \right) + Q = \rho c_p \frac{\partial T}{\partial t} \tag{4-1}
\]

where \(k\) is heat conductivity, \(\rho\) the density, \(c_p\) the specific heat of the material, \(Q\) the internal heat generation rate per unit volume, \(T\) the temperature, and \(t\) is the time.

The problem is schematically shown in Figure 4-1. Since the jet is circular, heat conduction in an axisymmetric geometry is considered. Only half of the plate is selected due to the symmetry. Although the heat transfer field can be assumed to be axisymmetric, the geometry of the domain is not strictly axisymmetric.
Experimental results

$S_3: \ h=\delta_h+\delta_v, \ \delta_v=\alpha_s(T^2+T_o^2)(T+T_o)$

Adiabatic in the holes

Figure 4-1 Schematic description of the problem

The symbols $S_i$ (i=1,2,3,4) denote the domain boundaries. The boundaries $S_2$, $S_3$, $S_4$ have known boundary conditions, which are

$S_2: \ \ \ q = -k \left( \frac{\partial T}{\partial r} \right)$ \hspace{1cm} \text{(4-2a)}

$S_3: \ \ \ h = h_r + h_c$ \hspace{1cm} \text{(4-2b)}

\hspace{1cm} $h_r = \sigma_s \varepsilon (T^2 + T_o^2)(T + T_o)$ \hspace{1cm} \text{(4-2c)}

$S_4: \ \ \ q = 0$ \hspace{1cm} \text{(4-2d)}

The boundary $S_2$ is not a real boundary between the plate and surrounding air. It is the boundary between the cylinder extracted from the plate and the original plate. Thus, the heat flux at this boundary was calculated according to Fourier’s law of heat conduction.

The boundary $S_3$ is the top surface of the plate, which is assumed free of water and such that it is exposed to air-cooling. Since the heat loss due to the convection in stationary air is smaller compared to the radiation at elevated temperature, the second part in Equation (4-
2b) is neglected in the analysis. In Equation (4-2c), $\sigma_{SB}$ is the Stefan-Boltzmann constant, taken as $5.67 \times 10^{-8} \text{W/m}^2\text{C}^4$, $T_\infty$ is the ambient temperature, $\varepsilon$ is the emissivity calculated by Equation (4-2e) (Sun et al., 2002), i.e.

$$\varepsilon = \frac{T}{1000} (0.125 \frac{T}{1000} - 0.38) + 1.1 \quad (4-2e)$$

It can be assumed that the highest temperature gradient is in the direction of the thickness of the plate. The surface temperature drops quickly due to high heat fluxes while the interior of the plate remains hot. The heat is, then, supplied from the interior of the plate and conveyed by conduction to the surface. If an insulated hole exists in the material, it will reduce the heat supply to the surface. Hence, the thermocouple holes must be taken into account when determining the domain for numerical calculations. The heat transfer in the holes is assumed adiabatic since the thermal resistance in contact between the insulator and the steel plate is large.

At the line of symmetry the heat flux on the boundary $S_4$ is 0. The space-and-time dependent heat flux components $q_j(r,t)$ ($j = 1 - 8$) on the boundary $S_1$ are unknown and to be estimated.

The internal heat is generated due to the decomposition of austenite during the quench operation. However, it is neglected in the current study since it requires researching the mechanism of the decomposition of austenite, modelling the decomposition of austenite, and testing the models, which is far beyond the purpose of this study. Ignoring latent heat evolution associated with decomposition of austenite during the quench operation may result in lower calculated heat fluxes than what they really are since more heat should be removed from the boundary to balance the latent heat.

The initial condition specifies the temperature distribution at time $t_0$. 

\[ T(r, z, t_0) = T_0(r, z) \]  \hspace{1cm} (4-3)

The initial temperature distribution is assumed parabolic in the direction of the plate thickness and linear in the radial direction due to the air-cooling during the delivery of the plate from the furnace to the jet line.

The partial differential Equation (4-1) with its boundary conditions transformed to integration equation and a set of non-linear equations eventually (refer to Appendix A for details), which are expressed by

\[
[A] \cdot \{T\}_{n+1} = \{B\} \tag{4-4a}
\]

where

\[
[A] = \theta \cdot [K(T_\theta, t_\theta)] + \frac{[C(T_\theta)]}{\Delta t} \tag{4-4b}
\]

\[
\{B\} = \left[-(1-\theta)[K(T_\theta, t_\theta)] + \frac{[C(T_\theta)]}{\Delta t}\right]\{T\}_n + \{F(T_\theta, t_\theta)\} \tag{4-4c}
\]

\[
\{F(T_\theta, t_\theta)\} = \theta\{F\}_{n+1} + (1-\theta)\{F\}_n \tag{4-4d}
\]

\[
\{T\}_\theta = (1-\theta)\{T\}_n + \theta\{T\}_{n+1} \tag{4-4e}
\]

Since the thermal properties are temperature dependent, \([A]\) and \(\{B\}\) are functions of temperature, and the equations are non-linear. Usually, Newton-Raphson method is used to solve such a system of equations. This method requires long computational times.

However, upon noticing that the changes of temperature-dependent material properties during a sequential interval (R-future time step) are small, the problem can be regarded as linear problem during a sequential interval. Thermal properties are estimated at the initial time step of each interval.
4.1.2 Update boundary conditions

The update to the assumed boundary conditions \( \Delta q \) is realized by solving the following system of equations:

\[
[D + \alpha I] \cdot \{\Delta q\} = \{f\}
\]  

(4-5)

where \( \alpha \) is regulation parameter, matrix \( D = X \cdot X^T \), refer to Appendix B for details.

The sensitivity coefficient \( x_{ij} \) in sensitivity matrix \( X \) denotes the changes of \( T_i \) with respect to the change in each of \( q_j \). It was obtained by solving sensitivity equations, which are derived by differentiating both sides of the temperature equations with respect to the unknown heat fluxes. Due to the similarities of the equations for the sensitivity problem to those for the direct problem, the conductance matrix, capacitance matrix and the solver used in the direct problem are used to solve the sensitivity problem with the modification of the initial and boundary conditions.

In practice, the number of spatial-dependent components that can be accurately estimated is less than or equal to the number of sensors, since sufficient information on the region of this component is required. Therefore, with temperatures known at eight locations, eight heat flux components corresponding to eight locations (as shown in Figure 4-1) can be estimated. However, knowledge of the heat fluxes (or temperature distribution) in between eight components is very important, since these varied heat fluxes may result in large radial heat conduction. The abrupt temperature drop at each element can be associated with the arrival of the rewetting front, and the rewetting front itself is considered to be dynamic boundary between the stable vapour-solid contact area and the stable liquid-solid contact area (Leidenfrost point). Assumptions on the spatial distribution of the heat fluxes are necessary for the elements in between the elements where heat flux components \( q_j(r,t) \) (j...
Experimental results

=1 - 8) are applied. These assumptions are based on the understanding of the heat transfer mechanisms during bottom jet cooling process, which include:

1) Heat fluxes $q_i(r,t)$ in the first zone are estimated by linear interpolation using the neighbouring heat flux components

2) Before the rewetting front arrives, heat fluxes $q_i(r,t)$ in the second zone are assumed to be equal to the heat flux at the neighbouring location downstream

3) After the rewetting front arrives, heat fluxes $q_i(r,t)$ in the second zone equals the heat flux of the neighbouring component upstream at the time when the rewetting front reaches that component

4) Before the rewetting front arrives, heat fluxes $q_i(r,t)$ in the third zone are assumed to be equal to the heat flux under the condition of air-cooling

5) After the rewetting front arrives, heat fluxes $q_i(r,t)$ in the third zone equals the heat flux of the neighbouring component upstream at the time when the rewetting front researches that component

The following example is given to help understand the above method to handle the unknown boundary conditions. As shown in Figure 4-2 (a), assuming at time $t_1$ the rewetting front is at where heat flux component $q_4$ is, certain heat flux $q_j$ between $q_4$ and $q_5$ is equal to $q_5$. At next time $t_2$ when the rewetting front reaches where the heat flux $q_i$ is (see Figure 4-2 (b)), the heat flux $q_i$ takes the value of $q_4$ at time $t_1$ rather than at current time ($t_2$).
Experimental results

Time = $t_1$

\[ S_3 \ h = h_r + h_c \quad h_c = \sigma \varepsilon (T^2 + T_a^2)(T + T_a) \]

Adiabatic in the holes

\[ q(r, t) = q_s \]

Rewetting front

Figure 4-2 Methods to handle the boundary conditions

These assumptions are valid when the water jet hits the centre of the plate and the rewetting front propagates in a perfect circle. In addition, the distance of neighbouring measurement location should be close enough that the heat fluxes are not significantly different before the arrival of the rewetting front. The location of the rewetting front was
determined based on the measured temperatures. The instant of time, which corresponded to the minimum of the second derivative of the cooling curves (maximum change of the cooling rate – suggesting the change from film boiling to transition boiling mode), indicated the arrival of the rewetting front (Filipovic, 1995c). The method was confirmed by image analyses.

**4.2 Material properties**

Thermal properties of the DQSK steel were given by Hauksson (2001). Heat conductivity is a linear function of temperature and changes in density and specific heat with temperature are considered to be negligible.

Conductivity: \( k = 60.571 - 0.03849T \) (W/m °C)

Density: \( \rho = 7800 \) (kg/m³)

Specific heat: \( c_p = 470 \) (J/kg °C)

**4.3 Verification of IHCM**

The verification of two-dimensional finite element model for the direct problem was performed by comparing results with the analytical solution for one-dimensional heat conduction problems and one-dimensional finite difference solutions using MATLAB.

The verification of the inverse heat conduction model (IHCM) was performed by comparing calculated boundary conditions with those applied. As shown in Figure 4-3, boundary condition \( q_1 \) was applied to the bottom of the first element at the start of the calculation. This boundary condition moved to the right element by element at a speed of 1.4 mm/s, by which the effect of the progression of the rewetting front is replicated. Under these boundary conditions, temperature profiles at location 1 and 2 were obtained as shown
in Figure 4-4. With internal temperature known at location 1 and 2, the boundary conditions at the bottom corresponding to each location were estimated by the inverse heat conduction model. The results are shown in Figure 4-5. It can be seen that the calculated heat fluxes by the current model match the applied very well.

Another assumption used in the IHC model was that the location of the rewetting front is associated with the second derivative of the cooling curve (function of temperature with time). This assumption was used in the IHC model to calculate the rate of progression of the rewetting front. The procedure was validated by calculating the second derivative of the temperature profiles shown in Figure 4-4 and comparing the obtained results with the assumed progression of the rewetting front of 1.4mm/s. Very good agreement was obtained.
Experimental results

\[ h = \sigma e (T_1^2 + T_2^2)(T + T_\infty), \text{ Insulation in the hole} \]

(a)

(b)

\[ q_1 = 0 \]

1.4 mm/s

\[ q_1 \text{- Applied heat flux} \]

Figure 4-3 Verify the inverse heat conduction model
Experimental results

Figure 4-4 Calculated temperatures

Figure 4-5 Comparison of the calculated heat fluxes with the applied
5. Experimental results

Twelve tests were performed on the UBC pilot ROT using DQSK steel plates. The effects of the cooling water temperature and flow rate were investigated. The experimental matrix is shown in Table 5-1.

Table 5-1 Experimental matrix

<table>
<thead>
<tr>
<th></th>
<th>20 °C</th>
<th>30 °C</th>
<th>40 °C</th>
<th>50 °C</th>
</tr>
</thead>
<tbody>
<tr>
<td>30 l/min</td>
<td>#4</td>
<td>#1</td>
<td>#5</td>
<td>#6</td>
</tr>
<tr>
<td>45 l/min</td>
<td>#7</td>
<td>#2</td>
<td>#8</td>
<td>#9</td>
</tr>
<tr>
<td>60 l/min</td>
<td>#10</td>
<td>#3</td>
<td>#11</td>
<td>#12</td>
</tr>
</tbody>
</table>

5.1 Visual observation

The initial colour of the plate before jet impingement was bright red, the surface was clean without obvious scale patches. Upon start of the experiment the water jet hits the bottom of the plate, the area just above the jet turned black immediately (see Figure 5-1a). It was assumed that the colour of the surface was associated with the surface temperature, the dark area corresponded to the zone where very large heat fluxes could be immediately achieved and a significant temperature drop could be observed. Also, one could speculate that very large temperature gradients would exist in the radial direction across the boundary of the dark zone, likely indicating the division between liquid-solid contact zone and vapour-solid contact zone. The size of the initial black area depended on the water flow rate and water temperature. It could be observed that shortly after the start of the experiment water covered the entire surface (phase 1) until a stable vapour layer was established (phase 2). The water covered a certain ring somewhat larger that the black zone itself and then was diverted from the plate in an arc without getting in contact with the rest of the plate (see
Figure 5-1b). Shortly after the start of phase 2 the black zone progressed radially outwards (phase 3). As shown in Figure 5-1c, the black region was apparently bigger than that at 2.58 seconds after impingement. The experiment ended after the temperature at location 8 dropped below saturation (monitored on the computer) and the black zone covered the entire surface.
Experimental results

(a)

(b)
Figure 5-1 Images during jet impingement cooling for test #5

(T\text{water}=40\,^\circ\text{C}, \text{Flow rate}=30\text{l/min})

a - image at 0.18 seconds after impingement
b - image at 2.58 seconds after impingement
c - image at 28.2 seconds after impingement

5.2 Analysis of the cooling curves

Cooling curves represent the surface or internal measured temperature variations with time. Figure 5-2 shows the cooling curves for test #8 measured at eight different radial internal locations from the centre of the test plate; the test was performed with a water flow rate and water temperature of 45 l/min and 40°C, respectively.
Experimental results

Figure 5-2 Measured internal temperatures in test #8 (r is radial distance from the centre of the plate)

Based on the visual observation of the tests and analysis of the cooling curves, three different cooling zones have been observed, which are sketched in Figure 5-3.

The cooling curves of test #8 may be regarded as typical for the given experimental matrix. Upon impingement the temperature at the first 2 locations (zone one) dropped most rapidly to about 300 °C at almost the same rate. Then they changed relatively slowly. The zone where TC1 and TC2 were located is considered to be the impingement zone, which can be different from the hydrodynamic impingement zone based on the pressure gradient. The radius of the impingement zone seems to be between TC2 and TC3, i.e. smaller than 31.75 mm or 1.67 times the nozzle diameter but larger than 15.9 mm or 0.84 nozzle diameters.
Experimental results

Figure 5-3 Heat transfer zones for bottom jet impingement cooling

The temperatures at the next three locations (zone two) exhibited initially rapid drops and the magnitude of these initial drops decreased with increased distance from the stagnation point. Then, the decrease of temperature became gradual until a rapid second temperature drop took place. The moderate temperature changes likely resulted from the formation of the vapour film on the surface of the test plate, which reduced the opportunity of the water-solid contact. When the surface temperature of the plate was not high enough to support the existence of a stable vapour film, the cooling curves showed the second sudden drops. This indicated the rewetting of the surface.

Since the vapour film around the impingement zone deflected the water from the plate rather than flow parallel to the plate. The cooling curves at locations 6,7 and 8 (zone three) showed the trend of small recovery after initial wetting. After the short recovery, the temperature started decreasing smoothly. The principal heat transfer at this time was by
radiation and convection. The cooling rate (The slope of the cooling curve) was close to that of air-cooling.

After a period of time, depending on the distance from the stagnation, the second sharp drop occurred from location 6 to 8 sequentially, which indicated the arrival of the rewetting front. After the second rapid drop, the slopes of cooling curves changed to much smaller values when the temperature went down below around 200 °C, suggesting a change in heat transfer mode to partial nucleate boiling or single phase forced convection. Finally, all eight curves merged into a uniform temperature.

Similar shapes of the cooling curves could be observed in all the tests with small variations caused by the water temperature and flow rate. As shown in Figures 5-4 and 5-5, with increasing water temperature and decreasing flow rate, the area of the first and second zone is reduced.

5.3 The rewetting front

The progression of the rewetting front was determined by the method suggested by Filipovic et al. (1995 c) and by image analysis, respectively.

By using the method of Filipovic et al. the progression of the rewetting front can only be determined in eight positions where temperatures are measured. In the current study measured interior cooling curves were used to estimate the rewetting front since the surface measurement is not accurate. For the imaging method, the radius of the black zone in the photo of the jet impingement was measured as a function of time since the start of cooling.
Experimental results

Figure 5-4 Measured internal temperatures in test #5

Figure 5-5 Measured internal temperatures in test #10
Figure 5-6 shows the variation of the position of the rewetting front with time. The results obtained by the two methods matched each other very well except for the size of the initial wetted zone. The rewetting fronts obtained from measuring the cooling curves indicated that the plate surface at locations 1 and 2 was wetted immediately upon jet impingement. The initial rewetting front was located between location 2 and 3. The result obtained from image analysis indicates that the radius of the black region is closer to the fourth thermocouple location. The different results at location 3 indicated that film boiling is not visually observed in the dark region although it may have occurred. The slope of the curves represents the rate of progression of the rewetting front.

Figures 5-7 and 5-8 show the influences of the water temperature and jet flow rate on progression of the rewetting front respectively. The progressions of the rewetting front are all obtained by using cooling curves. Generally, with decreasing water temperatures and increasing jet flow rates the velocity of the rewetting front increased.
Experimental results

Figure 5-6 The rewetting front

Figure 5-7 Influence of the water temperature on progression of the rewetting front
5.4 Boiling curves

The boiling curve represents the variation of the heat flux with wall superheat (difference between surface temperature and water saturation temperature) or surface temperature. Figure 5-9 represents the boiling curves for six different locations in test #8 obtained using the inverse heat conduction model. As the trend of the boiling curve at location 2 is similar to that of location 1 and the boiling curves at locations 7 and 8 are similar to that of location 6, they are not shown here.

As shown in Figure 5-10, the film boiling regime determined by a decrease of the heat flux with decreasing the superheat is not observed at the stagnation point. The heat flux increases with the decrease of wall superheat to approximately 220 °C and, after passing the critical heat flux, starts decreasing until single-phase convection is reached. Similar trends were found in other tests under lower jet subcooling and higher jet flow rate.
conditions. The vapour layer formation is thought to be suppressed by jet momentum and high water subcooling.

Figure 5-9 Boiling curves at six locations in test #8 with water temperature and flow rate of 40 °C and 45 l/min respectively.

During the ascending stage (220 °C <ΔT_{sat}<750 °C) of the boiling curve, Li (2003) postulated that similar behaviour to nucleate boiling heat transfer occurred since the amount of bubbles increased with decreasing wall superheat, while Hernandez (1995,1999) speculated that the observed rise of the heat flux is characteristic for transition boiling. Viskanta (1992) assumed, however, that heat transfer beneath the jet and extending several jet widths was by single phase forced convection. Careful examination of the boiling curves revealed a shoulder near the point C (Figure 5-10) in the nucleate boiling regime. This phenomenon is reproducible under different test conditions and has been reported by other
Experimental results

Researchers (Ochi et al., 1984; Ishigai et al., 1978; Hall et al., 2001a). Hall et al. attributed it to the transition from nucleate boiling to single-phase convection. However, current results seem inconsistent with their conclusion: 1) The wall superheat at which it occurs may be much higher than the inception temperature for nucleate boiling, 2) There is no obvious change in slope of the boiling curve in the vicinity of this shoulder, and 3) The deflection point (point D in Figure 5-10) is consistently present in boiling curves at any locations and in all tests and its position and change of the slope of the boiling curve clearly indicate the occurrence of the transition from nucleate boiling to single phase convection.

Figure 5-10 Boiling curve at the stagnation point for water temperature and flow rate 40 °C and 45 l/min respectively
Away from the stagnation point, the subcooling decreases since the water near the plate was heated and the momentum of the jet decreased. The ability of the liquid to wet the surface decreases. The boiling curves for locations 4 and 5 clearly indicate five distinct regimes. These regimes are initial cooling, film boiling, transition boiling, nucleate boiling and single-phase convection as indicated in Figure 5-11. The initial cooling stage is typically not observed in steady-state cooling tests, which may be postulated to be nucleate boiling and transition boiling. Upon jet impingement, the bubbles generated on the hot surface and the amount of bubbles increased with decreasing the surface temperature since wall superheat was very high. When the heat flux approached point B, the bubbles started to coalesce and form vapour blankets, which resulted in the reduction of the area of liquid-
solid contact and thus the heat flux started to decrease. When wall superheat reached point C, a stable vapour layer was built and heat transfer mechanism changed to film boiling. Unlike the boiling curve under saturated pool boiling conditions, the heat flux in the film boiling regime increases slightly with decreasing wall superheat. The point C may correspond to a relative maximum vapour layer thickness and it decreases with decreasing wall superheat (Hall et al., 2001a). Under saturated film boiling conditions, the primary heat transfer mode is radiation and conduction through the vapour layer. Thus, the heat flux decreases with decreasing surface temperature. However, under subcooled, forced convection boiling condition, the heat transfer in the film boiling regime is primarily realized through heat convection (Filipovic et al., 1995a), which increases with decreasing vapour layer thickness and increasing local subcooling. The boiling curve at location 5 exhibits a similar trend to that at location 4. Since the local water temperature at location 5 is higher than that of location 4, it may extract less heat from the hot plate to evaporate the same amount of water to initiate film boiling. Therefore, film boiling at location 5 corresponds to a higher wall superheat. In addition, low water subcooling reduces the ability of the water to wet the surface. Thus, the rewetting temperature for location 5 is lower than that for location 4.

At location 3, the boiling curve exhibits a pronounced local minimum at wall superheat of approximately 420 °C and no stable film boiling is observed. This is probably due to the fact that the water subcooling and momentum, which are supposed to suppress stable film boiling, are higher at location 3 than at locations 4 and 5.

After initial cooling, the vapour film built at locations 4 and 5 formed a wedge that directed water away from the plate surface. The major heat transfer mechanism at locations
Experimental results

6, 7, and 8 are air-cooling until the rewetting front arrives at each location. The boiling curve at location 6 reveals surface reheating at a wall superheat of about 620 °C during the transition from initial cooling to air-cooling (in Figure 5-9). After the rewetting front reached the measuring location, the heat flux increased monotonically to the critical heat flux and then dropped in the nucleate boiling regime. The trends seen in test #8 are typical and observed in all the tests.

5.5 Effect of subcooling and flow rate

The effect of subcooling on the boiling curves at stagnation at a flow rate of 60 l/min is shown in Figure 5-12. It is found that higher subcooling increases the heat transfer. Similar trends are observed regardless of flow rate. Interestingly, the critical heat flux shifts to lower wall superheat with increasing subcooling. This result is contrary to what was found by other researchers (Ishigai et al. 1978, Kumagai et al. 1995b, Robidou et al. 2002), which can be explained by comparing the cumulative heat flux at the time corresponding to the critical heat flux. The cumulative heat flux reflects the overall energy extracted from unit area of the plate during a period of time. As illustrated in Figure 5-13, the cumulative heat flux at point 1 (at which the critical heat flux occurs) is equal to the shaded area on the small graph ($\int_{\text{crit}}^{\text{c}} q^* dt$). At the times demarked by points 1, 2, and 3, the critical heat flux for water temperatures of 20 °C, 40 °C, and 50 °C occurs, respectively. We can see that it takes less time to reach point 1 than point 2. And the cumulative heat flux for the former is somewhat larger than the later. This means that more energy was extracted from the plate within shorter time, which results in a larger temperature drop on plate surface. The
cumulative heat flux at point 2 is higher than that at point 3 and corresponds to a shorter time, indicating extraction of more energy and a lower surface temperature.

Figure 5-14 shows that the effect of cooling water temperature on the critical heat flux at all eight locations is more evident closer to the stagnation. Lower water temperature corresponds to higher critical heat flux. This effect decreases as the distance from stagnation increases.

Figure 5-12 Influence of subcooling on the boiling curves at stagnation
Experimental results

Figure 5-13 Cumulative heat flux

Figure 5-14 Influence of the water temperature on critical heat fluxes at all locations

1 - $T_{\text{water}} = 20 \, ^\circ\text{C}$
2 - $T_{\text{water}} = 40 \, ^\circ\text{C}$
3 - $T_{\text{water}} = 50 \, ^\circ\text{C}$

Flow rate = 60 l/min
As shown in Figure 5-15, similar to the effect of water temperature, an increase of jet velocity enhances heat transfer, particularly in the region near the critical heat flux. The temperature of critical heat flux also shifts to lower wall superheat with increasing jet flow rate. Figure 5-16 shows the influence of jet flow rate on critical heat flux at all eight locations, it is found that generally, the higher the jet flow rate, the higher the critical heat fluxes. This trend is more pronounced closer to the stagnation point.

Figure 5-15 Influence of jet flow rate on the boiling curves at stagnation
Experimental results

Figure 5-16 Influence of jet flow rate on critical heat fluxes at all locations

5.6 Comparison with top jet cooling

Data from the work of Hauksson (2001) for top jet cooling were employed for comparison with results obtained from the current study. Hauksson investigated boiling heat transfer during impingement of a single jet from a circular nozzle on an upward facing hot steel surface. Experimental conditions of top jet cooling were similar to those of the current work except that the top jet has much higher impingement velocity and thus higher momentum than the bottom jet.

In Figure 5-17, interior cooling curves at the stagnation point obtained under the condition of water temperature 40 °C and a jet flow rate of 30 l/min are compared. The initial temperature of the plate was about 850 °C and the distance from the bottom of the hole to the plate surface is 1.005 mm for bottom jet experiment and 1.052 mm for top jet
Experimental results

It is found that the temperature drops more rapidly during top jet cooling than for the bottom jet cooling, especially when the temperature is above approximately 700 °C. The fact that the cooling curve obtained by the top jet cooling has a larger slope during the initial 0.5 s indicates that the top jet cooling is more effective than the bottom jet cooling during this stage. Figure 5-18 shows the comparison of the boiling curves corresponding to the cooling curves in Figure 5-17. Top jet boiling curve is higher than bottom jet boiling curve at all heat transfer regimes.

Figure 5-17 Compare the bottom jet with top jet cooling curve at stagnation

Figure 5-19 shows the variation of the ratio of the cumulative heat flux during bottom jet cooling to that during top jet cooling, which reflects relative cooling efficiency of bottom to top jet. The curves for the two conditions exhibit four stages: rapid drop, rapid increase, gradual increase, and constancy. If looking at the curve for a water temperature 40
°C and a jet flow rate 30 l/min, we can see that the ratio decreases initially from approximately 35% to 21% within a rather short period (about 0.14 s) upon impingement due to the fact that the cumulative heat flux for the top jet cooling increases faster than for the bottom jet cooling during this period. Next, it takes about 1.38 second for the bottom jet to reach about 77% cooling efficiency of the top jet, at which relative cooling efficiency starts to increase gradually until it reaches 90% at a time of 4.87 s. In the last stage, the curve is flat and its slope is nearly zero. A similar trend was found in other tests. The data of the first stage may be more important when estimating cooling efficiency of an industrial runout table since the strip moves at a very high speed and any stagnation point will not stay within impingement zone for more than 0.01 s continuously (assuming a nozzle diameter of 20 mm and a strip speed of 10 m/s). At this stage the cooling efficiency of bottom jets is about 20% to 35% of top jets.

Figure 5-18 Compare bottom jet with top jet boiling curve at the stagnation point
5.7 Comparison of experimental data with model predictions

Three test results were chosen for comparison with relevant correlations. One test result was obtained under the lowest water temperature and the highest flow rate condition (test #10), which resulted in the highest heat flux; another was obtained under highest water temperature and lowest flow rate conditions (test #6), which resulted in the lowest heat flux; the third one has the highest flow rate and water temperature (test #12).

5.7.1 Nucleate boiling

Equation (2-19) is frequently used by researchers to correlate heat fluxes to wall superheat in the nucleate boiling region. Various values of C, and n have been reported in the literature. Five groups of data are listed in Table 5-2 and used to compare with the current data.
Table 5-2 Coefficients for nucleate boiling Equation (2-19)

<table>
<thead>
<tr>
<th>Authors</th>
<th>C</th>
<th>n</th>
</tr>
</thead>
<tbody>
<tr>
<td>Copeland (Wolf et al., 1993)</td>
<td>740</td>
<td>2.3</td>
</tr>
<tr>
<td>Ishigai et al. (Wolf et al., 1993)</td>
<td>42</td>
<td>3.2</td>
</tr>
<tr>
<td>Miyasaka et al. (Wolf et al., 1993)</td>
<td>79</td>
<td>3</td>
</tr>
<tr>
<td>Wolf et al. (1996)</td>
<td>63.7</td>
<td>2.95</td>
</tr>
<tr>
<td>Katto and Monde (Wolf et al., 1993)</td>
<td>450</td>
<td>2.7</td>
</tr>
</tbody>
</table>

In Figure 5-20, the correlations show significantly higher heat fluxes than the calculated heat fluxes. A similar trend was reported by Hall et al. (2001a) and Ishigai et al. (1978). All these correlations were established by using data obtained under steady-state experimental condition. This indicates that the heat flux obtained under steady-state cooling condition is larger than that obtained under transient cooling condition in the nucleate boiling regime, since during steady-state cooling the jet can extract as much heat from the sample as it can at certain wall superheat.

Equation (5-1) was obtained by making a modification to Equation (2-19), which shifted the boiling curves to the right by introducing an effective superheat which is 20 °C lower than the actual superheat. Three correlations were plotted in Figure 5-21 in comparison with the current data. It shows that new correlations fit current data better than the old.

\[ q_{NB} = a \cdot (\Delta T_{sat} - 20)^n \]  

(5-1)
Figure 5-20: Compare different nucleate boiling correlations with test data

Figure 5-21: Compare modified correlations with test data
5.7.2 Critical heat flux

Ishigai and Mizuno (1974) experimentally investigated the influences of water subcooling and suggested a correlation for describing the critical heat flux as expressed by Equation (2-15). The values for the parameters in Equation (2-15) under the current test conditions are given in Table 5-3.

Table 5-3 Values for parameters in Equation (2-15)

<table>
<thead>
<tr>
<th>Test</th>
<th>H (m)</th>
<th>d_n (m)</th>
<th>ΔT_{sub} (°C)</th>
<th>v_n (m/s)</th>
</tr>
</thead>
<tbody>
<tr>
<td>#6</td>
<td>0.088</td>
<td>0.019</td>
<td>50</td>
<td>1.76</td>
</tr>
<tr>
<td>#10</td>
<td>0.088</td>
<td>0.019</td>
<td>80</td>
<td>3.53</td>
</tr>
<tr>
<td>#12</td>
<td>0.088</td>
<td>0.019</td>
<td>50</td>
<td>3.53</td>
</tr>
</tbody>
</table>

The calculated critical heat fluxes by using Equation (2-15) are plotted in Figure 5-22. Ishigai's correlation is in good agreement with the data corresponding to subcooling of 50 °C, but it overestimates the critical heat flux by nearly 50% for a subcooling of 80 °C.

![Figure 5-22 Comparison between critical heat flux correlation and data](image-url)
Miyasaka et al. (1980) presented the CHF correlations at stagnation based on their experiments, which were described using Equations (2-16) through Equation (2-18).

Physical properties of saturated water and vapour are given in Table 5-4 and properties are taken at atmospheric pressure.

Table 5-4 Physical properties of saturated water and vapour (Rohsenow et al. 1998)

<table>
<thead>
<tr>
<th>$\rho_l$ (kg/m$^3$)</th>
<th>$\rho_g$ (kg/m$^3$)</th>
<th>$c_p$ (J/kg °C)</th>
<th>$h_i$ (kJ/kg)</th>
<th>$\sigma$ (N/m)</th>
</tr>
</thead>
<tbody>
<tr>
<td>958.4</td>
<td>0.59</td>
<td>4216</td>
<td>2257.92</td>
<td>0.059</td>
</tr>
</tbody>
</table>

The calculated critical heat flux according to Equations (2-16) through (2-18) and the measured are given in Table 5-5. It is found that the calculated heat fluxes using Miyasaka’s correlation are much higher than those measured in the current test.

Table 5-5 Calculated critical heat fluxes according to Miyasaka’s correlation

<table>
<thead>
<tr>
<th>Test</th>
<th>$\Delta T_{sub}$</th>
<th>$v_n$ (m/s)</th>
<th>$q_{CHF}^{*}$ (MW/m$^2$) (Calculated)</th>
<th>$q_{CHF}^{*}$ (MW/m$^2$) (Measured)</th>
</tr>
</thead>
<tbody>
<tr>
<td>#6</td>
<td>50</td>
<td>1.76</td>
<td>9.9</td>
<td>5.14</td>
</tr>
<tr>
<td>#10</td>
<td>80</td>
<td>3.53</td>
<td>18.5</td>
<td>8.56</td>
</tr>
<tr>
<td>#12</td>
<td>50</td>
<td>3.53</td>
<td>12.2</td>
<td>7.41</td>
</tr>
</tbody>
</table>

5.7.3 Initial cooling

As discussed in the previous section (see Figure 5-10), heat transfer behaviour during the initial cooling stage at the stagnation point was postulated to be similar to nucleate boiling by the author, although the trend of the boiling curve is similar to transition boiling. In this section, the transition boiling correlations are used to compare with those measured in the current test.
Berenson et al. (1962) assumed that transition boiling was a combination of unstable nucleate boiling and unstable film boiling, and suggested a general form for calculating transition boiling, which is given by Equation (2-8). Equations (2-12) and (2-13) were obtained by Kandlikar et al. (1999) using pool boiling data from the literature for \( F \) and \( q^{*} \). The minimum heat flux (MHF) was estimated according to a model developed by Ochi et al. (1984) as given by Equation (2-7).

Figure 5-23 shows the comparison between experimental data and the correlation for transition pool boiling by Kandlikar et al. The calculated minimum heat fluxes are 0.54MW/m\(^2\) and 2.3MW/m\(^2\) respectively corresponding to conditions of test #6 and #10. It can be seen that the predicted transition boiling has a different trend from the experimental data. The experimental data exhibit a convex shape, while the correlation shows a concave shape.

After comparing the boiling curves in the initial cooling stage obtained at stagnation under different test conditions, it is found that they can be described by a second order polynomial function. The second order polynomial function has the following general form:

\[
q^{*} = a(750 - \Delta T_{\text{sat}})^2 + b(750 - \Delta T_{\text{sat}}) + c
\]  

(5-5)

where \( a \), \( b \), and \( c \) are parameters to be obtained by fitting the data. The value of 750 denotes the initial wall superheat (in °C). The values for \( a \), \( b \), and \( c \) for tests #6, #10, and #12 are listed in Table 5-6. The coefficient of determination (\( R^2 \)) for every fitting was close to 1, which indicates that Equation (5-11) fits the experimental data very well. For other tests, where test conditions are in between those of tests #6, #10, and #12, their boiling curves in the initial cooling stage can be obtained by linear interpolation using the data from neighbouring tests. For example, in Figures 5-24, boiling curves at the initial cooling stage...
Experimental results

(dashed line 2) for test #11 under water subcooling of 60 °C and jet flow rate of 60 l/min can be obtained using data from test #10 and #12, which have the same jet flow rate but different water subcooling of 80 °C and 50 °C. In Figure 5-25, transition boiling curve (dotted-dashed line 2), which corresponds to the test condition with a water subcooling of 50 °C and jet flow rate of 45 l/min, is obtained by interpolating data for flow rates of 30 l/min and 60 l/min. The interpolated heat flux shows a very good match to the experimental data. The maximum relative difference is within 5% except at the beginning when absolute heat fluxes are very small.

Table 5-6 Parameters for Equation (5-5)

<table>
<thead>
<tr>
<th>Test</th>
<th>a</th>
<th>b</th>
<th>c</th>
<th>R²</th>
</tr>
</thead>
<tbody>
<tr>
<td>#6</td>
<td>-27.434</td>
<td>22943</td>
<td>87188</td>
<td>0.9997</td>
</tr>
<tr>
<td>#12</td>
<td>-17.683</td>
<td>23381</td>
<td>84897</td>
<td>0.9998</td>
</tr>
<tr>
<td>#10</td>
<td>-18.374</td>
<td>25351</td>
<td>65957</td>
<td>0.9998</td>
</tr>
</tbody>
</table>

Figure 5-23 Compare with transition boiling correlation
Experimental results

Figure 5-24 Interpolation of the heat flux between different water temperatures

Figure 5-25 Interpolation of the heat flux between different flow rates
6. Summary and conclusions

6.1 Summary of the work

An experimental method was developed for studying boiling heat transfer during impingement of water jets on a downward facing hot steel surface. Temperature measurements were made at eight separate locations, each of which included a pair of intrinsic thermocouples with one measuring the bottom surface temperatures and the other measuring the internal temperatures at about 1 mm above the bottom surface. Measured internal temperatures were employed by the inverse heat conduction model. The cooling process for tests with flow rates lower than 45 l/min was filmed and the video signal was converted to a digital image for data analysis.

An inverse heat conduction model considering the progression of the rewetting front has been developed and verified. The model used a two dimensional, axisymmetric finite element method to solve the nonlinear transient heat conduction problem, and a function specification method for unknown surface heat fluxes. The zeroth-order Tikhonov regularization method was an effective method to stabilize the solution. The sequential-in-time solution procedure enabled the use of constant material thermal properties in the calculation of temperatures and sensitivity coefficients and thus eliminating reformulation of finite element conductance and capacitance matrices within a small time interval and reducing the computations significantly.

Both the effect of water temperature in the range of 20 °C to 50 °C and jet flow rate in the range of 30 l/min to 60 l/min on heat transfer were investigated. The existing
correlations for describing different boiling heat transfer regimes were compared with current data and new correlations were suggested.

6.2 Conclusions

Visual observations indicated that the entire cooling process consisted of three phases. In the first phase, which occurred upon jet impingement and within the next few seconds, a black zone right above the jet was observed and water covered the entire surface until a stable vapour layer was established (the second phase). In the second phase the water covers a certain ring somewhat larger that the black zone itself and then is diverted from the plate in an arc without getting in contact with the rest of the plate. Shortly after the start of the second phase the black zone progressed radially outwards until the whole plate surface turned black (the third phase).

Based on the visual observation of the tests and analysis of the cooling curves, three different cooling zones have been observed on a stationary plate. In the zone right above the jet including the stagnation point and the area in its vicinity, the heat transfer is single phase convection and nucleate boiling primarily. Outside zone 1, the vapour film is built and transition and film boiling is predominant. Strong boiling behaviour around the boundary zone 2 deflected the water film off the plate and left the rest of plate surface untouched by the water and the primary heat transfer is air-cooling, which is defined as zone 3. The boundaries of each zone vary with water temperature and flow rate and also with time.

Boiling curves exhibit different heat transfer regimes depending on the locations. At the stagnation point, the primary heat transfer mechanisms are nucleate and transition boiling. At locations outside the impingement zone film boiling or air-cooling were
predominant right after the initial cooling stage and transition and nucleate boiling heat transfer dominates later stages of the boiling curve.

Generally, with increasing subcooling the heat flux at the same wall superheat increases and the critical heat flux shifts to lower wall superheat at the stagnation point. This effect decreases as the distance from stagnation increases. Cooling water temperature also has an influence on the rate of progression of the rewetting front. The speed of the rewetting front increases with decreasing water temperature. Increasing jet flow rate has a similar effect to decreasing cooling water temperature except that it has no obvious influence on nucleate boiling.

Current investigation indicates considerable differences between various existing models for boiling heat transfer since their application is restricted to specific conditions. An empirical correlation, which is the second order polynomial function of the difference of initial wall superheat and transient wall superheat, was found to be suitable for describing boiling curves during the initial cooling stage at the stagnation point. The heat flux at conditions within its boundary test conditions can be calculated by interpolation of heat fluxes at boundary test conditions. By using this method, the model developed correlated the current data with a maximum deviation of less than 5%.
7. Recommendations for future work

Current research shows that heat transfer behaviour in jet impingement cooling is very complicated and further work is required to improve our knowledge of heat transfer on the industrial runout table.

- In the aspect of experiment, moving plate tests are suggested since it enables a number of new factors such as speed of plate, distances between jet lines to be considered, and is closer to industrial conditions. During the moving plate test, it is suggested to use slot type nozzle or multiple circular nozzles placed along the width of a plate to reduce temperature gradients in this direction.

- Under current test conditions temperature measurements made at the surface opposite to the cooled surface are valid for estimating the heat flux at the surface. This has been validated by the analysis in Appendix C. Thus, thermocouples can be placed at the back surface instead of at the bottom of the hole. By this way, both preparation of the plate and instrumentation of thermocouples are simplified, and the influence of the hole for an internal thermocouple on the local temperature field can be eliminated.

- The influence of other parameters such as nozzle to plate distance and initial plate temperature on various boiling heat transfer regimes should be investigated.

- In the aspect of numerical modelling, heat generation due to phase transformation should be incorporated into the current inverse heat conduction model to improve its
accuracy. Moreover, it is suggested to consider the hydrodynamics of the rewetting front from a phenomenological point of view in the model.

- It is suggested to develop an inverse heat conduction model using a gradient method, which requires solution of three equations: direct, adjoint and sensitivity for temperature, search direction and step size in the search direction on the whole time domain. To the best knowledge of the author, there is no report on the application of this method to the jet impingement cooling problem. It is comparable to the function specification method in terms of computational time (Dowding, 1999) and another advantage is that this method does not require assumption of a spatial distribution of heat fluxes.
References


References


Ishigai, S. and Mizuno, M., 1974, Boiling heat transfer with an impingement water jet (about the critical heat flux), Preprint of JSME, No.740-16, 139-142.


Li, D., 2003, Boiling water heat transfer during quenching of steel plates and tubes, Ph. D Thesis, MMAT, UBC.


References


Appendix A. Finite element formulation of the heat conduction problem

A.1 Problem statement

The governing equation for the general heat conduction problem in an isotropic solid with temperature-dependent thermal properties can be expressed by

\[ \frac{\partial}{\partial x} (k \frac{\partial T}{\partial x}) + \frac{\partial}{\partial y} (k \frac{\partial T}{\partial y}) + \frac{\partial}{\partial z} (k \frac{\partial T}{\partial z}) + Q = \rho C_p \frac{dT}{dt} \]  \hspace{1cm} (A-1)

where \( k \) is heat conductivity, \( \rho \) the density, \( c_p \) the specific heat of the material, \( Q \) internal heat generation rate per unit volume, \( T \) the temperature, and \( t \) is the time.

The heat conduction equation is solved subject to an initial condition specifying the temperature distribution throughout the solid at time zero and boundary conditions on all boundaries of the solid. The initial condition is expressed by

\[ T(x, y, z, 0) = T_0(x, y, z) \quad \text{in} \quad \Omega, \quad t=0 \]  \hspace{1cm} (A-2)

The boundary conditions are

1) A prescribed temperature distribution

\[ T = T(x, y, z, t) \quad \text{on surface} \ S_T \]  \hspace{1cm} (A-3)

2) A prescribed heat flux \( q(x,y,z,t) \) (positive into the surface) for the heat flow boundary condition

\[ [k \frac{\partial T}{\partial x} n_x + k \frac{\partial T}{\partial y} n_y + k \frac{\partial T}{\partial z} n_z] = q \quad \text{on surface} \ S_q \]  \hspace{1cm} (A-4)

3) A prescribed convective film coefficient \( h(x,y,z,t) \)

\[ -[k \frac{\partial T}{\partial x} n_x + k \frac{\partial T}{\partial y} n_y + k \frac{\partial T}{\partial z} n_z] = h(T - T_\infty) \quad \text{on surface} \ S_C \]  \hspace{1cm} (A-5)

where \( T_\infty \) is the ambient temperature.
Adiabatic boundary condition is a special heat flow boundary condition when the heat flux is zero. Radiation boundary conditions can be treated as a given heat flux boundary as expressed by Equation (A-6) or a given convective coefficient as expressed by Equation (A-7)

\[
q_r = \sigma \varepsilon (T^4 - T_m^4) \tag{A-6}
\]

\[
h = \sigma \varepsilon (T^2 + T_m^2)(T + T_m) \tag{A-7}
\]

where, \( \sigma_{SB} \) is Stefan-Boltzmann constant, taken as \( 5.67 \times 10^{-8} \) \( \text{W/m}^2\text{(°C)}^4 \), and \( \varepsilon \) is the surface emissivity.

A.2 Finite element formulation

The derivation of the finite element formulation for the general heat conduction problem is obtained by referring to Huebner et al. (2001). After discretization of the domain into a finite number of elements, the temperature \( T \) in each element is approximated by its nodal temperatures \( T_i \) via the interpolation functions \( N_i(x,y,z) \)

\[
T^e = \sum_{i=1}^{n} N_i(x,y,z) T_i \tag{A-8}
\]

where \( N_i \) are the shape functions, \( n \) the number of nodes in an element, and superscript \( e \) denotes element.

A weighted residual expression for Equation (A-1) on an element level gives

\[
\int_V W_i \left( \frac{\partial}{\partial x} (k \frac{\partial T}{\partial x}) + \frac{\partial}{\partial y} (k \frac{\partial T}{\partial y}) + \frac{\partial}{\partial z} (k \frac{\partial T}{\partial z}) + \rho C_p \frac{dT}{dt} \right) dx dy dz = 0 \tag{A-9}
\]

where \( W_i \) are the weight functions and \( W_i = N_i \) for Galerkin’s method (Huebner et al., 2001).

Applying Gauss’s theorem to Equation (A-9) yields to
Appendix A

\[ \int_{V}[k \frac{\partial N_i}{\partial x} \frac{\partial T^e}{\partial x} + k \frac{\partial N_i}{\partial y} \frac{\partial T^e}{\partial y} + k \frac{\partial N_i}{\partial z} \frac{\partial T^e}{\partial z}] dxdydz - \int_{V} N_i Q dxdydz - \int_{V'} N_i Q dxdydz = 0 \] (A-10)

Utilizing the definition of \( T^e \) (Equation (A-8)), Equation (A-10) can be further modified to

\[ \int_{V}[k \frac{\partial N_i}{\partial x} \frac{\partial T^e}{\partial x} + k \frac{\partial N_i}{\partial y} \frac{\partial T^e}{\partial y} + k \frac{\partial N_i}{\partial z} \frac{\partial T^e}{\partial z}] dxdydz \{ T^e \} - \int_{V} N_i Q dxdydz - \int_{V'} N_i Q dxdydz = 0 \] (A-11)

where \( \{ T \}^e \) is the column vector of nodal unknown for the element. Boundary \( S^e \) is composed of the boundaries \( S^e_f, S^e_c, S^e_, \) which are given by Equations (A-3), (A-4) and (A-5), respectively. Substituting Equations (A-4) and (A-5) into Equation (A-11) gives

\[ \int_{V} \rho C_p N_i N_j dQ \frac{d\{ T \}^e}{dt} + \int_{V} [k \frac{\partial N_i}{\partial x} \frac{\partial N_j}{\partial x} + k \frac{\partial N_i}{\partial y} \frac{\partial N_j}{\partial y} + k \frac{\partial N_i}{\partial z} \frac{\partial N_j}{\partial z}] dxdydz \{ T \}^e - \int_{V} N_i Q dxdydz - \int_{S^e_f} q N_i ds + \int_{S^e_c} h N_i N_j ds \{ T \}^e - \int_{S^e_} h T_\infty N_i ds = 0 \] (A-12)

In matrix form, Equation (12) becomes

\[ [C]^e \frac{d\{ T \}^e}{dt} + ([K_c]^e + [K_h]^e) \{ T \}^e = \{ F \}^e \] (A-13)

where \([C]^e\) is the element capacitance matrix, \([K_c]^e\) and \([K_h]^e\) are element conductance matrices related to conduction and convection, respectively. The expressions are given as follows:

\[ [C]^e = \int_{V} \rho C_p N_i N_j dxdydz \] (A-14)

\[ [K_c]^e = \int_{V} [k \frac{\partial N_i}{\partial x} \frac{\partial N_j}{\partial x} + k \frac{\partial N_i}{\partial y} \frac{\partial N_j}{\partial y} + k \frac{\partial N_i}{\partial z} \frac{\partial N_j}{\partial z}] dxdydz \] (A-15)

\[ [K_h]^e = \int_{S^e} h N_i N_j ds \] (A-16)
The vector \( \{F\}_e \) include following three heat load vectors accounting for the contribution from internal heat generation \( \{F_Q\}_e \), surface convection \( \{F_h\}_e \), and specified surface heat flow \( \{F_q\}_e \), respectively

\[
\{F\}_e = \{F\}_Q + \{F\}_h + \{F\}_q
\]  
(A-17)

\[
\{F\}_Q = \int_{\Gamma} N_i Q dxdydz
\]  
(A-18)

\[
\{F\}_h = \int_{\Gamma} hT_a N_i ds
\]  
(A-19)

\[
\{F\}_q = \int_{\Gamma} qN_i ds
\]  
(A-20)

The matrices in Equation (A-13) are normally evaluated by numerical integration. A popular method of numerical integration is the Gauss-Legendre quadrature (Huebner et al. 2001). Using this method for the two-dimensional case, the element conductance matrix can be expressed by

\[
[K_e] = \int_1^{M} \int_1^{M} [B]^T [k] [B] \det J dudv = \sum_{i=1}^{M} \sum_{j=1}^{M} W_i W_j [B]^T [k] [B] \det J
\]  
(A-21)

where \( W_i \) and \( W_j \) represent Gauss weight factors, \( M \) is the number of integration points. For 2-point or \( 2 \times 2 \) integration, \( M=2 \). \( [B] \) is the matrix of differential operators and \( [B]^T \) is the transpose of \( [B] \). They are defined as follows

\[
[B] = [B(x,y)] = \begin{bmatrix}
\frac{\partial N_1}{\partial x} & \frac{\partial N_2}{\partial x} & \ldots & \frac{\partial N_n}{\partial x} \\
\frac{\partial N_1}{\partial y} & \frac{\partial N_2}{\partial y} & \ldots & \frac{\partial N_n}{\partial y}
\end{bmatrix}
\]  
(A-22)

\[
[k] = \begin{bmatrix}
k & 0 \\
0 & k
\end{bmatrix}
\]  
(A-23)
where \( n \) is the number of nodes. The derivatives of shape functions with respect to \( x \) and \( y \) are calculated by

\[
\begin{bmatrix}
\frac{\partial N_i}{\partial x} \\
\frac{\partial N_i}{\partial y}
\end{bmatrix} = [J]^{-1} \begin{bmatrix}
\frac{\partial N_i}{\partial u} \\
\frac{\partial N_i}{\partial v}
\end{bmatrix}
\quad (A-24)
\]

where \([J]^{-1}\) is the inverse matrix of the Jacobian matrix \([J]\), which can be calculated by

\[
[J]^{-1} = \begin{bmatrix}
\frac{J(2,2)}{\det J} & -\frac{J(1,2)}{\det J} \\
-\frac{J(2,1)}{\det J} & \frac{J(1,1)}{\det J}
\end{bmatrix}
\quad (A-25)
\]

and

\[
[J] = \begin{bmatrix}
\frac{\partial x}{\partial u} & \frac{\partial x}{\partial v} \\
\frac{\partial y}{\partial u} & \frac{\partial y}{\partial v}
\end{bmatrix} = \begin{bmatrix}
J(1,1) & J(1,2) \\
J(2,1) & J(2,2)
\end{bmatrix} = \det \begin{bmatrix}
\sum_{i=1}^{n} \frac{\partial N_i}{\partial x} x_i & \sum_{i=1}^{n} \frac{\partial N_i}{\partial y} y_i \\
\sum_{i=1}^{n} \frac{\partial N_i}{\partial u} x_i & \sum_{i=1}^{n} \frac{\partial N_i}{\partial v} y_i
\end{bmatrix}
\quad (A-26)
\]

\[
det J = J(1,1) \cdot J(2,2) - J(1,2) \cdot J(2,1)
\]

The element capacitance matrix can be evaluated by

\[
[C]^e = \int \rho C_p N_i N_j dxdydz = \sum_{i=1}^{M} \sum_{j=1}^{M} W_i W_j \{N\} \rho C_p [N] \det J
\quad (A-27)
\]

The element load vector for a specific heat flux (\( q \) is positive into the surface) can be expressed by Equation (A-20). In a local coordinate system, the integral along the interested face changes to

\[
\{F_q\}_e = \int_{\gamma} q N_i ds = \int_{-1}^{1} q N_i \sqrt{J_{11}^2 + J_{12}^2} du \quad \text{or}
\]

\[
\{F_q\}_e = \int_{\gamma} q N_i ds = \int_{-1}^{1} q N_i \sqrt{J_{21}^2 + J_{22}^2} dv
\quad (A-28)
\]

where \( J_{11}, J_{12}, J_{21}, \) and \( J_{22} \) are elements of Jacobian matrix defined in Equation (A-26).

By applying the Gaussian quadrature integration to above equations, we have
\[ \{F_q\} = \sum_{i=1}^{M} W_i \{N\} \cdot q \sqrt{J_{11}^2 + J_{12}^2} \quad \text{for side 1,3} \quad (A-29) \]

\[ \{F_q\} = \sum_{i=1}^{M} W_i \{N\} \cdot q \sqrt{J_{21}^2 + J_{22}^2} \quad \text{for side 2,4} \]

Other element matrices and load vectors can be handled in the same way as what have been described above and thus are not given.

Equation (A-13) is a general nonlinear formulation of element equations for the transient heat conduction problem. The system equations are obtained by assembling the element equations following the standard procedure, which was introduced elsewhere (Reddy et al., 1992; Zienkiewicz and Tayler, 2000).

The governing equation for the two-dimensional axisymmetric heat conduction problem can be expressed by

\[ \frac{1}{r} \frac{\partial}{\partial r} (kr \frac{\partial T}{\partial r}) + \frac{\partial}{\partial z} (k \frac{\partial T}{\partial z}) + Q = \rho C_p \frac{\partial T}{\partial t} \quad (A-1') \]

and has similar boundary conditions to Equations (A-3)-(A-5) except that they are expressed in the r-z cylindrical coordinate system. The derivation of the finite element formulation for the two-dimensional axisymmetric heat conduction problem is not given here since it is similar to two-dimensional heat conduction problem. The following matrices and load vectors used for axisymmetric heat conduction problem are indicated in comparison to equations for the two-dimensional problem by adding the symbol "'".

The element conductance matrix is expressed by

\[ [K_e]' = \sum_{i=1}^{M} \sum_{j=1}^{M} W_i W_j [B]^T [k] [B] \det J \cdot \sum_{k=1}^{4} N_k (u_i, v_j) \cdot r_k \quad (A-21') \]

The element capacitance matrix can be expressed by
\[ [C]^r = \sum_{i=1}^{M} \sum_{j=1}^{M} W_i W_j \{N\} \rho C_p \{N\} \det J \cdot \sum_{k=1}^{4} N_k(u_i, v_j) \cdot r_k \quad \text{(A-27')} \]

and

\[ \{F_q\}_{1} = \sum_{i=1}^{M} W_i \{N\} \cdot q \sqrt{J_{11}^2 + J_{12}^2} \cdot \sum_{k=1}^{4} N_k(u_i, v_j) \cdot r_k \quad \text{for side 1,3} \quad \text{(A-29')} \]

\[ \{F_q\}_{2} = \sum_{i=1}^{M} W_i \{N\} \cdot q \sqrt{J_{21}^2 + J_{22}^2} \cdot \sum_{k=1}^{4} N_k(u_i, v_j) \cdot r_k \quad \text{for side 2,4} \]

\section*{A.3 Solution}

After numerical integration, according to Huebner (2001), Equation (A-13) can be further written as

\[ ([C(T)] \cdot \{\dot{T}(t)\} + [K(T, t)] \cdot \{T(t)\} = \{F(T, t)\} \quad \text{(A-30)} \]

A general family of time march schemes results by introducing a parameter \( \theta \) such that \( t_0 = t_n + \theta \Delta t \), where \( 0 \leq \theta \leq 1 \). Equation (A-30) can be re-written as

\[ ([C(T_\theta)] \cdot \{\dot{T}_\theta\} + [K(T_\theta, t_\theta)] \cdot \{T_\theta\} = \{F(T_\theta, t_\theta)\} \quad \text{(A-31)} \]

where the subscript \( \theta \) indicates the temperature vector \( \{T(t_\theta)\} \) at time \( t_\theta \) and introduces the approximations

\[ \{\dot{T}_\theta\} = \frac{\{T\}_{n+1} - \{T\}_n}{\Delta t} \quad \text{(A-32)} \]

\[ \{T_\theta\} = (1 - \theta)\{T\}_n + \theta\{T\}_{n+1} \quad \text{(A-33)} \]

Substituting the expressions for \( \{\dot{T}_\theta\} \) and \( \{T_\theta\} \) into Equation (A-31) yields

\[ [A] \cdot \{T\}_{n+1} = \{B\} \quad \text{(A-34)} \]

where
\[ [A] = \theta \cdot [K(T_\theta, t_\theta)] + \left[ \frac{C(T_\theta)}{\Delta t} \right] \]

\{B\} = \left[ -\left( 1 - \theta \right)K(T_\theta, t_\theta) + \left[ \frac{C(T_\theta)}{\Delta t} \right] \right] \{T\}_n + \{F(T_\theta, t_\theta)\}

\{F(T_\theta, t_\theta)\} = \theta \{F\}_{n+1} + (1 - \theta) \{F\}_n

and \{T\}_{n+1} and \{T\}_\theta are unknowns, \{T\}_n is known from the previous time step. If \( n=0 \), it is the initial temperature. When \( \theta=1/2 \), the above time marching scheme is the Crank-Nicolson time stepping technique (Huebner et al. 2001).

The system of equations (Equation (A-34)) can be solved using either linear solver with skyline compacted storage techniques or by nonlinear solver such as Newton-Raphson method and then by linear solution method.

**A.3.1 Solution of linear system equations**

In the current work, a variant of the Choleski factorization solver was realized with skyline storage. The algorithm is as follows (Zienkiewicz and Tayler, 2000):

Assuming a linear system of equations can be expressed as

\[ A \cdot X = B \]  \hspace{1cm} (A-35)

If a square matrix \( A \) is symmetric and positive definite, then we can construct a lower triangular matrix \( L \) whose transpose \( L^T \) can itself serve as the upper triangular part and a diagonal matrix \( D \) to replace equation \( A \)

\[ A = LDL^T \]  \hspace{1cm} (A-36)

Supposed that the first non-zero component of the \( i_{th} \) row in \( A \) is \( a_{i,m_i} \), and taking

\[ m_j = \text{Max}(m_i, m_j) \]  \hspace{1cm} (A-37)

the components in matrices \( L \) and \( D \) can be expressed as
Appendix A

\[ l_{ij} = (a_{ij} - \sum_{k=m_j}^{i-1} l_{ik} d_{kk} l_{jk} / d_{jj}) \quad i > j, \quad j = m_i, m_i+1, \ldots, i-1 \] (A-38)

\[ d_{ii} = a_{ii} - \sum_{k=m_i}^{i-1} l_{ik}^2 d_{kk} \quad i = 1, 2, \ldots, n \] (A-39)

The components in B can also be decomposed in the following way:

\[ B = LDL^T \] (A-40)

and the component in \( \hat{B} \) can be obtained by

\[ \hat{b}_i = (b_i - \sum_{j=m_i}^{i-1} l_{ij} \hat{d}_{jj} / d_{ii}) \quad i = 1, 2, \ldots, n \] (A-41)

Substituting Equations (A-40) and (A-36) into Equation (A-35) and we have

\[ L^T \cdot X = \hat{B} \] (A-42)

Thus solving Equation (A-35) is equivalent to solving Equation (A-42).

Making the following recurrence

\[ \hat{b}_j - l_{ij} \hat{b}_i \Rightarrow \hat{b}_j \quad i = n, n-1, \ldots, 2, \quad j = m_i, m_i+1, \ldots, i-1 \] (A-43)

then we get

\[ x_i = \hat{b}_i \quad i = n, n-1, \ldots, 1 \] (A-44)

Because matrix \([A]\) is symmetric, it is sufficient to store its components \((A[i,j])\) between the first non-zero component and the diagonal component of the lower triangle part of the matrix into a new one dimensional array \(A[k]\). The relationship between \(A[i,j]\) and \(A[k]\) and the position of the first non-zero component in each row of \(A[i,j]\), \(m_i\), can be determined by


\[ m_i = i - (AD[i] - AD[i-1]) + 1 \] (A-45)
where matrix $AD[i]$ records the position of each diagonal component in $A[k]$.

**A.3.2 Solution of a nonlinear system equations**

Here, the Newton-Raphson method is adopted because this method has good convergence property for most non-linear problems (Zienkiewicz and Taylor, 2000).

In Equation (A-34), if $[A]$ or/and $\{B\}$ are functions of temperature, the problem becomes non-linear, which is solved by the Newton-Raphson iteration method. The formulation used for iteration includes:

$$[J] \cdot \{\Delta T^k\} = -\{F\} \quad (A-46)$$

Neglecting the contributions from temperature-dependent heat vectors and an increment in matrix $\Delta K$ (Huebner, 2001), $[J]$ is approximated by the conductance matrix $[A]$ in Equation (A-34). Unbalance in nodal heat loads $\{F\}$ is

$$\{F\} = [A] \cdot \{T\}_{n+1} - \{B\}$$

The iteration is performed according to Equation (A-47) until $\Delta T^k$ satisfies the convergence criterion, which is that the sum of the squares of the deviation in the solution is smaller than a given tolerance ($1e-6$).

$$T^{k+1} = T^k + \Delta T^k \quad (A-47)$$
Appendix B. Formulation for inverse heat conduction problem

The present inverse heat conduction problem can be described mathematically as follows: Given measured temperatures $Y_i$ (i =1, 2, ..., M) at M discrete points, the heat flux profile given by its components $q_i$ (i =1, 2, ..., M) is to be estimated such that the calculated temperatures $T_i$ (i =1, 2, ..., M) match the measured temperatures as closely as possible. The matching can be implemented by minimizing the standard least squares norm with respect to each of the unknown heat flux components at the bottom surface of the plate.

In order to overcome the time lag of interior measurements, future temperature information was used (Osman, 1989). Suppose that the current time step is $P-1$, the heat fluxes are assumed to be constant over $R$ future time steps, i.e.

$$q^P = q^{P+1} = \ldots = q^{P+R-1}$$  \hfill (B-1)

Tikhonov zeroth-order regularization was introduced to the standard least squares technique to improve the stability of the solution (Liu, 2001). The zeroth-order regularization reduces the maximum magnitudes of estimated values of $q_i$. The sum of squares function modified by the addition of the zeroth-order regularization function can be expressed as:

$$S = \sum_{i=1}^{R} \sum_{j=1}^{M} (Y_{i}^{P+j-1} - T_{i}^{P+j-1})^2 + \alpha \sum_{j=1}^{M} (q_{i} - q_{i}^0)^2$$  \hfill (B-2)

where $S$ is the sum of squares, $q_{i}^0$ is the given heat flux, $\alpha$ is the regularization parameter.

When the value of $\alpha$ tends to zero, the solution exhibits oscillatory behaviour and can become unstable. On the other hand, when the value of $\alpha$ tends to be a large value, the
solution is overdamped and deviates from the exact results. By proper selecting the value of α, the fluctuations of the solution can be alleviated.

Minimizing Equation (B-2) by differentiating it with respect to each of the unknown heat flux components and then setting the resulting expression equal to zero gives:

\[
\frac{\partial S}{\partial q_j} = \sum_{i=1}^{M} \sum_{k=1}^{M} (Y_{i_k}^+, T_{i_k}^+) \frac{\partial T_i}{\partial q_j} - \alpha \sum_{i=1}^{M} (q_i - q_i^0) \frac{\partial q_i}{\partial q_j} = 0 \quad (B-3)
\]

That is

\[
\sum_{i=1}^{M} (Y_{i_k}^+, T_{i_k}^+) \frac{\partial T_i}{\partial q_j} = \alpha \sum_{i=1}^{M} (q_i - q_i^0) \frac{\partial q_i}{\partial q_j} \quad (B-4)
\]

where \(i=1, 2, ..., M\). In Equation (B-4), the first derivative of the dependent variable \(T_i\) with respect to the unknown parameter \(q_j\) is termed the sensitivity coefficient \(X_{ij}\), which can be written as:

\[
X_{ij} = \frac{\partial T_i}{\partial q_j} \quad (B-5)
\]

and

\[
\frac{\partial q_i}{\partial q_j} = \delta_{ij} \quad (\delta_{ij}=0 \text{ for } i \neq j; \delta_{ij}=1 \text{ for } i=j) \quad (B-6)
\]

The sensitivity coefficient \(x_{ij}\) was obtained by solving sensitivity equations, which were derived by differentiating both sides of the temperature equations (Equation (A-34)) with respect to the unknown heat fluxes \(q_j\) assuming matrices \(A\), and load \(F\) are temperature independent during this small time interval (Osman, 1997) and we get:

\[
[A] \cdot \{\frac{\partial T}{\partial q_j}\}_{n+1} = \{Q\} \quad (B-7)
\]

where
$[A] = \theta \cdot [K(T_\theta, t_\theta)] + \frac{[C(T_\theta)]}{\Delta t}$

$\{Q(T_\theta, t_\theta)\} = \theta \cdot \{\frac{\partial F}{\partial q_j}\}_{n+1} = \theta \cdot \int_{\Sigma_j} N_i \cdot ds$

Due to the similarities of the equations for the sensitivity problem and the direct problem, the conductance matrix, capacitance matrix and the solver used in the direct problem can be applied to the sensitivity problem with the modification of the initial and boundary conditions.

To solve the system of least squares equations (Equation (B-2)), an expression of $T_i$ as a function of $q_j$ is needed. Given a heat flux distribution $q^0_j$ the corresponding temperatures, $T^0_i$, are obtained by solving the direct problem. Expanding the temperature field $T_i$ in the first order Taylor series in terms of the given heat flux $q^0_j$ gives (Liu, 2001)

$T_i = T^0_i + \sum_{j=1}^{M} \frac{\partial T_i}{\partial q_j} (q_j - q^0_j)$  \hspace{1cm} (B-8)

and substituting Equation (B-8) into Equation (B-4), we get,

$\sum_{i=1}^{R} \left\{ \sum_{j=1}^{M} \left[ Y_i - T^0_i - \sum_{k=1}^{M} \frac{\partial T_j}{\partial q_k} (q_j - q^0_j) T^0_k \right] \cdot \frac{\partial T_i}{\partial q_j} \right\}^{p+1} = \alpha \sum_{i=1}^{M} (q_i - q^0_i) \delta_{ij}$  \hspace{1cm} (B-9)

after rearrangement, we have:

$\left[ \sum_{i=1}^{R} \sum_{j=1}^{M} \left( \frac{\partial T_i}{\partial q_j} \cdot \frac{\partial T_j}{\partial q_k} \right)^{p+1} + \alpha \cdot I \right] \cdot \Delta q_k = \sum_{i=1}^{R} \sum_{j=1}^{M} [(Y_i - T^0_i) \cdot \frac{\partial T_i}{\partial q_j}]^{p+1}$  \hspace{1cm} (B-10)

where the regularization matrix $\alpha I$ is expressed as:
Equation (B-10) can be written in matrix form as:

\[ \begin{bmatrix} a & 0 & \ldots & 0 \\ 0 & a & \ldots & 0 \\ \vdots & \ddots & \ddots & \vdots \\ 0 & 0 & \ldots & a \end{bmatrix} \]

\[ \begin{align*}
\alpha I &= \begin{bmatrix} a & 0 & \ldots & 0 \\ 0 & a & \ldots & 0 \\ \vdots & \ddots & \ddots & \vdots \\ 0 & 0 & \ldots & a \end{bmatrix} \\
\end{align*} \]

where

\[ D_{jk} = \sum_{l=1}^{R} \sum_{i=1}^{M} \left( \frac{\partial T_i}{\partial q_j} \cdot \frac{\partial T_i}{\partial q_k} \right)^{p+1-1} \]

\[ f_j = \sum_{i=1}^{R} \sum_{i=1}^{M} \left( (Y_i - T_i^0) \cdot \frac{\partial T_i}{\partial q_j} \right)^{p+1-1} \]

where \( j \) is free index, \( k \) is dummy index, and \( j, k = 1, 2, \ldots, M \).
Appendix C. Effect of the distance between sensor locations and cooled surface on heat fluxes

In the current study, temperatures measured by intrinsic thermocouples (sensors) placed behind the surface (at the bottom of holes approximately one millimetre above the bottom surface) were used to estimate surface heat fluxes. Generally, the further behind the cooled surface are the sensors located, the more significant the effect of temperature damping and time lag in the response of the internal sensors, which makes the analysis more difficult and the results less accurate. In this section, temperature variations with time at seven locations behind the bottom surface of the plate (7.62 mm thick), as shown in Figure C-1(a), were calculated by the two-dimensional axisymmetric heat conduction model with a similar domain and applied boundary conditions as shown in Figure 4-3 except that only one hole in the centre was considered. Then these seven temperature curves were used as "measured" temperatures for calculating heat flux $q_1$s using the inverse heat conduction model. The calculated $q_1$s are plotted in Figure C-1(b). It can be seen that the difference among the calculated $q_1$s and the applied is not obvious. In Figure C-1(c), relative errors are compared between the applied boundary condition $q_1$ and the calculated with temperatures at two locations. These two locations are 1 mm behind the bottom (means that the depth of the hole is 6.62 mm) and at the top surface. The relative errors between the applied and the calculated heat fluxes with the temperatures at 1 mm behind the bottom surface are less than 5% and in the second case, most of the errors are less than 5% except that four values are between 5% and 20%. However, these four errors occur at
heat fluxes less than 0.5 MW/m², which means their influence on calculation is relatively small, and the further behind the cooled surface are the sensors, the bigger the difference.

Figure C-1 Influence of the depth of the hole on calculated heat fluxes at surface

(a) Calculated temperatures at the bottom of holes with seven distances from the bottom surface
(b) Calculated heat fluxes with temperatures given in Figure C-1(a)
(c) Relative error between applied heat flux \(q^{\text{applied}}\) and the calculated \(q^{\text{cal}}\) with temperatures given in Figure C-1(a)
The analysis indicates that temperatures measured at the top surface can be used in the given case by the inverse heat conduction model to calculate heat fluxes at the bottom with acceptable accuracy. The purpose of this analysis is to simplify the preparation of test samples, while computational time is significantly increased since more future time steps are required by the inverse heat conduction model.
Appendix D. Effect of the surface thermocouple wire on the internal temperature measurements

In the current study, both surface and internal temperatures of the plate were measured. Li (2003), and Tszeng and Saraf (2003) found that the thermocouples mounted on the surface measured significantly lower temperatures than expected, likely due to the "fin effect", since the thermocouple wires extend and expose more surface to convection and thus augmenting the heat transfer in the vicinity of the welded points (White, 1984). In this section, the "fin effect" on the measured internal temperatures and calculated heat fluxes on the bottom surface using the inverse heat conduction model will be discussed.

Figure D-1 shows the applied heat flux \((q_i)\) and the domain for the analysis. The heat flux \(q_1\) is applied to the bottom of the plate. The heat flux applied to the edge of the first element (its length is the same as the radius of the thermocouple wire i.e. 0.127 mm) is supposed to have the same effect as a thermocouple wire with a diameter of 0.254 mm. The boundary conditions at other three surface of the domain are also given.

Figure D-2 exhibits the calculated surface and internal temperature profiles under both cases. Compared with the calculated temperatures ignoring the effect of the surface thermocouple the calculated temperatures considering the fin effect are lower. And the difference is significant between the calculated surface temperatures. The maximum difference between the calculated internal temperatures is 5.7 °C, which does not cause substantial difference in the calculated heat fluxes as show in Figure D-3. Most of the relative deviations ((\(q''_{\text{applied}}-q''_{\text{calculated}}\))/\(q''_{\text{applied}}\)) between the applied and the calculated heat fluxes are smaller than 5% as indicated by Figure D-4.
Figure D-1 The applied heat flux and the domain of the analysis.
Figure D-2 Calculated surface and internal temperature with and without considering the effect of the surface thermocouple.

Figure D-3 Comparison between the applied and the calculated heat fluxes.
Figure D-4 Relative error between the calculated and the applied heat flux