HEAT TRANSFER DURING MULTIPLE JET IMPINGEMENT ON THE TOP
SURFACE OF HOT ROLLED STEEL STRIP

By

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Abstract

The cooling which occurs on the runout table (ROT) is a key processing step for hot rolled steel strip. It determines the final microstructure and thus mechanical properties as well as flatness of the hot band. The use of multiple jets during ROT cooling results in interactions between neighboring water jets which can affect the overall heat transfer rate. The heat transfer which takes place during cooling with multiple jets is fairly complex and the available knowledge is limited. The research work described was done to obtain an understanding of the effect of varying nozzle-to-nozzle distance, plate speed and flow rate of the impinging water on the heat transfer taking place on the ROT.

Experiments were performed on the pilot scale runout table available at UBC, using instrumented test samples of steel. Each sample was instrumented with twenty thermocouples which measured the internal thermal history. This data was then used in conjunction with an Inverse Heat Conduction (IHC) model to calculate surface heat fluxes and temperatures. Some of the variables examined included: speed of movement of the test plate (0.22 m/s and 1 m/s), nozzle spacing (114.3, 76.2 and 38.1 mm) and water flow rate (15 l/min and 30 l/min). These experimental results provide important information for the development of improved runout table cooling models.

The results indicated that, during multiple jet cooling, high heat extraction takes place directly below the nozzles and adjacent to them due to direct impact of water. Lower heat extraction occurs at the locations between the nozzles, i.e. the interaction region. Visual observations of the tests indicate that, when the water jets hit the strip, a small darkened zone can be observed at the impingement point below each nozzle. In the

interaction region, the water flowing from the two adjacent jets interacts with each other and large splashing of the water is observed in this region. The dark zones below all three nozzles expand with cooling of the strip, indicating that the water front is progressing outwards from the stagnation line and more water solid contact is taking place. The boiling curves below each nozzle are similar to each other and clearly show the different boiling regimes while the boiling curve for the interaction region does not show these regimes as clearly. In the interaction region, heat transfer remains relatively low until the water completely wets the strip. The investigation of the effect of strip speed indicated that the heat fluxes are higher for lower strip speeds as the strip spends a longer time under the nozzle. This effect was seen both below the nozzle and in the interaction region. In general, increase in water flow rate increases heat fluxes at all measuring locations due to higher amount of water impinged on the strip surface. The nozzle configuration having two adjacent nozzles at 38.1 mm apart has more cooling capacity than the other two configurations indicating that, having two nozzles close to each other enhances heat transfer.

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Nomenclature

c_p	Specific heat (J/kg°C)
d_j	Circular jet diameter at exit of the nozzle diameter (nozzle diameter) (mm)
d_{ji}	Jet diameter at impingement (mm)
D	Diameter of test cell (mm)
g	Gravitational acceleration (m/s ²)
h	Heat transfer coefficient (W/m ² °C)
Z or H	Distance between the nozzle and the heated plate (mm)
k	Thermal conductivity (W/m°C)
L	Characteristic length (mm)
p	Pressure (Pa)
q''	Heat flux (W/m ²)
r	Radial coordinate/radius of impingement zone (mm)
t	Time (second)
T	Temperature (°C)
T_{sat}	Saturation temperature of fluid (°C)
v	Velocity (m/s)
v_j	Jet velocity at exit of the nozzle (m/s)
v_{ji}	Jet velocity at the impingement point (m/s)
W	Width (mm)

Greek symbols

ΔT_{sat}	Wall superheat, the difference between surface temperature and fluid saturation temperature (°C)
ΔT_{sub}	Subcooling, the difference between fluid saturation temperature and fluid temperature (°C)
ρ	Density (kg/m ³)

Subscripts

CHF	Critical Heat Flux
FB	Film Boiling
FNB	Fully Developed Nucleate Boiling
g	Vapour
i	Impingement
ini	Initial
j	Jet
l	Liquid
MHF	Minimum Heat Flux
n	Nozzle
NB	Nucleate Boiling
ONB	Onset of Nucleate Boiling
PNB	Partially Developed Nucleate Boiling
sat	Saturation
sub	Subcooling
surf	Surface
TB	Transition Boiling
w	Wall

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1. Introduction

Steels are one of the most widely used structural materials and there is a trend to develop advanced high strength steels (AHSS). These novel steels can, for example, substantially contribute to the improvement of fuel efficiency of vehicles by allowing vehicle weight to be significantly reduced. Similar arguments can be made for steels that are being used in pipelines or other construction applications. A substantial portion of these steels, e.g. 100% for line-pipe grades, is used as hot-rolled products. The superior properties of AHSS are associated with a complex microstructure which results from the austenite decomposition during cooling on the run-out table of a hot mill. Increasingly complex cooling paths are required to ensure that the desired properties are achieved. For example, a number of novel hot-rolled dual-phase (DP) steels for automotive applications are coiled close to room temperature.

A conventional hot strip mill consists of a reheating furnace, descaling units, roughing and finishing stands, runout table and downcoilers. After exiting the finish mill, steel strip enters the runout table at around 850-950°C moving with the typical speeds of around 5-10 m/s. The cooling process on the runout table is carried out by banks of round water jets, sprays or planar jets (water curtains). The final mechanical properties of hot rolled steel largely depend on the applied cooling regimes. Thus, thorough understanding of the heat transfer processes encountered on the runout table as well as development of accurate heat transfer models are crucial for obtaining hot rolled steel of desired quality.

Although, jet impingement cooling has been extensively used in industrial applications, very limited cooling data have been obtained for industrial conditions and the

information available is not well understood. Recent experimental studies have been carried out on stationary plates using a single jet. Although very useful information has been obtained on various boiling modes during jet impingement cooling of hot strips these tests have not been able to supply all the relevant information for runout table heat transfer modeling. An important parameter in water jet cooling is to understand the interaction between the jets.

Extensive work is ongoing at UBC to develop quantitative knowledge on heat transfer during ROT cooling using the pilot scale ROT facility available within the Centre for Metallurgical Process Engineering (CMPE), UBC. Using this facility instrumented moving steel strips were cooled by impinging water from three nozzles on the top surface of strip and measured thermal history was given as input to an IHC model so that surface heat fluxes and temperatures could be calculated. The video images from the tests were also analyzed to correlate the findings from measured data with visual observations. The research work described here mainly attempts to develop an understanding of heat transfer across the strip width, between two jets along with understanding the effect of varying nozzle-to-nozzle distance, plate speed and flow rate of the impinging water on the heat transfer taking place on the ROT. The experimental results and analysis from this work provide important information for the development of improved runout table cooling models.

2. Literature Review

In the steel industry, the mechanical properties of hot rolled products can be adjusted by obtaining the correct microstructure through controlled cooling. This cooling takes place on the runout table (ROT) and is achieved by impinging water from multiple jets onto the top and bottom of the hot rolled, moving steel strip. In multiple jet impingement, water from the top nozzles spreads radially, partially interacting with spread from adjoining nozzles and partially merging with neighboring streams in a continuous parallel flow formed by the subsequent jetline ^[1]. The prominent mode of heat transfer on the runout table is jet impingement boiling. Other modes of heat transfer, such as, free and forced convection to air and radiation to the surrounding also exist, but are less prominent.

Boiling heat transfer has been studied by many researchers during the past 50 years. This work has been summarized in a number of review papers ^{[2] [3] [4] [5]}. Steady state tests were done to develop a fundamental understanding of boiling heat transfer phenomenon while transient tests have been done to develop a better understanding of the phenomenon under industrially relevant conditions. Most of these tests were carried out on stationary strips, cooled by a single jet and very few were aimed at understanding the impact of moving strip and multiple jet impingement. They provided valuable information about the heat transfer mechanisms during cooling, but the effect of important parameters like speed of the strip and interaction between jets needs to be investigated systematically. The jet impingement boiling heat transfer investigated so far is reviewed in the following sections.

2.1 Typical Industrial Runout Table

The hot rolled steel strip, on exit from the finishing mill enters the runout table at about 850 °C. Typical runout tables consist of a set of cooling banks mounted at the top and bottom along with motorized rolls and a down coiler, as shown in Figure 2.1:

Figure 2.1: Schematic of the typical runout table (modified from ^[13])

Each cooling bank is made up of several headers, which in turn supply water to one or two rows of jets. Each row of jets is called a jet line, within each jetline an array of nozzles across the strip width are used. The nozzles used are typically either of spray, circular or slot-type. Spray nozzles are used for spray cooling, circular nozzles for laminar flow cooling and slot type nozzles for water curtain systems. These systems are schematically shown in Figure 2.2:

Figure 2.2: Three cooling systems used for ROT (modified from ^[7])

Tacke et al. ^[6] performed a study to gauge the cooling capacities of the three aforementioned cooling systems. It was found that water curtain and laminar flow cooling had higher cooling capacities than spray cooling thus greatly increasing overall heat transfer rates. Results from Kohring et al. ^[7] had similar findings which showed laminar flow cooling to be about 10-30% more efficient than spray. Water curtain and laminar flow are desirable as the jets can penetrate the vapor film on the cooled surface and thus long liquid-solid contact time ^[8]. Further Zumbrunnen et al. ^[9] and Chen et al. ^[10] were able to show that water curtain cooling provided a uniform cooling rate across the width of the strip since the water jet spanned the entire width of the strip. They also showed by comparing planar and circular jets that, planar jets were 48% more efficient at removing heat in the case of a test strip heated to 900°C and traveling at 10m/s. However, laminar flow cooling systems can provide more efficient cooling per unit

volume of water. The disadvantage for this type of cooling system is in non-uniform cooling rate that occurs across the width of the strip.

2.2 Hydrodynamics of Jet Impingement

Water impinging from a jet onto a solid surface can be divided into five different configurations i.e. free surface, plunging, submerged, confined, and wall. Schematics of two of these configurations are shown below in Figure 2.3.

Figure 2.3: Schematic of jet configurations (Modified from ^[2])

When water from nozzle enters an immiscible atmosphere like gas and travels comparatively undisturbed to the impingement surface, the jet is called a free-surface jet. In this case, the liquid flows from the edges of the solid surface without forming a pool on the surface. In the case of a plunging jet, the water from the nozzles enters into existing pool of water on the surface and the flow is disturbed by this pool. On the ROT the free surface jet (Figure 2.3a) is typically observed at the start of the process and plunging (Figure 2.3b) occurs later when a pool of water is present.

Representative conditions for a circular, free surface jet are shown in Figure 2.4 along with the inviscid (nonviscous) pressure and streamwise velocity distribution under the jet and away from it.

Figure 2.4: Inviscid pressure and velocity profile and flow regions for a circular jet (modified from ^[2])

As can be seen in Figure 2.4, three regions of flow on the surface can be identified, namely:

- A) The stagnation region where the velocity in the x-direction is close to zero.
- B) The acceleration region where the water velocity in the x-direction accelerates as it moves away from the centre of the jet.
- C) The parallel flow region where the water velocity in the x-direction is the jet velocity.

Dynamic contribution of the impinging jet leads to maximum pressure at the stagnation point, which subsequently decreases with increasing radial distance. A combination of the stagnation and acceleration regions is called the impingement region or impingement zone ^[11]. This scenario changes for plunging jets as the exchange of

momentum between the jet and liquid is large, causing the flow to decelerate and expand laterally just before impingement.

Liu et al. ^[12] developed a correlation which defines the radius of the impingement zone such that

$$1.57 < \frac{r}{d_{ji}/2} < r_t = 600 \left(\frac{v_{ji} \cdot d_{ji}}{v} \right)^{-0.422} \quad (2.1)$$

Liu ^[13] experimentally found that the boundary of the thermodynamic impingement zone, (zone of constant heat flux) for a circular jet ranged from 1.67 to 2.11 times the diameter of the circular water jet. Ochi et al. ^[14] defined the impingement zone as the zone where the streamwise velocity is in proportion to the distance from the stagnation point and found the impingement zone to be bound by $r/d_j \leq 1.28$, along a surface of a strip impinged by a circular jet of water. The pressure distribution also determines the local saturation conditions of the liquid along the surface. Variations in the saturation temperature of the liquid T_{sat} cause variations in the degree of local subcooling, ΔT_{sub} , and wall superheat, ΔT_{sat} . Hauksson ^[11] found that this effect causes up to a 5°C increase in the saturation temperature at stagnation.

2.3 Experimental Methods for Jet Impingement Boiling

Performing a direct study of heat transfer on a full scale runout table is difficult and unrealistic. Therefore, researchers have performed experiments on small laboratory scale apparatus or relatively large pilot scale facilities, which include one or an array of jets with round or rectangular nozzles impinging on stationary or moving test specimens. These tests are performed under both steady state and transient cooling conditions. Table 2-1 provides a summary of these tests reported in the literature.

Table 2-1: Jet impingement test conditions

Reference	Experimental parameters				
	Test method	No. of nozzles, Nozzle spacing (mm)	Sub-cooling, ΔT_{sub} (°C)	Strip speed, m/s	Sample material, and dimension (mm)
Ishigai et al. 1978 ^[15]	Transient	Single	5-55	0	Stainless steel, 50×12×2mm
Miyasaka et al. 1980 ^[16]	Steady state	Single	80	0	Copper with platinum foil surface, D=1.5mm
Hatta et al. 1983 ^[17]	Transient	Single	30-85	0	18-8 stainless steel, 200×200×10mm
Ochi et al 1984 ^[18]	Transient	Single	5-80	0	Stainless steel, 210×50×2mm
Kumagai et al. 1995a ^[19]	Transient	Single	0-50	0	Copper 150×120×20mm
Kumagai et al. 1995b ^[20]	Transient	Single	14-50	0	Copper 150×120×20mm
Mitsutake et al. 2001 ^[21]	Transient	Single	20-80	0	Copper, brass and carbon steel D=94mm, t=60mm
Chen et al. 1991 ^[22]	Transient	Single	75	0	Stainless steel, 355×254×6.35mm
Hall et al. 2001a ^[23]	Transient	Single	75	0	Copper, D=112, t=25.4
Filipovic et al. 1995a,b,c ^{[24][25][26]}	Transient	Single	45-75	0	508 ×38.1×25.2 mm, Copper
Robidou et al. 2002,2003 ^{[27][28]}	Steady state	Single	5-15	0	8 Copper heaters, each 10×10×5mm
Hauksson 2001 ^[11] Meng 2002 ^[29]	Transient	Single	30-95	0	Stainless steel, DQSK 280×280×7.62mm
Monde et al. 1980 ^[30]	Transient	2-4 jets	<5	0	Copper D=25
Sakhuja et al. 1980 ^[31]	Transient	Array of jets (jetline)	18-77	0	Copper 51×152×102mm
Slayzak et al. 1994 ^{[32][33]}	Steady state	2 jetlines of 6 jets each, 6.3mm	30	0	Steel 0.66×35.7×260mm

Table 2-1: Jet impingement test conditions - continued

Author	Experimental parameters				
	Test method	No. of nozzles, Nozzle spacing (mm)	Sub-cooling, T_{sub} (°C)	Strip speed, m/s	Test cell material, and size (mm)
Haraguchi et al. [34]	Transient	3/5,80/40	30	0	Copper 5×300×300mm
Liu et al. [13]	Transient	2/3, 80/90/140	13-30	0	Steel 280×280×7.6mm
Hatta et al. [35]	Transient	Single	8-18	0.48, 0.72, 1.2, 1.8, 2.4 m/min	Stainless steel 240×100×10mm
Zumbrunnen et al. [9]	Transient	Single	25	0.8 max.	Stainless steel 1.6×4.13×27.56cm
Chen et al. [8][10][36-38]	Transient	Single	22-27	Upto 1.5	Low carbon steel 6.35×355×254mm
Prodanovic et al. [39]	Transient	Single	20-30	0.3 and 1	Low carbon steel 6.38×430×1200mm

D = diameter, t = thickness

During steady state tests, the heat is continuously supplied to the test specimen with the rate equal to or greater than the rate of heat extraction. The temperatures were measured by the thermocouples placed at selected interior positions. The heat flux estimation was done directly from the heat input supplied by the heat source and heat transfer coefficients were calculated by solution of Fourier's law of heat conduction and Newton's law of cooling.

During transient tests, the test specimen was heated to a specified temperature and then cooled by jet impingement. The temperatures were measured by intrinsic thermocouples located at specified positions. The time-temperature history recorded during the test was analyzed by numerical methods, such as inverse heat conduction model, to determine the heat flux and heat transfer coefficients.

The boiling curves for steady state conditions may look different from those obtained by transient experiments, but the trends are quite similar.

The boiling curves measured by Robidou et al. ^[27] at three different locations can be divided into four distinct regimes with different heat transfer mechanisms and extent of heat extraction. The boiling curves measured at the stagnation line are significantly different from the ones measured in the parallel flow region. In general, the boiling curve at stagnation has higher heat fluxes in comparison to those in the parallel flow region. The transition boiling regime is observed to occupy a broader range of temperatures at the stagnation line and the heat fluxes are as high as the critical heat flux. In the parallel flow region the film boiling appears at about 200 K smaller heater temperatures. The high heat fluxes associated with the transition boiling regime are probably due to the micro-bubble emission boiling phenomenon which enhances the heat transfer in this regime.

The emphasis of the study carried out by Hall et al. ^[23] was on the role of the distance from the stagnation point and they presented the boiling curves measured at five different locations. The heat flux was observed to increase quickly with the decrease in the wall temperatures on start of jet impingement. In the parallel flow region, heat fluxes reach maximum value and decrease to a minimum heat flux. The heat fluxes then rise to the critical heat flux and start to decrease till the single phase convection is reached. In the stagnation region, stable film boiling and minimum heat flux were not observed. This study also demonstrated the effectiveness of using transient state tests to obtain spatial distributions of boiling data for a wide range of surface temperatures.

2.4 Boiling Heat Transfer

2.4.1 Pool Boiling Curve

In jet impingement cooling, heat is extracted from the solid surface and the temperature of the solid decreases. This extraction of heat can be quantified in terms of heat flux. Also, the difference between surface temperature and cooling fluid temperature is called wall superheat. The variation in heat flux can be expressed as a function of wall superheat, and this representation is called a “boiling curve”.

Before starting the investigation of boiling heat transfer during jet impingement, it is necessary to understand different heat transfer mechanisms that occur during pool boiling. During pool boiling, the solid surface is covered by a pool of water and cooling occurs at the interface between the water and the solid surface. Figure 2.5 shows a typical pool boiling curve obtained under steady state conditions.

Four different heat transfer regions exist on the boiling curve. They are single-phase forced convection, nucleate boiling, transition boiling, and film boiling, changing with increasing wall superheat.

Figure 2.5: Typical pool boiling curve

Until Point A on the pool boiling curve; the heat extraction from the heated surface is by means of single-phase forced convection. Point A indicates a transition from single-phase forced convection to the incipience of nucleate boiling, also known as onset of nucleate boiling (ONB). Small bubbles begin to form in tiny crevices on the surface. Heat is transferred from the heated surface via conduction from the plate through the liquid, evaporation of the liquid, and micro-convection within the fluid caused by bubble growth and detachment. As the surface temperature is increased to point A' on the pool boiling curve, the heat transfer mechanism changes from partially developed nucleate boiling to fully developed nucleate boiling. During fully developed nucleate boiling region maintenance of bubble growth and subsequent detachment into the colder fluid above is sustained. A continued increase in the surface temperature causes a subsequent

increase in the heat flux throughout the nucleate boiling region up to a maximum point known as the critical heat flux (point B). This is due to an enhancement in mixing associated with micro-convection during bubble growth and departure, and with intimate water-solid contact on the heater surface.

With increasing wall superheat further, the point at which the critical heat flux occurs will change into the transition boiling regime, found between points B and C on the boiling curve. With it comes a sharp decrease in heat flux with an increase in wall superheat. Conditions are highly dynamic such that short periods of nucleate boiling and film boiling occur on the same heated surface during wet and dry periods, respectively. The variation in heat flux with wall superheat is a result of the change in the fraction of time each boiling mode is present on a given area ^[40]. With continued temperature increase, this would result in a decrease in the area fraction covered by the liquid-heat surface interface. This drop in heat flux is mainly a result of significant bubble generation and coalescence of bubbles on the heated surface since water-vapour has a significantly lower capacity to remove heat than that of liquid water.

With a continued rise in wall superheat, the heat flux drops to a minimum heat flux, which is denoted as the Leidenfrost point (point C). Point C, marks the final transition to a region where the heated surface is able to supply enough heat energy to support a continuous vapor barrier. The mode of heat transfer is primarily by convection of the vapor, with radiation becoming dominant at higher surface temperatures ^[2]. From point C towards continuously increasing wall superheats, a stable film boiling regime is said to exist. This effectively separates the liquid contact from the heated surface.

Increase in surface temperature is a result of poor heat transfer capacity of water-vapor in removing heat.

2.4.2 Jet Impingement Boiling Curve

Ishigai et al. ^[15] obtained data by cooling a steel strip from an initial temperature of approximately 1000°C with an impinging top jet of water with jet velocity of 2.1 m/s, sub-cooling of 35 °C and taking measurements at stagnation. Figure 2.6 (results taken from ^[15]) shows the boiling curve obtained by these transient jet impingement experiments.

Film boiling prevails from the beginning of the cooling to the point E. In this period, a stable vapor film is formed between the cooling surface and the water. They reported that the cooling surface is visible through the water and the vapor film, and the vapor interface is smooth. Due to this stable and smooth vapor interface, no boiling noise is heard during the experiment. The heat flux decreases as the wall temperature monotonously decreases. At point E, the stable film collapses with generation of the boiling noise. The heat flux has its minimum at this point. As the temperature decreases, the heat flux sharply increases from the point E to D. But it remains almost constant over the interval from D to C, i.e. the boiling curve has a 'shoulder' in this region. The vapor interface looks white, probably due to the frequent and instantaneous liquid-solid contacts from the end of the film boiling to the point C. The cooling surface is observed to get wet when the temperature passes point C. The heat flux increases and the wet zone spreads to the whole surface. In the meantime, the boiling changes from transition boiling to nucleate boiling passing through the maximum heat flux point (i.e. point B). As the

temperature falls further, the boiling ceases, and heat is transferred under non-boiling condition. This corresponds to the left side of point A in Figure 2.6.

Figure 2.6: Sub cooled water jet impingement boiling curve (modified from ^[15])

2.4.3 Jet Impingement Heat Transfer Mechanisms

Jet impingement boiling heat transfer shows different regimes, depending on the velocity of the jet and the moving strip, temperatures of the liquid and impinging surface, flow rate of the liquid, roughness and oxidation of solid surface as well as the thermo-physical properties of both the liquid and the solid. During transient cooling of a hot steel strip by a liquid jet, different heat transfer rates are observed at different distances from the stagnation point. This results in large variations of the surface temperature and

consequently different boiling regimes occur along the plate at the same time. It is expected that all different boiling mechanisms that take place during pool boiling occur for jet cooling on the runout table since the strip surface is subjected to a range of temperatures. Identified regions within the boiling curve are related to the impingement zone, the parallel flow zone and the radiation zone^[41, 42, 43].

The prime cooling mechanisms in the impingement zone are forced convection and radiation. The region beside the impingement zone is parallel flow zone having stable film boiling as dominant heat transfer mechanism. In addition, radiation is present and would be significant in the high surface temperature region. Viskanta et al.^[44] have observed jet impingement cooling for an industrial run out table and Kokado et al.^[45] for a stationary strip and found the varying modes of heat transfer to be consistent with the above mentioned regimes.

Figure 2.7: Heat transfer regimes adjacent to an impinging jet on a stationary strip (modified from^[44])

Figure 2.7 shows top jet impingement and the various heat transfer regimes as observed by Viskanta et al. ^[44]. In region I, close to the nozzle, they observed single phase forced convection. Further away from the center as the surface temperature increases, the onset of nucleate boiling was observed and, further away from the jet, transition and film boiling was observed.

These mechanisms are also affected by nozzle shape and size, height of nozzle from the strip surface and the distance between two nozzles or two jetlines.

2.4.4 Single Phase Forced Convection

When the wall temperature is lower than or equal to the saturation temperature of the liquid at a given pressure, there is no boiling and the mode of heat transfer is single-phase forced convection. This mode of heat transfer is defined by the region, to the left of point A in Figure 2.5.

Wolf et al. ^[46] found that the heat transfer in the single phase convection regime is strongly influenced by the streamwise distance from the stagnation line. The single phase convective heat transfer coefficient was observed to vary over the surface as the boundary layer developed and was dependant on the hydrodynamics of the water which is function of jet velocity.

2.4.5 Nucleate Boiling

In this region, the wall temperature exceeds the saturation temperature of the liquid and discrete bubbles begin to detach from the surface and enhance the local fluid motion, causing the convection heat transfer coefficient to increase. With increasing heat flux or wall temperature, the generation of vapor progresses from a few relatively small bubbles at point A to many larger bubbles coalescing near point B. Point A in Figure 2.5

marks the onset of nucleate boiling (ONB) and the region between points A' and B on Figure 2.5, represent fully developed nucleate boiling (FNB). The transition from single-phase convection to fully developed boiling takes place between points A and A' and is called partially developed nucleate boiling (PNB).

Wall superheat has a strong effect on the heat flux in the nucleate boiling regime under both steady state and transient cooling conditions. The effects of subcooling and jet velocity on nucleate boiling have also been investigated widely. Miyasaka et al. found that increasing jet velocity and subcooling, independently, shifts the onset of nucleate boiling to higher heat flux and wall superheat ^[16]. Robidou et al also observed a similar effect of subcooling on nucleate boiling ^[27].

Many researchers, in general, found in the fully developed nucleate boiling regime that, the heat flux is almost independent of both subcooling and jet velocity under steady state cooling conditions ^{[47][48][15][46][27]}. Monde and Katto ^[48] investigated the effect of subcooling at stagnation and found that subcooling influenced the nucleate boiling at low wall superheat but not at higher wall superheats.

Under transient cooling conditions, the observations are different. Kumagai et al. ^[20] reported that boiling curves for subcoolings of 14 to 50 °C at stagnation coincided with each other in the nucleate boiling regime. While Ochi, et al. ^[18] reported that the heat fluxes shift to higher values and differences in heat fluxes were evident in the nucleate boiling regime.

Robidou et al. ^[27] found under steady state cooling conditions that in the fully developed nucleate boiling regime, the boiling curves in different locations merged and heat fluxes are independent of the distance from the stagnation line. Wolf et al. ^[46]

reported similar findings. However, under transient cooling conditions, this conclusion was not supported by the data of Hall et al. [23].

2.4.6 Critical Heat Flux

Point B on Figure 2.5 is the point of maximum or critical heat flux, at which many large bubbles are formed leading to maximum heat extraction from the surface. Raising the wall superheat beyond this point leads to a coalescence of those large bubbles formed, which prevents the liquid reach the solid surface; this causes the heat fluxes to drop from this point onwards.

Various steady state and transient jet impingement cooling studies have led to the conclusion that both, jet velocity and water subcooling raise the critical heat flux (CHF) point to higher values [46][15][16][20]. In general, water subcooling is believed to have a significant influence on CHF in comparison to jet velocity.

2.4.7 Transition Boiling

In Figure 2.5 the region between points B and C marks the transition boiling regime. As the coalescence of large bubbles formed at CHF starts, it leads to formation of vapor blankets on the surface and the heat flux start to drop with rising wall superheat. This vapor blanket is unstable and collapses intermittently resulting in large fluctuations in surface temperature and heat flux. Therefore, transition boiling is considered as combination of nucleate boiling during liquid-solid contact and unstable film boiling during vapor-solid contact. This region is considered as the least understood of all boiling regimes, due to the inherent complexity of this phenomenon and the difficulties associated with experimental studies [18]. In the transition boiling region the intermittent liquid-solid contact is increasingly reduced and the heat flux values keep on dropping with

rising wall superheat till the point of minimum heat flux (MHF i.e. point C on Figure 2.5), which gives rise to stable film boiling from MHF onwards. Transition boiling regime of heat transfer during cooling has been investigated by various researchers and is quite relevant for runout table cooling and will, thus, be reviewed here.

Robidou et al.^{[27] [28]} found that, at the stagnation point, very high heat fluxes can be extracted in the transition boiling regimes. They related this phenomenon to the break up of vapor bubbles into many micro-bubbles, which induces a local mixing of the fluid. This leads to better wetting of the surface and, hence, better heat transfer. The heat flux starts to decrease at the onset of transition boiling. This suggests that a partial dry-out of the surface occurs until a minimum in the heat flux is reached. Then the bubbles start to break up into microbubbles and the heat transfer increases. This minimum is shifted to higher heat fluxes and smaller surface temperatures, as subcooling increases. The effect of jet velocity on transition boiling at stagnation is not significant, probably due to its small variation. For a surface jet, the nozzle to strip spacing is found to have a similar effect on transition boiling as that of increasing jet velocity.

Ishigai et al.^[15] found in the transition boiling region that the curve is shifted to higher heat flux and higher wall superheat by increasing the impinging velocity or the subcooling.

Ochi et al.^[18] and Kumagai et al.^[19] measured heat transfer at locations downstream from the stagnation point and found the heat transfer in transition boiling region to decrease as the distance from the stagnation point increased.

2.4.8 Minimum Heat Flux

Point C in Figure 2.5 represents the minimum heat flux at which the transition from transition boiling to film boiling takes place, and is commonly referred to as the Leidenfrost point. After this point, the intermittent solid-liquid contact ends and a stable vapor film is formed changing the mode of heat transfer to film boiling. The temperature corresponding to the minimum heat flux point is considered very important in jet impingement boiling as it is often related to progression of the rewetting front and also known as rewetting temperature ^[25].

The MHF shifted to higher wall superheats and heat fluxes with the increase in subcooling. Although, the heat flux is found to increase with the increase of jet velocity, minimum film boiling temperature is unaffected by the jet velocity ^{[15] [18] [27] [49]}.

2.4.9 Film Boiling

In the film-boiling region, from point C on Figure 2.5 and beyond, liquid-solid contact does not occur as the surface is fully covered by the vapor film. The heat transfer from the surface to the liquid is across a vapor film and forced convection within the vapor film is the primary mode of heat transfer at first and radiation becomes a more dominant mode of heat transfer as the temperature is increased ^[2].

Ishigai et al. ^[15] observed film boiling only at a subcooling lower than approximately 50°C. In the film boiling region, no liquid-solid contact is observed, and the beginning of the contact coincides with the minimum heat flux point.

Liu and Wang ^[49] experimentally showed that subcooling and jet velocity have a strong effect on the heat flux in film boiling. The effect of velocity was similar to what

was presented by Ishigai et al. ^[15], where the curve shifts upwards with increasing velocity.

2.5 Effect of Multiple Jet Impingement on ROT heat transfer

In multiple water jet impingement, water from a top nozzle spreads radially, partially interacting with spread from adjoining nozzles and partially merging with neighboring streams in a continuous parallel flow formed by the neighboring row of nozzles ^[50]. Of interest is the optimum design of the cooling configuration, such that the desired cooling rates and uniform temperature distribution is achieved. In situations where jet lines are used which contain many nozzles positioned across the width of the sheet, the interaction of the water on the strip surface is quite complicated.

Filipovic et al. ^[50] identified three cooling regions on the top surface:

- 1) Impingement cooling zone, also called as “effective cooling circle” ^[7]. In this zone, the fresh water from the jet comes in direct contact with the hot strip, hence the subcooling is very high and this prevents the development of film boiling. The dark circle (i.e. black zone) observed under the jet, is considered visual proof of this phenomenon ^[51].
- 2) The interaction zone of two neighboring jets in the same row.
- 3) The interaction zone between two jetlines.

Filipovic et al. ^[52], concluded in their planar jet experiments that heat extraction in the film boiling region accounts for approximately 45% of the total extraction rate, and heat extraction in the top and bottom jet impingement regions account for about 40% of the total. They also found that, heat extraction on the top surface is about 1.7 times

greater than that at the bottom surface and attributed this to pooling of the water occurring only on the top surface.

Monde et al.^[30] examined the effects of multiple impinging jets on nucleate boiling heat transfer for saturated water. Circular, free-surface jets, numbering between two and four, impinged at various locations on the heated surface. They concluded that the degree of scatter in the data is typical for nucleate boiling and hence the number of placement of jets has little or no effect on the heat flux. However, Wolf et al.^[2] indicated on careful scrutiny of their data, that scatter is not random and for comparison with respect to position, when the number of jets is fixed, sizeable and consistent differences in the heat flux exist for a fixed wall superheat. Although, the related physical mechanism was difficult to infer from their limited study, the results suggested that multiple jets and their relative placement do have an effect on nucleate boiling heat transfer but that additional investigations are necessary.

Monde et al.^[30] also examined the effects of multiple jets, and their relative positions, on the critical flux. They concluded that multiple jets yield higher values of CHF than single jet configurations (for a fixed fluid, nozzle diameter and jet velocity) due to shorter flow lengths on the impingement surface.

Sakhuja et al.^[31] examined cooling characteristics for an array of circular water jets used to quench a copper slab (positioned vertically). The jets were equally spaced in a staggered geometry, with nozzle spacing ranging from 4 to 12 nozzle diameters. They reported the nucleate boiling heat flux to be independent of the nozzle-to-plate distance and nozzle diameter but dependant on the wall superheat, nozzle-to-nozzle spacing and square root of the jet velocity. Maximum heat fluxes were observed for nozzle spacing of

approximately 8 to 10 diameters. No information was provided concerning the number of jets or the velocities investigated, nor were any boiling curves provided to reveal the velocity dependence.

Slayzak et al. ^{[32][33]} experimentally studied the interaction both for planar water jets and for circular water jets on a laboratory scale. The diameter of planar jets was 5.1 mm and the diameter for circular jets was 4.9 mm in their tests. The jetline to jetline spacing was set as either 81 mm or 51 mm. Temperature measurements were done on bottom surface of the plate. During their experiments, the steel plate was ohmically heated by direct current to achieve a uniform and constant surface heat flux. For the case of jetline spacing of 51 mm, local heat transfer coefficient distributions of adjacent jetlines (both planar jet or circular jet) were presented which provide a great deal of information. Firstly, there were three high heat transfer regions along the impinging surface, two at the stagnation points of two jetlines and third at the interaction of two jetlines. Considering the diameter of water jets applied, the impingement cooling zone boundary of the water jets was at approximately 2 times of their diameters. The central high heat transfer region occurred in the middle between two stagnation points where it is obviously outside the impingement cooling zones. However, not only the value of local heat transfer coefficients in this area but also their influential scope is comparable to those associated with the jet impingement cooling zones. Slayzak et al explained that this is because of the formation of an interaction fountain locally. Secondly, the effect of impinging velocities of cooling water on the heat transfer is clearly indicated. The local heat transfer coefficient increases with increasing impingement velocity of cooling water. In addition, Slayzak et al also found that decreasing the ratio of impinging velocities of

neighboring water jets, local heat transfer coefficients beneath the weaker water jet reduced by the effects of crossflow imposed by the stronger jet.

Haraguchi and Hariki ^[34] also investigated the effect of interference flow and jet spacing on the uniformity and capacity of cooling arrays of circular water jets in a typical runout table temperature range with the help of a radiation thermometer. A 300 mm square and 5 mm thick copper plate was heated to 1000 °C and cooled from 800 °C by jet water at 30 °C. The water jet diameter, the jetline to jetline distance and the water jet to plate distance were set at 16.7, 450, 1500 mm respectively. The water jet spacing in one jetline was set as either 40 mm (Case A) or 80 mm (Case B). The average heat transfer coefficients along the width of the copper plate in the temperature range of 800 to 500 °C were depicted. Considering the diameter of water jets of 16.7 mm in their test, the impinging surface of copper plate under Case A should have been wholly covered by impingement cooling zones and the impinging surface of copper plate under case B should have been only partly covered by impingement cooling zones. Therefore, they concluded that the reduction of water jet spacing in one row greatly improves the uniformity across the plate wide direction. It was also interesting to notice that the local heat transfer coefficients at edges of impingement cooling zone were lower than those at positions well inside an impingement cooling zone even for Case A, which obviously resulted from the collision of water flows from neighboring water jets.

In order to investigate the effect of interaction of neighboring water jets on the heat transfer, Liu et al. ^[14] conducted five tests on the pilot scale runout table facility at UBC, by applying two or more water jets with diameters of either 18.92 mm or 30 mm.

The distance from water jets to steel plate was fixed at 1500 mm for all of the five tests.

The test parameters are given in table 2-2.

Table 2-2: Test parameters ^[14]

No. of nozzles	Nozzle spacing, mm	Water temperature, °C	Water flux, l/min
2	140	13	58/58
2	140	27	60/60
2	140	28	30/60
2	80	28	30/30
3	90	30	30/30/30

They concluded that the interaction of neighboring water jets has significant effect on the heat transfer of a heated steel plate. When the two flows collide, a strong interaction fountain is observed, accompanied by heavy water splashes. The cooling intensity of cooling water is weakest at the water flow interaction area because of the stagnation of water flows from opposite directions. When there is difference in the water flux of two neighboring water jets, local heat transfer coefficients beneath the weaker water jet are reduced by the effect of crossflow imposed by the stronger jet. From the visual observations they found that the cooling intensity at the interaction increases when the impingement zones of two neighboring water jets collide with each other.

2.6 Effect of Strip Speed on ROT Heat Transfer

The effect of movement of the strip on heat transfer has not received much attention. Some of the work done on moving strip, jet impingement heat transfer will be reviewed in this section.

Hatta et al. ^[35] investigated the cooling process of a moving hot steel plate under a laminar water curtain in their laboratory. Experiments were conducted at five different

moving velocities (0.48, 0.72, 1.2, 1.8 and 2.4 m/min respectively). It clearly indicated that the steel strip's moving velocity has a very significant influence on thermal evolution of the strip. The higher the strip speed, the higher the temperature at a certain position, which means that the steel strip loses less heat when it moves faster for the same cooling conditions. Thus, the strip velocity greatly reduces the heat extraction by impinging cooling water.

The motion of the steel strip affects the flow of cooling water on the surface of the strip and therefore may change the cooling mechanisms involved in the cooling process, if the variation of strip velocity is big enough. Although the water jet impingement tests of Hatta et al were not conducted under a industrial full-scale condition, their results shed some light on the effect of the strip's moving velocity on the thermal evolution of the strip.

Zumbrunnen et al. ^[53] estimated the effect of plate motion on the heat transfer in the film boiling regime. The estimations were carried out mostly through a numerical analysis by using the Navier-Stokes equations to determine the heat and mass transfer distributions for a moving flat surface in which both the effects of surface temperature and surface motion were considered. They showed that the heat transfer is dependant on the plate velocity. Zumbrunnen et al. ^[9] also performed some stationary as well as moving plate experiments using planar jets, impinging on the top surface of the plate. The maximum plate speed was 0.8 m/s and initial temperatures were approximately 800 °C. They identified several possible improvements to their apparatus. They found that the peak heat transfer coefficient occurred on the downward side of the stagnation point where nucleate boiling is more effective. Numerical solutions showed that surface

motion affected local heat transfer coefficients near the stagnation line only when the surface temperature varied spatially.

Chen et al. ^{[11] [36] [37] [38]} and Han et al. ^[8] used 355 mm wide, 254 mm long and 6.35 mm thick low carbon steel plate for bottom jet cooling and initial strip temperatures of 122 °C with boiling and 85 °C without boiling. A rectangular test plate moved at speeds from 0–1.5m/s while being impinged by a 4.76 mm diameter circular nozzle where water jet temperatures ranged from 22–27°. Flow pattern studies indicated that the moving strip elongated the water film in the moving direction where heat transfer was enhanced. The temperature contours indicated that the temperature gradients were large in the impingement region and become smaller away from the cooling zone ^[37]. The cooling zone, found within the stagnation zone, formed into an elliptical shape. As the fluid flowed upstream of the stagnation point it quickly lost its momentum to the opposing viscous forces, while the flow downstream gained momentum due to the surface motion and propagated longer than in the stationary case. Except that the flow downstream had a ragged boundary due to increased turbulence. It was observed that as the plate moving speed increased the cooling length became shorter since the duration of the plate underneath the jet decreased. They showed some interesting findings in that the peak heat flux for the stationary plate was found to be higher than that in the moving plate because of the highly transient initial behaviour of cooling for the stationary case. As the plate speed increased the peak heat transfer coefficient decreased from 110 kW/m²K ($u = 0\text{m/s}$) to 65 kW/m²K ($u = 0.32\text{m/s}$) to 50 kW/m²K ($u = 1\text{m/s}$). In the case of peak heat fluxes, a value of $24 \times 10^6 \text{ W/m}^2$ was found for stationary plate which was higher than the maximum heat flux of $17 \times 10^6 \text{ W/m}^2$ in the moving case. Based on these

studies, Chen et al. suggested using a planar jet system and staggered arrangement for circular jets ^[11].

Prodanovic et al. ^[39] used the same experimental set-up as the present study to investigate the jet impingement cooling of moving strips with speeds of 0.3 m/s and 1 m/s, respectively. The initial temperature was 850 °C and a top jet was used. They found that the cooling rates under the jet increase, with decreasing strip speed and increasing flow rate of the impinging water.

2.7 Importance and Objectives of the Study

A comprehensive literature review presented in the previous chapter highlighted the available knowledge of controlled runout table cooling. Runout table (ROT) cooling is a key processing step for hot rolled steel strip. It determines the final microstructure and thus mechanical properties as well as flatness of hot band. The cooling process on the ROT is carried out with water jets, sprays or water curtains onto the steel strip or plate. The use of multiple jets results in interactions between neighboring water jets affecting the overall heat transfer rate. The heat transfer which takes place during cooling with multiple jets is fairly complex and the available knowledge is very limited.

The goal of the research is to perform some industrially relevant multiple water jet experiments using moving samples. During this research the effects of:

- nozzle-to-nozzle distance
- strip speed
- flow rate of impinging water

on ROT heat transfer was studied.

3. Test Facility and Procedures

The literature indicates that the steady state or transient tests done to investigate jet impingement heat transfer on the runout table are typically single jet, stationary strip tests, carried out using a small scale apparatus. These conditions are far from industrial runout table conditions and although they provide important information towards a fundamental understanding of the heat transfer, results from these tests can not be directly applied to industrial runout tables. In particular these tests are not able to simulate moving strip conditions or multiple jet interactions. It is practically difficult to use an actual industrial runout table to carry out these investigations. A pilot scale runout table facility is a good option, on which the tests can be carried out very close to industrial conditions and the results can be directly applied to industrial runout tables. A unique pilot scale runout table facility is available at UBC in the Department of Materials Engineering. The facility was custom built and is able to simulate cooling of large moving test samples with multiple jet, top and bottom cooling. A comprehensive description of this facility and the initial tests done using it is given by Prodanovic et al. [39].

3.1 Description of the Cooling Facility

The schematic of the pilot scale runout table facility is shown in Figure 3.1. It consists of a moving bed mounted on roller chains (1) and placed on steel tracks, an electric furnace pressurized with inert gas to prevent scale formation during heating period (2), a high torque hydraulic motor with a control valve (3), and a cooling section with water jets (4). The specifications of the facility are given in Table 3-1.

Figure 3.1: Schematic of the pilot-scale runout table facility a) Side view, b) Front view

Table 3-1: The specifications of the facility.

Facility	Specification
Electric heat furnace	Denver fire clay (208V, 92A, 60Hz) Maximum heating temperature: 1287 °C
Water pump	15HP, 3600RPM, 60Hz
Upper tank	1.5×1.5×1.0m
Heater in upper tank	30 kW
Lower tank	3×0.7×1.2m

The tracks are about 15 m long and the cooling section is located 10 m downstream from the furnace. The speed of the strip can be controlled by supplying the appropriate voltage signal from the control computer to the hydraulic drive system. The cooling section consists of an overhead tank or upper tank, a containment tank or lower tank, a heater in the overhead tank, a pump, two headers and nozzles attached to headers. An overhead water tank (volume of 2 m³) is located at the top of the 6.5 m high tower. The water can be circulated between the overhead and containment tank in a closed loop and heated with the heater in the overhead tank. The test section is equipped with two industrial size headers having up to three nozzles each. One header is connected to nozzles with the flexible hose to facilitate the adjustment of the distance between two jet

lines. The headers can be supplied with water from the overhead tank as well as directly from the pump using a by-pass line. In the current study, each nozzle is a “U” shaped stainless steel tube having an inner diameter of 19 mm. The nozzle is connected to a header via a flexible hose through a mechanical valve and a flowmeter. The flexible hose makes it possible to change the distance between two nozzles and still keep them in line. The flow through each nozzle can be adjusted using the mechanical valve and the water flow rate can be measured using the flowmeter. The water supply to each header is controlled using a solenoid valve installed before each header in the pipe line. After cooling the strip, the water is collected in a containment tank, which is placed at the bottom of the tower and returned to the overhead tank via a pump. The multiple use steel frame was designed and constructed for the present study. The blocks can be attached to the frame and were machined such that the end of each nozzle could be held rigidly in place yet the distance between them could be easily adjusted by moving the blocks. The frame can act like a platform which can move vertically or horizontally, so that the stand-off distance and distance between two jetlines can be adjusted. Experimental data including the strip temperature, water flow rate and strip speed are collected via two external data acquisition boards (Daqbook 2005 and INet100) and transferred to a PC using DASyLab 7.0 data acquisition software at the frequency of 31 Hz.

Table 3-2: Measurement errors ^[39]

Quantity	Error
Steel Temperature	$\pm 2^{\circ}\text{C}$ $\text{Temp} \leq 277^{\circ}\text{C}$
	$\pm 0.75\%$ $\text{Temp} > 277^{\circ}\text{C}$
Water Flow Rate	$\pm 1 \text{ l/min}$
Thermocouple Hole Depth	$\pm 0.01\text{mm}$
Strip speed	$\pm 0.05 \text{ m/s}$
Water Temperature	$\pm 1^{\circ}\text{C}$

The estimated experimental errors associated with data collection are presented in Table 3-2. The errors in water flow rate, water temperature, and strip speed measurements were estimated ^[39] from experience in measuring the quantities.

3.2 Test Strip and its Instrumentation

HSLA (High Strength Low Alloy) steel strips supplied by Dofasco in the as rolled condition were used as test samples. The 6.65 mm thick strips were cut to the dimensions of 1200 × 430 mm. The chemical composition of the HSLA steel is given in Table 3-3.

Table 3-3 Chemical composition of HSLA steel, wt %

C	Mn	P	S	Si	Cr	Ni	Mo	Al	N	Ti	V	Nb
0.0512	1.289	0.012	0.0041	0.1015	0.0434	0.0127	0.0106	0.0395	0.0045	0.0032	0.0061	0.0689

Each instrumented strip was used only once, to avoid difference in the cooling response of the strip due to dimensional changes or warpage and surface oxide formation. Visual observations indicated that warpage took place near the end of each test and its effect on the overall heat transfer and jet flow on the strip surface was neglected in this study.

Approximately 25 to 30, Type K thermocouples (Omega INC-K-Mo-1.6mm) were used to measure the transient temperature history of the strip during each test. As shown in Figure 3.2, the locations of the thermocouples were chosen such that the thermal histories below and at the positions in-between the nozzles can be measured in the lateral direction. Thermocouples were also positioned along the length of the test sample. Each thermocouple was instrumented in the strip by drilling a 1.6 mm diameter hole from the back of the strip, and positioned approximately 1 mm below the strip surface.

Figure 3.2: Thermocouple configuration (small dots represent TC locations)

Table 3-4: Nozzle Configurations

Set No.	Centre to centre distance between nozzles, mm
1	114.3 (4.5")
2	76.2 (3")
3	38.1 (1.5")

The thermocouple wires entering the hole were separated by ceramic tube insulator and each thermocouple was spot welded to the top of the flat bottom hole. Figure 3.3 shows the installation of the thermocouple. The welded thermocouples were anchored to the back of the strip with a screw so as to hold the thermocouples in place.

Figure 3.3: Installation of the thermocouple

3.3 Test Procedure

The strip was cleaned with alcohol and its thickness was measured at several places using a micrometer before starting the instrumentation. The depth of each thermocouple hole was precisely measured ($\pm 0.01\text{mm}$) using a vernier caliper.

The strip was bolted to the carrier and the thermocouples were spot welded in the holes. Then the strip was put on the rig carrier and its place was fixed by adjusting the side guides of this carrier to ensure central thermocouple under centre nozzle. The water pump was turned on and the flow of the nozzles was set to desired value and the strip was moved under the nozzles. The distance between the nozzles was adjusted by moving the nozzle blocks so that the water jets impinge at specific locations (marked with a pen on the quench surface) on the top surface of the strip. These locations were determined depending on the specified nozzle configuration for particular test. After adjusting the nozzle positions, the strip was put into the electric furnace and the furnace opening was closed with the refractory bricks and glasswool. The furnace temperature was set to about $920\text{ }^{\circ}\text{C}$. The thermocouples were connected to the data acquisition system (Daqbook) which was connected to the computer through an Ethernet cable. The flow rate, strip speed and the readings from the thermocouples were displayed on the computer screen by using the DasyLab 7.0 in which the module was designed to display and control these parameters. The desired input voltage for the hydraulic pump was set depending on the required speed of the strip for the particular test.

Water was pumped up to overhead tank and the heater in this tank was turned on and set to the temperature $2\text{ }^{\circ}\text{C}$ above the desired temperature of the water. Once the water temperature in the overhead tank reached the set value it was circulated in the

closed loop between overhead tank and containment tank to ensure uniform water temperature which dropped to its desired value during circulation.

The strip was heated to about 920 °C and kept at that temperature for ~ 30 minutes to ensure that it is heated uniformly. When all the temperature readings were at the same value the water circulation was started through the header and nozzles by-passing the overhead tank. The video camera used to record the test was put into place and the strip was taken out of the furnace. The data capturing was started and the camera was turned on. The strip was moved with the desired speed in forward direction and below the impinging jets and stopped after it passed the jets completely. The solenoid valve before the header was turned off to stop the water flow through the nozzles. When the water flow through all the three nozzles was stopped completely, the strip was moved in backward direction towards the furnace well past the cooling section. The water flow was started again and the strip was moved below the water jets for a second pass. This procedure was repeated several times till the internal thermocouple temperatures dropped below 100 °C.

The water flow rate, speed of the strip and centre-to-centre distance between two nozzles were varied in this study while the stand-off distance was kept constant at 1.5 m and the cooling water temperature at 30 °C.

3.4 Data Analysis

The internal time-temperature data obtained from the test was processed using an existing 2-D, axisymmetric inverse heat conduction model ^[54] developed at UBC. Using this IHC model the calculated surface temperatures and surface heat fluxes were obtained. The video and still images obtained during the test were analyzed using Matrox

Inspector 4.0 software for image analysis and compared with the surface temperature profiles obtained from the IHC model to predict the overall heat transfer phenomenon taking place during multiple jet impingement cooling on the top surface of the moving strip.

The jet impingement cooling process is a highly transient problem and a significant thermal gradient exists through the thickness of the strip. Due to the progression of the wetting front on the surface of the strip, the distribution of the heat flux on the boundary varies both temporarily and spatially. In multiple jet impingement study with moving strip, the interaction between jets and the change in hydrodynamics on the moving surface of the strip adds to this complexity. Hence, each thermocouple hole was analyzed separately using the existing 2-D inverse heat conduction model. This model is based on Finite Element Method (FEM) and is used to estimate the boundary conditions (heat flux) by knowing the temperature data at interior known locations in the test strip.

In the present study, an axisymmetric 2-D IHC model was used to analyze the domain as shown in Figure 3.4 below. Since the jet as well as the thermocouple holes are circular, heat conduction in an axisymmetric geometry is considered. Only half of the area around the hole is selected due to the symmetry. Assumption is that the heat transfer takes place predominantly through strip thickness. Heat transfer in the lateral direction assumed negligible. Thus, The heat transfer is predominantly in the z-direction toward the surface cooled by water impingement and only weakly in the r-direction across the sample due to the presence of the hole used to install the thermocouple. Heat transfer in

the circumferential direction (i.e. θ direction) was ignored since a small area (6 mm) around the thermocouple was chosen as the calculation domain.

Figure 3.4: Domain used for IHC analysis

The flow of the heat in the sample can be described according to

$$\frac{1}{r} \frac{\partial}{\partial r} \left(k_r r \frac{\partial T}{\partial r} \right) + \frac{\partial}{\partial z} \left(k_z \frac{\partial T}{\partial z} \right) = \rho C_p \frac{\partial T}{\partial t} \quad (3-1)$$

using a 2-D axisymmetric geometry. Only half of the strip is selected due to the symmetry. Referring to Figure 3.4, the applicable boundary conditions and initial condition for the model are defined as follows:

At the bottom of the domain (i.e. S_1), $z = 0$ mm, at r -positions from 0.8 to 6 mm heat transfer is due to radiation as shown in Equation [3-2].

$$-k \frac{\partial T}{\partial z} \Big|_{z=0mm} = h(T - T_{\infty}) = q \quad [3-2]$$

Where $h = h_r$ and h_r is given in Equation [3-3].

$$h_r = \sigma \varepsilon (T^2 + T_{\infty}^2)(T + T_{\infty}) \quad [3-3]$$

In Equation (3-3), σ is the Stefan-Boltzmann constant, taken as $5.67 \times 10^{-8} \text{ W/m}^2 \text{ K}^4$, T_{∞} is ambient temperature (25°C), ε is emissivity, i. e.

$$\varepsilon = \frac{T}{1000} \left(0.125 \frac{T}{1000} - 0.38 \right) + 1.1 \quad [3-4]$$

At the right side of the domain (i.e. S_2), $r = 6$ mm at all z positions (i.e., $z = 0$ to 6.65 mm) a semi-infinite solid is assumed such that thermal conduction is present as shown in equation [3-5].

$$-k \frac{\partial T}{\partial r} \Big|_{r=6mm} = q \quad [3-5]$$

At the centerline of the sample (i.e. S_4), $r = 0$ mm and $z = 5.65$ to 6.65 mm, an adiabatic condition exists due to symmetry.

$$-k \frac{\partial T}{\partial r} \bigg|_{r=0} = 0 \quad [3-6]$$

At the quenched surface (i.e. S_3), $z = 6.65$ mm $r = 0$ to 6 mm, heat flux is unknown.

The initial condition used in the model is shown in Equation [3-7] and is based on the assumption that the sample starts at a uniform temperature, T_i :

$$T(r, z) \big|_{t=0} = T_i \quad [3-7]$$

Along with, time-temperature data, the thermo-physical properties of the steel are provided as input to the IHC code. The HSLA steel undergoes phase changes, from austenite to ferrite and pearlite, during cooling process. It is beyond the scope of this study to go into investigations of phase transformation latent heat and its effect on material properties and was neglected.

The thermo-physical properties of the HSLA steel used for present study were taken as the values for low carbon steel, AISI 1008, as it is the closest composition with known properties ^[11].

Table 3-5: Thermo-physical properties of AISI 1008 steel ^[11]

Property	Value
Conductivity: k	$60.571 - 0.03849 \cdot T$ [W/m°C]
Density: ρ	7800 [kg / m ³]
Specific heat: c_p	470 [J / kg °C]

Equation 3-1 was solved subject to boundary conditions 3-2 to 3-6 and initial condition 3-7 using the FE method. For the FEM calculations, four-node linear rectangular elements were used. The mesh between the thermocouple tip and the top surface was refined to a size of 0.1 mm due to large temperature gradients in this region. The total number of elements in the domain was 559 and the total number of nodes was 614.

The IHC code is 2 dimensional hence it can be used to model the presence of a single thermocouple hole, but it can not be used to analyze multiple thermocouple holes across the strip width. This code can be used under steady state or transient heat transfer conditions. However, it does not take into account the heat generated inside the sample due to phase changes occurring during cooling.

The surface temperature contour plots were developed using a quasi-state approach. Knowledge of the surface thermal history at specific locations across the strip width in conjunction with the plate speed allowed calculation on the spatial distribution of temperatures for each pass.

The cooling of the test strip during the test was recorded using a digital video camera. The recordings were processed with a frequency of 30 Hz using Adobe Premier pro 1.5. The digital images were analyzed using a Matrox Inspector v4.0.

4. Results and Discussion

In the present study, novel experiments were done to investigate the heat transfer during multiple jet impingement on a top surface of a moving hot steel strip. Eight tests were carried out to investigate the effect of flow rate, strip speed and nozzle spacing on the heat transfer during the runout table cooling process. The details of the tests conducted are shown in Table 4-1 below:

Table 4-1: Test matrix

Test No.	Water flow rate (l/min.)	Strip speed (m/s)	Distance between 2 jets (mm)
1.	15	1	114.3
2.	15	0.75	76.2
3.	15	0.93	38.1
4.	30	1	114.3
5.	30	1	76.2
6.	30	1	38.1
7.	15	0.22	76.2
8.	30	0.22	76.2

In this chapter, the cooling curves i.e. the plots of internal temperature against time, and the calculated boiling curves are presented. The general trends observed during all tests are demonstrated and discussed using test # 4 as a typical example. While the visual observations and their resemblance with the calculated data analysis is demonstrated and discussed using test # 7. Finally, some comments on the effects of flow rate, strip speed and nozzle spacing are made.

4.1 Cooling Curves

The measured internal temperatures of the strip plotted as a function of time are raw data from the test representing cooling curves. Figure 4.1 shows the spatial distribution of the thermocouples across the strip width along with the top view of two nozzles spaced 114.3 mm apart. The locations 50.8 mm and 63.5 mm are specifically indicated in this Figure as these positions are selected for the cooling curves shown in Figure 4.2. The data given in this figure are taken from test # 4. Four typical cooling curves are shown, which consists of two below each of the nozzles and two at the intermediate positions in-between two nozzles (i.e interaction region).

Figure 4.1: Schematic top view of the thermocouple locations and two nozzles across the strip width

Figure 4.2: Typical measured data for 4 locations across the strip width for test # 4 (Centre-to-centre nozzle distance = 114.3 mm, water flow rate = 30 l/min and strip speed = 1 m/s)

As shown in Figure 4.2, when the water jets hit the strip above the thermocouple locations, the temperature at each location rapidly drops. The strip position where thermocouples are located immediately leaves the jets and due to the thermal mass of the strip (i.e. the parts of the strip which are not affected by the water jet are still at higher temperatures), the temperatures recover. This phenomenon occurs every time the jets hit the strip and is called a “pass”. Figure 4.3 shows a closer view of the first pass during the test. It can be clearly seen that the heat extraction is much more at locations directly below the nozzles as compared to those at locations in between nozzles (interaction region).

Figure 4.3: Measured thermal history during the first pass showing the difference in cooling between centre/side nozzle and interaction positions

Referring to Figure 4.2, the amount of cooling during a particular pass varies, as the strip is cooled and the temperature of the strip goes down. This is attributed to the different boiling heat transfer modes encountered (from film boiling at the start of the test to transition and nucleate boiling at lower temperatures) which depend on the surface temperature of the strip.

4.2 Heat Fluxes

Figure 4.4 shows the variation of the calculated heat fluxes with time for the thermocouple location below the centre nozzle. The heat fluxes start to drop from higher values during the initial passes (higher temperature); then increase up to a maximum value and start to decrease again. This variation of heat fluxes appears to be associated with the different heat transfer modes encountered during cooling.

Figure 4.4: Calculated heat flux below centre nozzle as a function of time for test # 4 (Centre-to-centre nozzle distance = 114.3 mm, water flow rate = 30 l/min and strip speed = 1 m/s)

Figure 4.5 shows the variation of these heat fluxes with time for a thermocouple location in the interaction region (i.e. at 50.8 mm from the centerline location). The heat fluxes at this location are low for the initial passes, which increase to maximum and then decrease again. Also, at times greater than 450 s the heat fluxes under the jet and in the interaction region are similar. It might be due to change of heat transfer mode to nucleate boiling, wherein the strip surface is completely wetted by water even in the interaction region. Overall the heat fluxes in the interaction region are much lower than below the nozzle (see Fig. 4.4) throughout the whole cooling process.

Figure 4.5: Calculated heat flux at interaction (50.8 mm) as a function of time for test # 4 (Centre-to-centre nozzle distance = 114.3 mm, water flow rate = 30 l/min and strip speed = 1 m/s)

Figure 4.6 shows the distribution of peak heat fluxes at different locations in the lateral direction (across the width) for pass 2 of test # 4. Similar trends are observed in most passes during a test and this figure is representative of those trends. It can be seen that high heat extraction takes place directly below the nozzles and adjacent to them due to direct impact of water. Lower heat extraction occurs the interaction region. In general the heat fluxes decrease from locations below nozzles to those far from them as the cooling zones change from impingement zone to parallel flow zone. In the current example, the minimum peak heat flux is observed at around 50.8 mm from the centerline.

Figure 4.6: Peak heat fluxes in lateral direction for pass # 2 of test # 4 (Centre-to-centre nozzle distance = 114.3 mm, water flow rate = 30 l/min and strip speed = 1 m/s)

4.3 Visual Observations

The cooling of the test strip during tests was recorded using a digital video camera and those recorded images from tests were observed carefully to develop an understanding of the cooling that occurred during a test. These visual observations were correlated with the calculated data from the internal temperature measurements during the test to understand the overall heat transfer on the surface of the strip.

When the strip was taken out of the furnace, it was bright red in color and at a temperature of about 850 – 900 °C. The temperature of the strip drops to about 800 °C, depending on other parameters of the test, before it reaches the water jets. When the water jets hit the strip, a small darkened zone can be observed at the impingement point below each nozzle. The color of the strip turned gray in the region next to the stagnation point as well as in the region where interaction of water takes place between two adjacent

nozzles. It could be due to the vapor layer that is formed when the jet hits the strip at such a high temperature. In the interaction region, the water flowing from the two adjacent jets interacts with each other and large splashing of the water is observed in this region. Outside this region, which is not affected by the water, the strip is still red hot in color indicating that there is no solid-liquid contact in this area and this area is still at a very high temperature. Considerable amount of water vapor was observed to come out from the strip surface.

Figures 4.7 to 4.12 shows the images captured from passes 2 through 7 for test # 7 respectively and the surface temperature (calculated using FEM IHC analysis) contour plots developed for each pass. The image from pass 1 was not included as it was almost identical to pass 2. The white circles shown in the strip images represent the TC locations in the lateral direction. Figure 4.13 shows the variation of the surface heat fluxes during each pass, at these TC locations in the lateral direction and Figure 4.14 shows the variation of surface temperature at the same locations in the lateral direction.

During pass 2, very small dark zones are observed below both side nozzles and the dark zone is even smaller below the centre nozzle. The sizes of these dark zones can be correlated with the size of the area indicated by the dark blue color (indicating surface temperatures below 300 °C) on the contour plot of pass 2. These dark zones are elliptical in shape rather than circular due to the movement of the strip. The area in between the nozzles and that just next to side nozzles, is gray in color and affected by water presence. The area far from the side nozzles can be seen to be red hot and these visual effects from strip image match well with the surface temperature contours.

In pass 3, the dark zones below all three nozzles can be seen expanded in area in the image as well as on the contour plots, indicating that the water front is progressing outwards from the stagnation line and more water solid contact is taking place. The other red hot areas on the strip surface change in color to blackish gray as the temperatures are dropping in those areas as well, though not significantly.

In pass 4, the dark areas expand further and the thermocouple locations next to centre as well as side nozzle is now, present on the boundary between dark and gray areas. Thus, the heat fluxes for this location (refer to Figure 4.13) rise to about 7 MW/m^2 from 5 MW/m^2 of pass 2, resulting in a higher drop of surface temperatures (refer to Figure 4.14).

In pass 5, the dark areas actually reach beyond the thermocouple location next to the nozzles and there is complete liquid-solid contact in these areas on the strip surface. This is reflected in heat fluxes as well which rise significantly to about 12 MW/m^2 from 7 MW/m^2 in the previous pass.

This tendency continues through pass 6, 7 and 8, the dark zones from 2 adjacent nozzles meet each other at the interaction region and uniform heat fluxes (about 3 MW/m^2 for pass 8) can be seen for all the thermocouple locations (refer to Figure 4.13), resulting in uniform surface temperatures.

The speed of progression of the black zones and their relative sizes depend on the testing parameters such as, water flow rate, strip speed etc, but the fundamental behavior stated here is consistent for all the tests.

Figure 4.7: Visual observations and calculated surface temperature contour plots during pass 2 of test # 7

Figure 4.8: Visual observations and calculated surface temperature contour plots during pass 3 of test # 7

Figure 4.9: Visual observations and calculated surface temperature contour plots during pass 4 of test # 7

Figure 4.10: Visual observations and calculated surface temperature contour plots during pass 5 of test # 7

Figure 4.11: Visual observations and calculated surface temperature contour plots during pass 6 of test # 7

Figure 4.12: Visual observations and calculated surface temperature contour plots during pass 7 of test # 7

Figure 4.13: Peak heat fluxes in lateral direction for different passes of test # 7 (Centre-to-centre nozzle distance = 76.2 mm, water flow rate = 15 l/min and strip speed = 0.22 m/s)

Figure 4.14: Variation of surface temperatures in lateral direction for different passes of test # 7 (Centre-to-centre nozzle distance = 76.2 mm, water flow rate = 15 l/min and strip speed = 0.22 m/s)

4.4 Boiling Curves

The boiling curves are the plots representing the variation of the heat flux with wall superheat (the difference between surface temperature and water saturation temperature) or surface temperature. Figure 4.15 shows some representative boiling curves from three different locations for test # 4, using heat fluxes and surface temperatures calculated from the inverse heat conduction model. The heat fluxes shown in these plots are the peak heat fluxes for a particular pass and the corresponding surface temperature is the entry surface temperature for that pass, i.e. the temperature of the strip at that location, when the water jets first hit the strip during that pass. The locations represented in these curves are the ones directly below the centre and side nozzle and the position in between the two nozzles (the interaction region) which was seen to have minimum heat fluxes.

It can be seen from Figure 4.15 that the boiling curves calculated at locations directly below the nozzles are quite different from the boiling curve at the interaction region. The boiling curves below each nozzle are similar to each other and clearly show the different boiling regimes, while the boiling curve for the interaction region does not show these regimes. In the interaction region, heat transfer remains relatively low until the water completely wets the strip (at a temperature of about 200 °C) and the boiling curve is quite similar beyond that point.

Figure 4.15: Typical boiling curves for 3 locations across the strip width for test # 4 (Centre-to-centre nozzle distance = 114.3 mm, water flow rate = 30 l/min and strip speed = 1 m/s)

The boiling curves for locations below the centre and side nozzles show that the film boiling regime (determined by a decrease of the heat flux with decreasing surface temperature) is observed. The heat flux decreases with decreasing surface temperature until it reaches a point of minimum heat flux at an entry temperature of about 640 °C. The heat flux starts increasing with decreasing surface temperature from this point onwards, indicating the transition boiling regime. The heat flux increases with decreasing surface temperature of the strip till the maximum heat flux point is reached at an entry temperature of about 350 °C. Beyond the maximum heat flux point, the heat flux again decreases with decreasing surface temperature of the strip indicating that the heat transfer mechanism in this region is nucleate boiling. Finally, at an entry temperature just below 200 °C, the strip is completely wetted by water and the centre, side and interaction regions all exhibit similar heat transfer. Further, at around 100 °C, the heat transfer mechanism changes from nucleate boiling to single-phase convection.

The boiling curve at the interaction region indicates that the regime with significantly lower heat fluxes is extended down to about 290 °C. This expanded regime could be a result of two factors. First, the water is not directly impinging from nozzles in this region so the effect of direct impact of the water jet is not there, rather this zone falls into the parallel flow zones of the two adjoining jets. Thus, although the strip surface in this region is covered by water, it is not wetted by water i. e. there is partial or full vapor film between the water and the strip surface. Second, there is strong interaction between water from the two adjoining jets in this region and interaction fountains are observed at these locations, where the water splashes out at the interaction. This agitating water can neither form a stable vapor film on the hot strip surface nor is this film intermittently broken by the impact of water from a jet. Thus, stable film boiling or transition boiling can not be observed in this region. Also the amount of water affecting this region is reduced by the splash at interaction. The combined effect of all these factors in the interaction region is the reason for this extended region of very low heat fluxes. This extended region of low heat fluxes can be associated with the gray area observed in interaction region before black zones from two adjacent jets meet each other.

The heat fluxes start to increase with decreasing surface temperature beyond 290 °C. This effect can be explained with the help of visual observations on the surface of the strip. At surface temperatures, between 250-300 °C for the present test under discussion, the wetted zones of the adjoining jets are seen to start to come together and overlap each other. With a further decrease in entry temperature, the water completely covers the area in-between the two jets. Thus, the water comes in direct contact with the strip surface after the entry temperature decreases below about 290 °C, changing the heat transfer

mechanism to nucleate boiling or transition boiling and leading to an increase in heat fluxes till it reaches a maximum heat flux at about 170 °C for this boiling curve. The heat fluxes again start to decrease beyond the maximum heat flux with decreasing surface temperatures and the heat transfer mechanisms could be single-phase forced convection through nucleate boiling.

In general, the heat fluxes for locations in the interaction region are much less than that for locations below nozzles along all the regimes of the boiling curves above 200°C. This is mainly due to the fact that in this region strong water splash takes place due to interaction between water jets from adjoining nozzles and this region falls in the parallel flow zone between jets. The extent of interaction and its effect on heat transfer can be expected to vary depending on the important parameters such as nozzle spacing, strip speed and water flow rate.

4.5 Effect of Strip Speed

Two different tests were carried out by changing the speed of the strip from 0.22 m/s and 0.75 m/s. In this section, we will discuss the effect of strip speed on the heat flux at different locations in lateral direction and the entry temperature as the strip is cooled down.

In Figure 4.16, the peak heat fluxes at different locations in the lateral direction are shown for the pass at a start temperature of about 600 °C, for the test where the centre-to-centre distance between two adjacent nozzles was 76.2 mm and the water flow rate was constant at 15 l/min. As stated earlier, the peak heat fluxes at the locations below nozzles are much higher as compared to locations in between and the lowest heat

fluxes can be seen at 38.1 mm which is the mid point between both jets and interaction occurs there.

Figure 4.16: Peak heat fluxes in lateral direction for varying strip speed at $\sim 600^\circ\text{C}$ (Centre-to-centre nozzle distance = 76.2 mm, water flow rate = 15 l/min)

At all the locations, the peak heat fluxes are higher for the test with lower speed and the effect is similar both under the nozzles and in the interaction region. This clearly indicates that the heat fluxes increase with decreasing strip speed at locations below and in between two nozzles.

In Figure 4.17, peak heat fluxes are shown as a function of entry temperature for a location below the centre nozzle. It can be seen that the heat fluxes are higher for lower strip speeds. The maximum heat flux of around 16 MW/m^2 can be seen for slower plate speeds, which is consistent with the observations of Chen et al. [11] [36] [37] [38] in the literature. The maximum heat flux point shifts to lower entry temperatures for the strip moving at higher speed and the effect of strip speed is most significant in the region of

maximum heat flux. The effect of strip speed is not significant in the nucleate boiling regime and its extent diminishes at the low entry temperatures. When the strip is moving with slower speed, water spends more time on the surface of the strip and the resulting liquid solid contact can be established even at comparatively high temperatures, which leads to shifting of maximum heat flux point to higher entry temperatures for slower moving strips. When the strip entry temperature goes down beyond point of maximum heat flux the liquid-solid contact area increases and the heat transfer mechanism changes to nucleate boiling from transition boiling, which means there is more and more liquid-solid contact irrespective of the speed of the strip and the time strip spends below water and this leads to a minimal effect of strip speed on heat flux at lower entry temperatures.

Figure 4.17: Typical boiling curves for location below centre nozzle for varying strip speed (Centre-to-centre nozzle distance = 76.2 mm, water flow rate = 15 l/min)

In Figure 4.18, peak heat fluxes are shown as a function of entry temperature for a location in the interaction region. Similar effects as observed from Figure 4.17 can be seen in the interaction region, with the difference that the heat fluxes are much lower.

Figure 4.18: Typical boiling curves for location in interaction region (50.8 mm) for varying strip speed (Centre-to-centre nozzle distance = 76.2 mm, water flow rate = 15 l/min)

4.6 Effect of Water Flow Rate

Two flow rates i.e. 15 l/min and 30 l/min were used in the present study to investigate the effect of water flow rate. In this section, we will discuss the effect of water flow rate on the heat flux at different locations in lateral direction and the effect of the same with the surface temperature as the strip is cooled down in various passes.

In Figure 4.19, the peak heat fluxes at different locations in the lateral direction are shown for the pass at the start temperature of about 600 °C, for the test where the centre-to-centre distance between two adjacent nozzles was 76.2 mm and the speed of the strip was constant at 0.22 m/s. As stated earlier, the peak heat fluxes at the locations

below nozzles are much higher as compared to locations in between. Although the peak heat fluxes at locations close to nozzles are significantly higher than those at interaction for higher flow rate, which might be due to the fact that the diameter of impinging water jet with higher flow rate is larger leading to a bigger dark zone than that formed by lower water flow rate jet. In general, the increase in flow rate increases heat fluxes at all measuring locations due to the higher amount of water impinged on the strip surface.

Figure 4.19: Peak heat fluxes in lateral direction for varying flow rate at $\sim 600^\circ\text{C}$ (Centre-to-centre nozzle distance = 76.2 mm, water flow rate = 0.22 m/s)

In Figure 4.20, the peak heat fluxes are plotted against the entry temperatures for the measuring location below the centre nozzle. The effect of water flow rate is prominent in the maximum heat flux area where the higher water flow rate leads to higher heat fluxes. These effects are not that significant in the nucleate boiling region.

Figure 4.20: Typical boiling curves for location below centre nozzle for varying water flow rate (Centre-to-centre nozzle distance = 76.2 mm, Strip speed = 0.22 m/s)

Figure 4.21: Typical boiling curves for location at interaction (38.1 mm) for varying water flow rate (Centre-to-centre nozzle distance = 76.2 mm, Strip speed = 0.22 m/s)

In Figure 4.21, the peak heat fluxes are plotted against the entry temperatures for the measuring location in the interaction region. It can be seen that the maximum heat flux point is shifted to higher temperatures with rise in the water flow rate and higher water flow rate leads to higher heat fluxes in maximum heat flux region and beyond. The effect of water flow rate is insignificant in the nucleate boiling region.

4.7 Effect of Nozzle Configuration

The distance between two adjacent nozzles was varied by keeping the water flow rate of 30 l/min and strip speed of 1 m/s constant. Three different tests were carried out with the nozzles at 38.1 mm, 76.2 mm and 114.3 mm apart from each other. An average heat flux across the width of the strip was determined by summing the calculated heat fluxes from centre to side and dividing by the distance between the two nozzles. This quantity gives the average amount of heat extracted along the length between two nozzles indicating the efficiency of that nozzle configuration which can be compared with other configurations. The variation of average heat flux per mm with entry temperature for each pass is plotted in Figure 4.22 for three different nozzle configurations.

As seen in Figure 4.22, the nozzle configuration having nozzles at 38.1 mm apart, is more efficient than the other two configurations indicating that, having nozzles close to each other enhances heat transfer. The difference in heat extraction is particularly pronounced in the region around the maximum average heat flux per mm which falls between 200 °C to 400 °C. While for the nozzle configuration of 76.2 mm and 114.3 mm, the difference in heat extraction is not significant.

Figure 4.22: The variation of average heat flux per mm with entry temperature for three different nozzle configurations (Water Flow Rate = 30 l/min, Strip speed = 1 m/s)

To explain the reasons behind the efficient heat extraction for the nozzle configuration of 38.1 mm, the peak heat fluxes below the centre and side nozzle and from a location in the interaction region were plotted against the normalized distance between nozzles for each nozzle configuration (Figure 4.23 and 4.24).

It can be seen from Figure 4.23 that, at 460 °C, the heat fluxes at interaction are less than those for the locations below nozzles. Also the difference between below nozzles and at interaction is less for the configuration of 38.1 mm in comparison to other two configurations.

It can be seen from Figure 4.24 that, at 280 °C, the heat fluxes in the interaction region are less than those for the locations below nozzles for the configurations of 76.2 mm and 114.3 mm but there is no difference between heat fluxes at interaction and those

below nozzles for the configuration of 38.1 mm in comparison to other two configurations.

Figure 4.23: Comparison of heat fluxes for three nozzle configurations at 460 °C, along the normalized distance (water flow rate of 30 l/min and strip speed of 1 m/s)

Figure 4.24: Comparison of heat fluxes for three nozzle configurations at 280 °C, along the normalized distance (water flow rate of 30 l/min and strip speed of 1 m/s)

Figure 4.25 and 4.26 show the plots of peak heat flux against entry temperature for the location below centre nozzle and that at interaction, respectively, for the three different nozzle configurations.

It can be seen from Figure 4.25 that heat extraction below centre nozzles is very similar for all the three nozzle configurations.

Figure 4.25: Typical boiling curves for location below centre nozzle for varying nozzle configurations (Water flow rate = 30 l/min, Strip speed = 1 m/s)

Figure 4.26 shows that for the location in the interaction region the configuration of 38.1 mm shows the highest heat extraction in comparison to the other two configurations. This is a consequence that the black zones formed around two neighboring jets have less distance to travel before they meet each other which results in higher heat extraction even at interaction. While for the other two configurations, where the nozzles are further apart from each other, the black zones will reach each other later leading to lower heat extraction.

Figure 4.26: Typical boiling curves for location in interaction region for varying nozzle configurations (Water flow rate = 30 l/min, Strip speed = 1 m/s)

From Figures 4.23 to 4.26, it is clear that for all the configurations, locations below nozzles experiences very high heat fluxes but it is only for the configuration of 38.1 mm that even the locations in the interaction region experience high heat fluxes, resulting in the higher average heat flux per mm values for this configuration.

5. Summary and Conclusions

5.1 Summary

A novel experimental study was carried out to investigate the heat transfer during multiple jet impingement on a top surface of a moving hot steel strip using the pilot scale ROT facility at UBC. The purpose of the study was to understand the effect of varying nozzle-to-nozzle distance, strip speed and flow rate of impinging water on the heat transfer taking place during ROT cooling. Experiments were performed on a pilot scale runout table using test strips moving at speeds of 0.22 m/s and 1 m/s. One jetline having three circular jets, was used in each experiment. The temperature of the cooling water was kept constant at 30°C for all experiments and the water flow rate was varied from 15 l/min. to 30 l/min. The nozzle spacings of 114.3 mm, 76.2 mm and 38.1 mm were used. The effect of strip speed, water flow rate and nozzle configuration on cooling curves and heat fluxes were examined. These experimental results provide important information for the development of improved runout table cooling models.

During the experiments, the internal temperature change in instrumented steel samples was measured at various locations. The internal time-temperature data obtained from the test was processed using an existing inverse heat conduction model developed at UBC. The model used two dimensional, axi-symmetric finite element method to solve the transient heat conduction problem. Measured internal temperatures were processed in the inverse heat conduction model and the surface temperatures and surface heat fluxes were obtained.

The video and still images obtained during the test were analyzed using Matrox Inspector 4.0 software for image analysis and compared with the surface temperature profiles obtained from the IHC model to predict the overall heat transfer phenomenon taking place during multiple jet impingement cooling on the surface of the moving strip.

Average heat flux per unit length of the strip in lateral direction was used to compare overall cooling effectiveness of each nozzle configuration and optimum nozzle configuration is suggested for the given set of parameters.

5.2 Conclusions

Following conclusions can be drawn from the present study:

- During multiple jet cooling, high heat extraction takes place directly below the nozzles and adjacent to them due to the direct impact of water. Lower heat extraction occurs at locations between the nozzles, i.e. in the interaction region. This is mainly due to the fact that limited water is available and momentum of the water is lower in this region.
- Visual observations of the tests indicate that, when the water jets hit the strip, a small darkened zone can be observed at the impingement point below each nozzle. These dark zones are elliptical in shape and expand with cooling of the strip, indicating that more water solid contact is taking place. At one point of time, the dark zones from the two adjacent nozzles meet each other in the interaction region and uniform cooling is achieved along the strip width. The boiling curves below each nozzle are similar to each other and clearly show the different boiling regimes while the boiling curve for the interaction region does not show these regimes as clearly. In the interaction region, heat transfer remains relatively low until the water completely wets the strip.

- The investigation of effect of the strip speed indicated that the heat fluxes are higher for lower strip speeds, below the nozzles as well as in the interaction region. The maximum heat flux point shifts to lower entry temperatures for the strip moving at faster speed and the effect of strip speed is most significant in the region of maximum heat flux. The effect of strip speed is not significant in the nucleate boiling regime and its extent diminishes at the low entry temperatures.
- An increase in water flow rate increases heat fluxes at all measuring locations due to higher amount of water impinged on the strip surface. This effect is seen both below the nozzles as well as in the interaction region. The effect of water flow rate is most prominent in the maximum heat flux area where the higher water flow rate leads to higher heat fluxes. The effect of water flow rate in the nucleate boiling region is minimal.
- The nozzle configuration having nozzles 38.1 mm apart is more efficient than the other two configurations examined indicating that, having nozzles close to each other enhances heat transfer. For all the configurations, locations below nozzles experience very high heat fluxes but it is only for the configuration of 38.1 mm that even the locations in the interaction region experience high heat fluxes, resulting in the higher average heat flux per mm values for this configuration. Thus, decreasing the nozzle distance leads to rapid overlapping of dark zones and hence much more efficient and homogeneous cooling across the width.
- Control of the heat transfer occurring in the interaction region is critical for ROT cooling. Effective cooling of the interaction region leads to increase in overall effectiveness of cooling as well as more uniform cooling across the width of the strip.

- Optimum spacing of the nozzles in a jet-line is related to the strip speed as well as the water flow rate and water available for cooling.

5.3 Recommendations for Future Work

Considering the complexity of the jet impingement cooling on the ROT, further work is mandatory to improve our knowledge of this topic. Suggestions for future work are:

1. The black zones and their progression need to be quantified.
2. The effect of strip speed needs to be investigated in more detail with the aim to find out the correlation between change in speed and change in corresponding heat fluxes.
3. The effect of nozzle configuration should be explored with more sets of nozzle spacing and changing flow rates.
4. The hydrodynamics taking place on the top surface of the strip during multiple jet impingement need to be looked into.
5. The existing IHC code should be upgraded to account for heat generation due to phase transformation and the code should be expanded to account for heat transfer in three dimensions.

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