Experimental Investigation of High-temperature Steel Plate Cooled by Multiple Nozzle Arrays

Qian XIE, Bingxing WANG,* Yu WANG, Zhaodong WANG and Guodong WANG

The State Key Laboratory Rolling and Automation (RAL), Northeastern University, Shenyang, 110819 China.

(Received on December 7, 2015; accepted on March 24, 2016; J-STAGE Advance published date: May 26, 2016)

Thermo-mechanical controlled process (TMCP) technology has been widely used to improve controlled cooling technology during the hot rolled plate manufacturing process. Multiple impinging jets are the main cooling form used in plate cooling process of the steel mills. In this work, we mainly focused on the surface flow field and heat transfer of high-temperature plate (700°C) cooled by nine-nozzle arrays with different Reynolds numbers. The distribution of turbulent kinetic energy and water velocity vector were numerically studied. By analyzing the simulation results and measuring temperature data, the maximum heat flux, the corresponding surface temperature and time at each measurement point were found to be dependent on the distance from the impinging point and the jet velocity. No obvious maximum heat flux elevation for the locations between jets was observed, where the wetting speed was also slow with Reynolds numbers lower than 8 278. These results are valuable in interpreting the heat transfer mechanism under jet arrays and predicting the cooling rate, and forecasting the microstructure of the steel product.

KEY WORDS: multiple impinging jets; nozzle arrays; heat transfer; flow structure; steel plate cooling.

1. Introduction

The hot rolled steel plate has been extensively employed in the field of construction engineering, machinery manufacturing, container manufacturing and shipbuilding. During the plate manufacturing process, the thermo-mechanical controlled process (TMCP) technology can effectively improve plate mechanical properties without adding extra elements. One important part of TMCP technology is to let the plate rapid cooling to a certain temperature at an ultra-fast cooling rate, which will have a strong effect on the microstructure and phase distribution of the plate. Although, laminar cooling has been the most common cooling method in commercial steel mills in nowadays. The disadvantages of laminar cooling still need to be fixed, such as the laminar cooling water accumulates on the plate, the surface flow field interferes cooling capacity and uniformity, most of the cooling water slides and falls from the plate instead of directly contacting the plate for the presence of Leidenfrost phenomenon.^{1,2)}

The cooling regions in impinging cooling are classified as the wetted region and the unwetted region. At the impinging point, the temperature drops quickly, and the surface appears to be a dark circle, which keeps expanding as the cooling goes on. The maximum heat flux is believed to occur at the position a little behind the wetting front, where the transition from transient boiling to nucleate boiling happens.³⁾ Nitin Karwa and his coworkers^{4,5)} mainly focused on

the flow patterns and the heat transfer distribution during the cooling process by impinging on the stainless cylinder specimen of 900°C with different jet velocities, and their conclusions about heat flux distribution and surface flow field were valuable. Meanwhile, their results also revealed that the effect of jet velocity on heat transfer at stagnation region was comparably low.

The maximum heat flux of a single jet is attained in the stagnation region, and the heat flux decreases rapidly with the increasing distance from the impinging point. In order to achieve uniform cooling in industrial application, multiple jets have been widely used. Tie Peng⁶⁾ studied submerged cooling with jet arrays; they wanted to cool down the overheated CPU chips from 75.7°C, more efficiently. In this case, 7 mm nozzle spacing, and 5.0 mm jet-to-target distance were set by optimizing the cooling parameters. Many studies^{7–9)} had been focused on the flow field and heat transfer of multiple air jet arrays, of which the pressure and velocity distribution characteristics are similar to the water cooling condition. The flow pattern includes stagnation region, decelerated flow, recirculating flow and vortex. Kwon and his coworkers^{10,11)} numerically studied the cooling process of a steel plate in a run out table (ROT). The thickness of residual water increased with flow rate. The cooling efficiency decreased with the density of the nozzle arrangement, and the average heat flux did not increase monotonously with the increased flow rate. Therefore, a proper flow density was required to achieve the optimum cooling effect.

Flows of multiple jets are complicated, they interact with each other, and the boiling and heat transfer states

* Corresponding author: E-mail: wbxang@126.com

DOI: <http://dx.doi.org/10.2355/isijinternational.ISIJINT-2015-691>

are affected by the decelerated flow and recirculating flow. Moreover, the cooling performance of multiple jets impingements on the high-temperature steel plate has been rarely reported. In this work, we aimed at extending the understanding of heat transfer features on high-temperature steel plate during the cooling process of multiple jet arrays. Based on the flow field features, the experiments of transient cooling were carried out by cooling 700°C stainless steel with multiple nozzles arrays to analyze the temperature data at different locations.

2. Experimental Set-up and Procedure

2.1. Measurement Apparatuses

The experiment was carried out on the specially designed cooling platform as shown in **Fig. 1**. The water flow rate and air pressure were set to the target value before the cooling. The specimen was heated to 700°C by two ceramic heating panels. Then it was transferred to the trestle table, which was above the water tank. The heat losses during the transportation were ~25°C.

AISI 304 stainless steel specimen was chosen for its stable austenitic structure and the excellent oxidation resistance at high temperature. As shown in **Fig. 2**, the size of the specimen is 150×80×20 mm (Length, Width, Thickness), nine K-type sheathed thermocouples with a temperature range of 0–900°C of 3 mm diameter were customized to obtain the temperature data. The arrangement of thermocouples was aimed to get the temperature data at the center line of the impinging region. The measurement point 9 was at the center impinging point and the measurement

point 4 is at the outer impinging point. The temperature data in the area between the jets and outside the jets were recorded simultaneously.

The accurate temperature detection position located at the head of the thermocouple. The measuring component inside the thermocouple was attached to the sheath to make sure a fast temperature response. The distance between the center of the hole and the impinging surface was 2.5 mm, with the inserted depth of 30 mm. In order to keep the thermocouples contact closely with the plate, the paste of high temperature thermal conductive ($\lambda = 9.1 \text{ W} \cdot \text{m}^{-1} \text{ K}^{-1}$) was utilized to fulfill the inside of holes. The temperature data were recorded by a moveable dependent device (midi LOGGER GL220) at a frequency of 10 Hz. Before every experiment, the steel was polished to eliminate the influence of surface roughness, and its surface roughness was about 0.4 μm .

2.2. Nozzle Structure

Velocity uniformity of jet arrays is an important factor in the study. As shown in **Fig. 3**, to guarantee the good velocity uniformity, a flow-equalizing device was designed and built up from the polylactic acid by 3D Printers (Wiiibox) technology. The machining accuracy is 0.1 mm. The diameter of entrances is 24.4 mm with M35 screw thread; A clapboard achieves damping effectiveness with four 10 mm holes, so inlet water cannot directly impinge on the bottom position. The lower chamber can maintain a uniform pressure distribution, which is helpful to obtain the same jet velocity at the outlet of each nozzle. The length, diameter, and interval of nozzles are 20, 3, and 25 mm, respectively. The deviation of outlet jet speed is within 5%.

Fig. 1. The flow diagram of the specially designed cooling platform.

Fig. 2. The schematic of plate with distribution of thermocouples and jet arrays.

Fig. 3. The structure and digital graph of the flow-equalizing device.

2.3. Numerical Model

The calculation domain and the associated boundary conditions of the cooling process are presented in Fig. 4. The purpose of the numerical study focused on the flow field under multiple jets. Considering the lack of the reliable boundary data and fundamental mechanisms about the high-temperature cooling process, the numerical study of heat transfer process was not implemented here.

The numerical calculation is based on the ANSYS CFX software (version 15.0). The volume of fluid (VOF) model is adopted to calculate the steady states flow field, and the reliable k - ϵ model is used to calculate the turbulence. Each nozzle entrance is set to the velocity input condition with water fraction of 1, which means the nozzle is full of water. Non-nozzle portions of the top surface are set to the opening boundary conditions without the water back flow. The average grid size in the calculation domain is 1.5 mm, and the grids near cooling surface are refined to 0.5 mm. The convection and diffusion terms of the governing equation are discretized by using the second-order upwind scheme. The simple algorithm is employed to calculate the pressure.¹²⁾

2.4. Data Reduction

During the impinging cooling process, standard temperature measurement techniques such as infrared thermography cannot be used for the disturbance of surface-water and vapor. As a result, the surface temperature and heat flux are obtained by inverse heat transfer method based on thermal conduction differential equation and temperature curves readings from the inside of the plate.^{13,14)}

The two-dimension unsteady thermal conduction differential equation is written as:

$$\frac{\partial T}{\partial \tau} = \frac{\lambda(T)}{C(T)\rho} \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \dots \dots \dots (1)$$

Fig. 4. The calculation domain and position of the visualization planes.

Where T is temperature, t is time, τ is the time interval, λ is heat conductivity, C is specific heat, ρ is density. The third boundary condition is adopted, and the initial and boundary conditions are written as follows:¹⁵⁾

$$T(x, y, 0) = \phi(T), \left(t = 0, 0 \leq x \leq \frac{Tk}{2}, 0 \leq y \leq \frac{B}{2} \right) \dots (2)$$

$$-\lambda(t_{Tk}) \frac{\partial T}{\partial x} = q_x, \left(x = \frac{Tk}{2}, t > 0 \right) \dots \dots \dots (3)$$

$$-\lambda(t_B) \frac{\partial T}{\partial y} = q_y, \left(y = \frac{B}{2}, t > 0 \right) \dots \dots \dots (4)$$

$$\frac{\partial T}{\partial x} = \frac{\partial T}{\partial y} = 0, (t > 0, x = 0, y = 0) \dots \dots \dots (5)$$

Where, Tk is thickness of the plate, B is width, ϕ is heat capacity of steel plate. The Crank-Nicolson difference method is employed for thickness and width direction, the computing time step is 0.1 s, the grid size is 1 mm². The strand diameter of thermocouples is 3 mm and the temperature responsiveness is less than 0.1 s. In each time step, the difference between the measured and calculated values is used as the criterion for a new heat flux. The calculation was implemented in commercial software MATLAB. The detail of the calculation can be found in Wang's work.¹⁶⁾

The local convective heat transfer coefficient, h , between the jet flow and plate surface, is defined as follows:

$$h = \frac{q}{(T_s - T_w)} \dots \dots \dots (6)$$

The major uncertainty of temperature fluctuations can be ascribed to the positional deviation of thermocouples. Therefore, all the measurements were repeated twice, the maximum uncertainty of the average data was about 5%. The uncertainty of steel thermal properties was assumed to within 10% in the temperature range of 200–700°C. The water temperature was controlled within $\pm 0.5^\circ\text{C}$ at the point of cooling. The jet velocity was controlled within a tolerance of ± 0.2 m/s. The experimental parameters are listed as follow: the plate initial temperature is 700°C, $\Delta T_{sub} = 82^\circ\text{C}$, $D = 3$ mm, $H = 90$ mm, V is 1.9, 2.6, and 3.9 m/s and the corresponding Re is 6 367, 8 278, and 13 087, respectively.

3. Results and Discussion

3.1. Analysis of Flow States

As indicated in Fig. 5(a), at the beginning of the impingement (0.034 s after the water impingement on the surface), the small black circle zone appearing in the stagnant zone is called wetted region, which indicates that the cooling water contacts the surface directly.¹⁷⁾ With increasing boiling frequency, the very high heat flux is achieved within a short period with a significant drop in surface temperature. The sputtered water drops collided with each other, and no obvious boiling bubbles were observed.

Figures 5(b)–5(c) shows the expanding of the wetted region. The dark zone spread radially as soon as the cooling began, and the intermediate region between jets did not wetted yet. The volume of accumulated water on the plate increased with time. The sputtered water drops started to

Fig. 5. Digital graphs of the variation of flow fields and the growth of wetted region in experiment C2 after water impinging on the plate, (a) 0.034 s, (b) 0.34 s, (c) 0.67 s, (d) 1.00 s, (e) 2.00 s and (f) 3.00 s.

merge into a flow between the jets and drained off through the gaps among jets.

In Figs. 5(d)–5(f), the wetted regions kept expanding and connected with each other at the time of 1 s. The accumulated water maintained a relatively stable status and corresponding with the expansion of the wetted region, where the water did not slip on the vapor film at a fast speed anymore. Abundant bubbles, which derived from the boiling on the plate surface and the doped air during the interaction of upwash flow, were mixed into the accumulated water.

The simulated flow states are shown in **Figs. 6(a)–6(b)**, where the thin flow layers were distributed around each impinging point. With high horizontal velocity, the thin flow layers collided with each other and formed the upwash flows, which located at the interval of the impinging points. The kinetic energy of upwash flow decreased with height, and the speed vector at the top layer flow reached the minimum.

The position of visualization plane is shown in Fig. 4. The k and projection of the velocity vector for the vertical plane 1 at the center jet and its neighbors are presented in **Figs. 7(a) and 7(c)**. The k was relatively high in the vertical region that closed to the jets. The height of upwash water, determined by the collisions among the adjacent jet flows, can reach ~50 mm. Part of upwash water entered into the adjacent jet regions and re-impinged on the plate surface. The other part of upwash water was drained by flowing to the adjacent and ambient jets. Moreover, Matsumoto¹⁸⁾ had reported both of the drained mechanisms. In the gaps between jets, the formed vortices in accumulated water, which resulted from the interaction of the upwash flow and falling water, showed higher k value. In Fig. 7(b), the vortex

Fig. 6. The distribution of velocity vector in the volume that the water fraction is above 0.05, $Re=8\,278$, (a) standard view, (b) top view.

between adjacent jets can be observed, the upwash flow was closer to the outside jets. The height of accumulated water decreased along the outward direction. The region with higher k was corresponding with the collision of parallel flow at the normal direction of plane 2. At the horizontal plane exhibited in Fig. 7(d), the higher k region was distributed at the peripheral area of the jets, where the overflowing water fell and collided with the parallel flow.

3.2. Analysis of Surface Temperature Curves

For the disturbance of surface-water and vapor on the surface of the plate, the surface temperature was evaluated by the inverse heat conduction method. **Figure 8** shows the corresponding transient surface temperature curves. In the period before the cooling start, the temperature on the plate was uniform, which indicated the proper heating process. The cooling curves herein were similar to the previous

Fig. 7. The distribution of k and velocity vector of each plane, (a) plane 1, (b) plane 2, (c) plane 3, (d) plane 4, horizontal plane at height 3 mm.

reports about the single impinging cooling.^{19,20)} Since the jet impinged on the surface (0.0 s), a sharp temperature decrease was observed at the stagnant points in the wetted states regardless of the variation of Reynolds number (Re), as shown in Figs. 8(a)–8(c). The wetting process initiated in 0.034 s after the beginning of impingement, as shown in Fig. 5(a). The Fig. 5 indicated that it took a certain time for the wetted region to reach the periphery of the stagnation points. When the wetting front arrived at a particular spatial location on the surface, a sharp drop in surface temperature can be observed.^{5,21)} The surface cooling rate at different spatial locations varied with the jet Reynolds number.

At Re of 13 087, it took 5.8 s for the surface temperature cooling down from 650°C to 150°C at point 1, which was 15 mm away from the point 4. This time was nearly 2 times higher than the time interval at the stagnation point 4 (3.0 s). With the decrease of Re , the time interval at the stagnation point did not change prominently, just increased to 3.9 s ($Re=6\ 367$). However, the difference between surface temperature curves can be discovered at the point 1, 6, 7, which were away from the impinging points. With the decrease

of Re , the temperature dropped at a slow speed, which can be accounted for the less volume and momentum of the splashed water, as well as the decreased wetting speed. Meanwhile, these areas kept unwetted for a longer time and film boiling was the dominant form of heat transfer.

Figure 9 presents the variation of surface temperature with cooling time for the different Reynolds numbers. With Reynolds number increasing from 6 367 to 13 087, the surface cooling rate was further enhanced, particularly in the radial locations which were away from the stagnation point. At 0.5 s, the temperature at impinging points showed an extreme decrease, and the moment when temperature curves begin decrease was related to the expanding of the wetted region. The lowest temperature region located at each impinging point were found to increase with the distance from the impinging point. At 1.0 s, the temperature at point 6 and 7 were lower than at point 2, which proved the wetting speed of the parallel flow collision region was slightly faster than the relative outer region, where the distances between measurement points 2, 6, 7, and the nearest impinging points are 10 mm. However, the lowered temperature at

Fig. 8. The variation of surface temperature curves with cooling time and Re.

Fig. 9. The variation of surface temperature of entire cooling area with cooling time and Re.

point 6 than at point 7 in all the experimental cases can be attributed to the upwash flow was much closer to point 6. The vortex at point 6 with higher k can effectively take away the bubbles that attached to the surface. This behavior was helpful to increase boiling frequency and heat extraction quantity, and the lower surface temperature was the critical factor that drive the movement of wetting front.¹⁷⁾ Additionally, more water was accumulated at point 6 than point 7, and the increase of water volume was also beneficial for the acceleration of the wetting process.²²⁾

The temperature difference between measurement points reached the maximum at 1 s after the cooling start in all the experimental conditions. At Re of 13 087, the temperature difference between point 7 and 9 was 280.8°C and then drastically dropped to 95.1°C after another 1 s cooling. It can conclude that the heat transfer efficiency at a certain position decreased with extending the impinging cooling time.

3.3. Analysis of Heat Transfer Coefficients

The heat transfer coefficient (HTC, h) curves at various locations are shown in Fig. 10. The HTC behaved nonlinearly as a function of surface temperature. The different cooling trends were presented at various places. Figure 10(a) shows the curves at impinging points, where the HTC increased shapely when the temperature dropped from 650°C to 500°C. With further decreasing the temperature, the slope of HTC curve flattened gradually. After reaching the maximum of HTC around 150–180°C, the HTC decreased sharply, and the heat transfer mechanism eventually changed into the forced convection heat transfer at low surface temperatures, which was confirmed in Nallathambi's study.²³⁾ However, at point 9, the maximum of HTC ranged from 13 275.8 to 14 232.1 with different Reynolds numbers.

The HTC curves at locations that are 5 mm away from impinging points are shown in Fig. 10(b), these curves were similar to the curves at impinging points, but both of the HTC increasing speed and the maximum HTC were lower than those at the impinging point. At Re of 13 087, the HTC of point 8 at 500°C was ~30% lower than it at point 9. The maximum of HTC was in the range 200–220°C, which was slightly higher than the corresponding temperature at point 9.

The measurement points of 6, 7, and 2 are 10 mm to the nearest impinging points, and the corresponding HTC curves are shown in Fig. 10(c). The increase of HTC were sluggish at the initial cooling process (temperature above 600°C) for the film boiling dominated the heat transfer. Although the cooling water accumulated at the position between the jets, the surface kept unwetted until the wetting front expanded to here from the impinging position. The points 6 and 7 located at the interval of the jets while the point 2 located at the outside of jets. Therefore, their temperatures along with the maximum HTCs behaved differently. The temperature of the maximum HTC at the inner location was about 300°C, while at the outer location was about 200°C.

In order to further clarify the featured HTC, the comparison of the data herein with the previous data from Leocadio²⁴⁾ and Wang¹⁶⁾ are shown in Fig. 11. Although the experimental conditions were not the same, these curves behaved similarly. All curves monotonously increased with

Fig. 10. The variation of HTC with surface temperature at different locations and Re.

Fig. 11. The comparisons of HTC results with the previous literature.

the temperature decreased from 800 to 200°C except the obvious difference in the value of HTC. The increasing speed and maximum of HTC herein were higher than the previous results because the gathered data in this study were much closer to the real ones. The main possible reason for the difference was due to the deviation of data recording process. For one reason was that the temperature dropped abruptly at the initial cooling stage, the rapid response of temperature detector and the accurate temperature of specimens were critical for recording the temperature. As a result, the tight junction between thermocouples and plate, as well as the fast response speed of thermocouple were required. Otherwise, the recorded temperature data and HTC would be lower than the real values. Another reason can be attributed to the different surface roughness and physical property of specimens. The nucleation process of boiling bubbles²⁵⁾ were affected by the roughness of the surface, so as to the HTC, which mainly depend on the boiling heat transfer process.

3.4. Effect of Jet Velocity on the Maximum Heat Flux, T_{Max} , and Wetting Trend

The maximum heat flux varies with the position, as shown in Fig. 12. The higher maximum heat flux was found to be along with shorter cooling time and higher surface temperature. During the transient cooling process, heat flux increased sharply due to the high boiling driving force under high surface superheat. As a result, it reached the maximum quickly and followed by a slow decrease.²⁶⁾ During the

Fig. 12. The maximum heat flux and T_{Max} at different locations and Reynolds numbers.

initial cooling process, the dominant boiling mechanism is transient boiling. Once heat flux reached the maximum heat flux, the boiling mechanism changed from transient boiling into nucleate boiling.^{5,26)} What's more, the higher maximum heat flux indicated the faster boiling frequency and shorter bubble detach time.^{27,28)}

The highest maximum heat flux was at point 9 where was the center impinging point, and the maximum heat flux decreased with the increasing spatial distance to the nearest impinging point without the influence of Re. At impinging point 9, the fluctuation of maximum heat flux was within 0.29 MW/m² (or 6.7%) when the Re ranged from 6 034 to 13 087. From the above results, it can be concluded that the Re had little effect on the maximum heat flux at the impinging point, and this tendency agreed well with the previous studies of single impinging jet.^{5,26)} It is worthy to note that the variation of water flow rate only resulted in small changes in contact pressure and saturation temperature. Such minor change could not cause a significant change in heat flux.

At measurement points of 3, 5, and 8, which spatial location is 5 mm away from the nearest impinging point. The maximum heat flux results here were lower than those at the impinging points. At Re of 13 087, the average maximum heat flux at these points was 3.68 MW/m², which was ~18.9% lower than that at the impinging points. At Re of 6 034, the lowest maximum heat flux was at point 5. However, with the increase of Re, the maximum heat flux did not increase proportionally. At Re of 13 087, the measurement points of 2, 6 and 7, where spatial location are 10 mm away from the nearest impinging point, displayed the average maximum heat flux of 3.73 MW/m², which was ~17.7% lower than that at the impinging points. Meanwhile, the flow states at point 2 and 6 were parallel flow and vortex flow, respectively. The similarity of maximum heat flux at these locations indicated the boiling process was unaffected by the complex flow structure.

Combining with the analysis of HTC, it concluded that the boiling process did not fundamentally alter in the presence of multiple jets. For the strongest cooling location was still at the impinging point, and the vortex flows between jets had little effect on the maximum heat flux.

In the industrial application, the impinging time at a certain position is momentary for the plate always moving at a fast speed. The influenced factors such as the jet velocity and water temperature, both help to enlarge the total heat removal quantity by expanding the effective cooling area rather than substantially elevating the cooling intensity of impinging region.^{5,26)} Therefore, the faster-wetted region expanding speed and less wetting delay time were crucial for the elevation of the cooling ability under moving cooling condition.²⁹⁾ The wetting delay time, t^* , which also named resident time,³⁰⁾ was defined as the period between the beginning of jet impinging on the surface and the moment wetting front began moving³¹⁾ in the present study. Moreover, the t_{Max} represented the cooling time when a location reach its maximum heat flux (s). The relationship between t^* , t_{Max} , and Re can be seen in Fig. 13, in which the patterns of t^* and t_{Max} as a function of measurement points²¹⁾ were similar. This similarity can be attributed to that the maximum heat flux lay in the wetting front region.^{17,30,32)}

Fig. 13. The t^* and t_{Max} at different locations and Reynolds numbers.

However, a reverse trend between maximum heat flux and t^* was found at each location when Fig. 10(a) was taken into consideration, where a higher maximum heat flux corresponded with a shorter t^* and t_{Max} .

At measurement points 4 and 9, the surface wetted immediately after the impingement regardless of the variation in Re. In the inner region, the biggest t^* and t_{Max} existed at the points 7 and apparently higher than those at points 6, and the distances of point 6, 7 to the nearest impinging point were both 10 mm. The observed time differences can be attributed to the center of upwash flow was much closer to point 6. The upwash flow accompanied with much cooling water and stronger kinetic energy, as shown in Fig. 6(b).

At Re of 13 087, the measurement points of 3, 5, and 8, which were 5 mm to the nearest impinging points, became wetted as soon as the impingement started. It took an extra 0.3 s for these points to reach the maximum heat flux when compared with the impinging point. The points located 5 mm away from impinging points were in the stable parallel flow (as shown in Figs. 6 and 7) in spite of collision of multiple jets were involved in. Therefore, the wetting trend patterns were similar in these regions.

At Re of 13 087, the differences between t^* and t_{Max} ranged from 0.47 s to 0.8 s at all the measurement points. These differences increased to 0.87 s and 1.0 s when the Re decreased to 8 326 at points 6 and 7, which were in upwash region between nozzles. However, the time differences at points 1, 2, and 3 kept constant with decreasing Re. A possible explanation was the insufficient cooling water in the region between jets. Moreover, the volume of cooling water was an important factor that affected the wetting process.^{29,30)} Part of cooling water discharge to the outside region (points 1, 2, and 3) driven by the vortex flow at the inner area, this behavior was supported by the characteristic of t^* for points 1, 2, and 3. For the observed t^* which did not decrease with the increase of Re in the outer region was different from the findings in single jet experiments, where the Re and flow rate dramatically affected the movement of wetting front.^{29,30)} The discovered time differences evidenced the drained water would increase the flow rate in the outer region and reduce the impact of Re on flow rate.

4. Conclusions

(1) The flow structure and heat transfer of high-temperature plate cooled by nine-nozzle arrays were studied experimentally and numerically. The results revealed that the water accumulated in the area between adjacent jets, part of accumulated water re-impinged on the surface and part of it drained off from the jets gaps with a low speed.

(2) The temperature differences between measurement points reached the maximum 1 s after the cooling started at Re 13 087. The heat transfer efficiency decreased with extending the cooling time when impinged at a certain position.

(3) At impinging points, increasing Re from 6 034 to 13 087 would have a significant effect on the wetted region expanding speed rather than on the maximum heat flux and the corresponding temperature. The faster-wetted region expanding speed is necessary to increase the heat transfer ability in industrial application.

(4) The wetted region started expanding from the centers of each impinging point, and the maximum heat flux at each measurement point decreased with the distance to the impinging point, both of which were similar to the single impinging jet. The parallel flow collision region had little effect on maximum heat flux, but had a more obvious effect on the t^* , the t^* at point 6 was less than it at point 7 because the center of inside upwash flow was closer to point 6.

The experimental data can be utilized to interpret the heat transfer process under jet arrays, predict the cooling rate, and forecasting the microstructure of the steel product. However, these results were subjected to the experimental condition, in which the linearly arranged nine-nozzle arrays were equidistance of 25 mm. The effect of nozzle distance on the flow field and heat transfer and optimum nozzle arrangement parameters still need to be further studied, additionally, the plate moving speed, jet diameter, jet speed should also be taken into consideration.

Acknowledgement

This work was supported by the National Science Foundation of China (Grant No. 51404058 and 51234002).

Nomenclature

T_s :	surface temperature (°C)
T_w :	water temperature (°C)
ΔT_{sub} :	liquid subcooling (°C)
q :	surface heat flux (W/m ²)
h :	heat transfer coefficient (HTC) (W/m ² °C)
k :	turbulence kinetic energy (m ² /s ²)
t^* :	wetting delay time, the interval between the beginning of jet impinging on the surface and the moment wetting front begins moving (s)

t_{Max} :	cooling time when a location reach its maximum heat flux (s)
T_{Max} :	surface temperature when a location reach its maximum heat flux (°C)
H :	distance between the nozzle exit and the impingement surface (mm)
D :	nozzle diameter (mm)
V :	nozzle impinging speed (m/s)

REFERENCES

- 1) B. S. Gottfried, C. J. Lee and K. J. Bell: *Int. J. Heat Mass Transf.*, **9** (1966), 1167.
- 2) J. J. Carbajo: *Nucl. Eng. Des.*, **84** (1985), 21.
- 3) D. H. Wolf, F. P. Incropera and R. Viskanta: *Int. J. Heat Mass Transf.*, **39** (1996), 1395.
- 4) N. Karwa, L. Schmidt and P. Stephan: *Int. J. Heat Mass Transf.*, **55** (2012), 3677.
- 5) N. Karwa, G. R. Tatiana, P. Stephan and T. Cameron: *Exp. Therm. Fluid Sci.*, **7** (2011), 1435.
- 6) P. Tie, Q. Li and Y. Xuan: *Appl. Therm. Eng.*, **31** (2011), 2757.
- 7) J. Y. San and M. D. Lai: *Int. J. Heat Mass Transf.*, **44** (2001), 3997.
- 8) L. F. G. Geers, K. Hanjalic and M. J. Tummers: *J. Fluid Mech.*, **546** (2005), 255.
- 9) L. F. G. Geers, M. J. Tummers and K. Hanjalic: *Exp. Fluids*, **36** (2004), 946.
- 10) M. J. Kwon and I. S. Park: *ISIJ Int.*, **53** (2013), 1042.
- 11) M. J. Kwon and I. S. Park: *ISIJ Int.*, **53** (2013), 2192.
- 12) M. J. Cho, B. G. Thomas and P. J. Lee: *Metall. Mater. Trans. B*, **39** (2008), 593.
- 13) J. P. Holman: *Heat Transfer*, McGraw-Hill, New York, (1972), 391.
- 14) M. N. Ozisik and H. R. B. Orlande: *Inverse Heat Transfer Fundamentals and Applications*, Taylor&Francis, New York, (2000), 49.
- 15) S. C. Chapra and R. P. Canale: *Numerical Methods for Engineers*, McGraw-Hill, New York, (2015), 832.
- 16) H. Wang, W. Yu and Q. Cai: *J. Mater. Process Technol.*, **212** (2012), 1825.
- 17) A. K. Mozumder, M. Monde and P. L. Woodfield: *Int. J. Heat Mass Transf.*, **48** (2005), 5395.
- 18) R. Matsumoto, I. Ishihara, T. Yabe, K. Ikeda, S. Kikkawa and M. Senda: *Proc 5th ASME-JSME Thermal Engineering Joint Conf.*, New York, NY, (1999), 1.
- 19) I. Filipovic, F. P. Incropera and R. Viskanta: *Exp. Heat Transf.*, **8** (1995), 97.
- 20) M. Monde: *J. Heat Transfer*, **109** (1987), 991.
- 21) A. K. Mozumder, M. Monde, P. L. Woodfield and M. A. Islam: *Int. J. Heat Mass Transf.*, **49** (2006), 2877.
- 22) C. Agrawal, R. Kumar, A. Gupta and B. Chatterjee: *Exp. Therm. Fluid Sci.*, **42** (2012), 25.
- 23) A. K. Nallathambi and E. Specht: *J. Mater. Process Technol.*, **209** (2009), 5325.
- 24) H. Leocadio, J. Cesar, A. Fabio and C. Silva: *Mech. Eng.*, **5** (2009), 3.
- 25) B. J. Jones, J. P. McHale and S. V. Garimella: *J. Heat Transf.*, **131** (2009), 315.
- 26) C. Agrawal, R. Kumar, A. Gupta and B. Chatterjee: *Int. J. Heat Mass Transf.*, **55** (2012), 4772.
- 27) S. G. Kandlikar, V. R. Mizo, M. D. Cartwright and M. D. Ikenze: *ASME Proc. 32nd National Heat Transfer Conf.*, Baltimore, (1997), 11.
- 28) D. P. Rini, R. H. Chenand and L. C. Chow: *J. Heat Transf.*, **124** (2002), 63.
- 29) K. H. M. Abdalrahman, U. Alam and E. Specht: *14th Int. Heat Transfer Conf.*, ASME, New York, NY, (2010), 475.
- 30) J. Hammad, Y. Mitsutake and M. Monde: *Int. J. Therm. Sci.*, **43** (2005), 743.
- 31) B. D. G. Piggott, E. P. White and R. B. Duffey: *Nucl. Eng. Des.*, **36** (1976), 169.
- 32) J. Hammad, M. Monde and Y. Mitsutake: *Therm. Sci. Eng.*, **12** (2004), 19.