

Surface heat flux of upward circular water jet impingement on a moving hot steel sheet¹

Keisuke Nagata^a, Takumi Hirao^b, Kengo Terasaki^c, Tokuma Hikata^d, and Hitoshi Fujimoto^{e*}

^a Graduate School of Energy Science, Kyoto University, Sakyo-ku, KYOTO 606-8501, Japan

Now, Panasonic Energy Corporation

^b Graduate School of Energy Science, Kyoto University, Sakyo-ku, KYOTO 606-8501, Japan

Now, Kansai Electric Power Corporation

^c Graduate School of Energy Science, Kyoto University, Sakyo-ku, KYOTO 606-8501, Japan

Now, JFE Corporation

^d Graduate School of Energy Science, Kyoto University, Sakyo-ku, KYOTO 606-8501, Japan

Now, Kansai Electric Power Corporation

^e Graduate School of Energy Science, Kyoto University, Sakyo-ku, KYOTO 606-8501, Japan

¹Abbreviations

CHF, critical heat flux; MHF, minimum heat flux; ROT, run-out table; TMCP, thermo-mechanical control process

Abstract

This study experimentally investigated the boiling heat transfer characteristics of a circular upward water jet issued from a 7-mm-diameter nozzle impinging on a moving hot 0.3- mm-thick stainless-steel sheet. The moving velocity (1.5, 3.0, and 4.4 m/s), temperature (100–590 °C), and water flow rates (2.5, 3.0, and 3.5 L/min) were varied during the test. The surface heat flux in the jet impact region depended significantly on the initial temperature of the solid. The surface heat flux increases as the solid temperature increases, attains a peak value ($\sim 30 \text{ MW/m}^2$) before decreasing, and reaches a valley value before increasing again. Furthermore, the moving velocity of the solid influences the cooling characteristics, considering that the duration of water contact of a local point on the test sheet is shorter for larger velocities. Moreover, the maximum value of the surface heat flux in the jet impact region can be roughly correlated with the local temperature of the solid. The maximum heat flux peaked at approximately 190 °C and reached a valley at approximately 370 °C. Three empirical correlations for predicting the maximum heat flux in the nucleate, transition, and film boiling are built as functions of the local temperature. The equations in the transition and film boiling can predict the results within a $\pm 20\%$ error margin.

KEYWORDS: run-out table cooling; inverse analysis; boiling heat transfer; impinging jet to a moving solid

1. Introduction

Thermo-mechanical control process (TMCP), combined with hot rolling and subsequent cooling processes, is widely used in the steel-making industries to produce the desired metal structure and properties of steel products [1]. After finishing mills, run-out-table (ROT) cooling is the key component of TMCP. Hot steel sheets are rapidly cooled by water jet impingement [2]. Researchers are widely interested in evaluating the surface heat flux in the jet impact region, considering that heat transfer data help improve the temperature control of steel in the ROT cooling process.

Numerous experimental studies have been conducted on the boiling heat transfer characteristics of water jet cooling to a static hot solid [3]. The water jet impinges on the same point, forming a low-temperature area in the jet impact region. The low-temperature area spreads radially over time [4, 5]. Furthermore, the heat transfer rate is significantly influenced by several factors, such as the temperature of the solid, jet velocity, water temperature, and the type of solid material. Empirical correlations capable of predicting the surface heat flux in the jet impact region have been built as functions of these parameters, depending on the boiling mode [3].

In the actual ROT cooling process, water jets impinge on a moving solid. Unlike in the case of a static solid, the low-temperature zone appears downstream from the jet impact point [6], as shown in Fig. 1. The cooling characteristics of the impinging jet are significantly different from those of a static solid because the contact time of a liquid jet with a local solid is short. Much fewer studies have been conducted on moving solids than static solids. In addition to the abovementioned factors, the moving velocity is also an influencing factor [7, 8]. Therefore, fundamental data on heat transfer are still lacking. In addition, empirical correlations available for predicting the stagnation heat flux for moving solids

are desired.

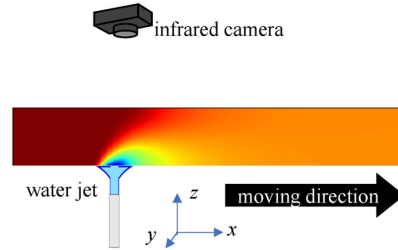


Fig. 1 Schematic of water jet cooling of a moving hot solid.

Several studies have been conducted on the heat transfer characteristics of water jet impingement on a moving hot solid using a pilot plant or an actual cooling system [8-12] and a laboratory-scale setup [6, 7, 13-17]. Most of them have evaluated the heat transfer rate in the jet impact region by inversely solving the transient heat conduction equation using the measured temperatures with thermocouples embedded in a moving hot solid or attached to a solid surface. Because it is experimentally challenging to set up many thermocouples in a small space, the measured temperature distribution in a solid is coarse, thereby preventing surface heat flux measurements from being obtained at high spatial resolution.

The measurement of surface heat flux with high spatial resolution is strongly desired in the industry. A significant gap exists between the literature and the actual industrial demand. Therefore, we developed an evaluation method [18, 19] for surface heat flux in jet impingement on moving hot solids, as shown in Fig. 1. In the evaluation method, the quasi-steady state temperature profile was captured using an infrared camera set on the reverse side of jet impingement. The distribution of surface heat flux on the cooled side was calculated through the steady-state heat conduction analysis using the

measured temperature. Further, we studied the boiling heat transfer characteristics of a single water jet impinging on a moving hot solid based on the evaluation method [19-21]. In these studies, the velocity of the hot solid was limited to less than or equal to 1.5 m/s. However, experimental data using a higher moving velocity are required to meet industrial demands.

This study was scheduled to obtain the fundamental boiling heat transfer data of jet impingement on a moving hot solid at a velocity larger than that in our previous works ($= 1.5, 3.0$, and 4.4 m/s), although the moving velocity was smaller than the actual ROT cooling [12]. The test coolant was tap water at 17 °C, and a 0.3 -mm thick stainless-steel sheet (SUS430) was used. We conducted experiments by varying the solid's moving velocity, the test sheet's initial temperature ($= 100$ – 590 °C), and water flow rates ($= 2.5, 3.0$, and 3.5 L/min). The surface heat flux in the jet impact region was investigated in detail. Data analysis revealed that the peak value of the surface heat flux in the jet impact region can be roughly correlated with the local temperature of the solid. Three empirical correlations for predicting the maximum heat flux in nucleate, transition, and film boiling were proposed as functions of the local temperature.

2. Experimental methods

2.1 Experimental setup

Fig. 2(a) shows a schematic of the experimental apparatus used to measure the surface heat flux in circular jet cooling on a moving hot steel sheet. The setup comprises a test stainless steel sheet mounted on a linear motor actuator, a pipeline system for producing an upward circular jet, and observation equipment for observing the coolant flow and measuring the temperature. Although the experimental

setup was designed to be similar to that used in our previous study [21], it was significantly upgraded to achieve a large moving velocity of the solid.

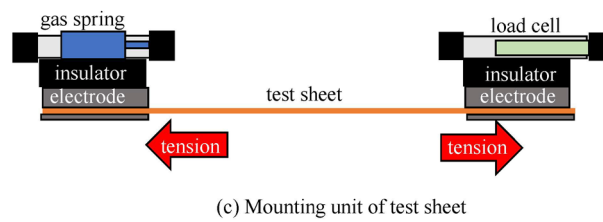
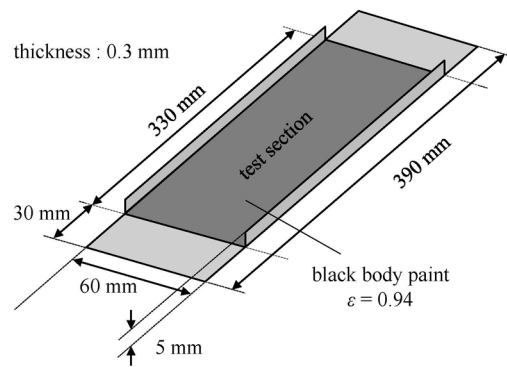
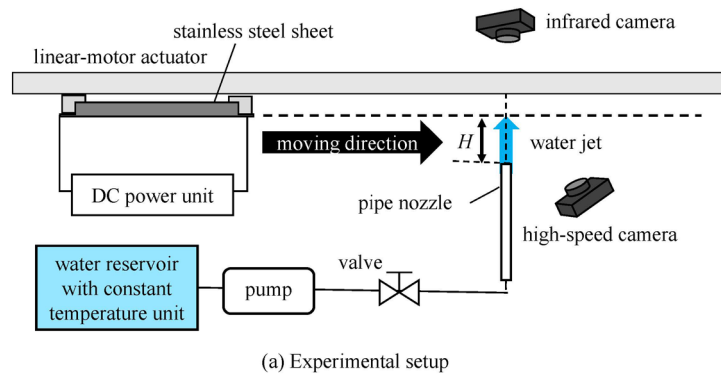


Fig. 2 (a) Experimental apparatus, (b) dimensions of test sheet, and (c) mounting unit.

The experimental procedure is as follows: Initially, the 0.3-mm thick test sheet was set at one end of the actuator with a stroke of 2500 mm (Festo ELGA-TB70, Germany) and heated using electrical resistance heating through a DC power supply with a peak output of 3 kW (Takasago FX010-300) until the temperature was slightly higher than the preset temperature ($T_s = 100\text{--}590\text{ }^\circ\text{C}$). The heating was stopped and the actuator was activated. The hot test sheet accelerated, reached a preset velocity ($V_s = 1.5, 3.0, \text{ and } 4.4\text{ m/s}$), and entered the test section. The lower surface of the test sheet was cooled using a single circular water jet. The temperature profile of the upper surface was measured using a thermographic camera (Nihon Avionics, U100SP). Or the coolant flow motion was recorded with a high-speed camera (Ditect, HAS-L1). The test sheet was then decelerated and stopped at the other end of the actuator. Experiments were conducted by varying the initial temperature of the solid, moving velocity, and water flow rate. Over 10 runs were performed for each experimental condition to ensure the repeatability and scattering of the measured data.

Each component of the experimental setup (pipeline system, test sheet, and measurement equipment) is described in detail herein. The test coolant used was tap water with a temperature of $17\text{ }^\circ\text{C}$, stored in a reservoir with a constant temperature unit (Asone, CARRY COOL 301 CN). A mechanical pump was used to transport the water into the cooling section via a regulating valve. A pipe nozzle with an inner diameter of 7 mm and a length of 400 mm was used to create an upward water jet. The volume flow rates of the water were $Q = 2.5, 3.0, \text{ and } 3.5\text{ L/min}$, indicating that a fully developed turbulent flow was achieved in the pipe. The nozzle-to-plate spacing was set as $H = 20\text{ mm}$. The flow rate, Q , was measured directly from the volume of water discharged during the sampling period ($= 30\text{ s}$). The jet impact velocities considering the gravity effect were $v_i = 0.88, 1.14, \text{ and } 1.38\text{ m/s}$. The jet impact

velocity was lower than the moving velocity of the test sheet in the experiments. The corresponding Reynolds numbers based on the jet impact velocity and diameter just before impact were $Re_j = 6,800$, 8,500, and 10,100, respectively.

The dimensions of the test sheet, made of stainless steel (SUS430), are shown in Fig. 2(b). The test section was 330 mm long and 60 mm wide. The test sheet was 0.3 mm thick. The arithmetic mean surface roughness of the test sheet on the cooled side was within 0.2 μm . A thin coat of black body paint with an emissivity of 0.94 was added on the reverse side to accurately measure the temperature. The longer sides of the sheet were bent 5 mm from the edge at a right angle. The U-shaped cross-section of the test sheet prevented unwanted buckling of the test sheet owing to thermal stresses [19]. The test sheet was firmly attached to a pair of mount units, which comprised electrode plates, thermal and electrical insulators, a gas spring to pull the test sheet in the longitudinal direction, and a load cell to measure the tensile force on the test sheet, as shown in Fig. 2(c).

Several preliminary cooling tests were conducted by varying the tensile force. Pulling the test sheet prevented unwanted buckling, which caused irregular coolant flows. The magnitude of the tensile force did not influence the heat transfer characteristics unless any unwanted buckling of the test sheet occurred. In regular experiments, we set the tensile force to approximately 150 N.

The flow motion of water in the jet impact region was observed with a high-speed camera with a resolution of 320×240 pixels at 1500 fps. A metal halide lamp with a power of 500 W was used as a light source for video recording.

The temperature profile of the test sheet on the reverse side of the jet impingement surface was measured using an infrared camera with a resolution of 320×240 pixels at 60 fps. The infrared camera-

to-test sheet spacing was approximately 300 mm, and the corresponding observation area was 180×135 mm. The temperature measurement and video recording were exclusively conducted under darkroom conditions to minimize the measurement errors originating from any light sources.

The measurement accuracy was within $\pm 2\%$ of the indicated temperature of the thermal camera (specified by the manufacturer). The thermal camera was calibrated using a commercial calibrator for radiation thermometers (CES, BX-500) in the temperature range of 100-500 °C at an emissivity of 0.95, as listed in Table 1. The accuracy of the calibrator was within ± 2.5 °C (specified by the manufacturer). At higher temperatures, the temperatures indicated by the thermal camera were slightly lower than the data indicated by the calibrator. Herein, the temperatures are presented based on the values indicated by the thermal camera; no adjustments using Table 1 were made.

In this section, we elucidate the reasons behind the selection of the current experimental conditions. Considering the industrial context of our study, which focuses on ROT cooling in steel-making industries, water and stainless steel were chosen as the test coolant and test material, respectively. The maximum value of the initial temperature in our study ($T_s = 590$ °C) was intentionally kept lower than that in actual ROT cooling (~ 900 °C) [1, 2] due to limitations imposed by the experimental equipment. Nevertheless, our experiments successfully assessed the boiling heat transfer characteristics in the nucleation, transition, and film boiling regions. Furthermore, the size of the test sheet and the water flow rate was intentionally kept smaller than those in actual ROT cooling [12] due to the utilization of a laboratory-scale setup. The moving velocity of the test sheet ($V_s = 1.5, 3.0, \text{ and } 4.4$ m/s) was increased compared to our previous studies [19-21], although it remains smaller than the industrial operation (which typically exceeds 10 m/s) due to constraints posed by the experimental facilities.

Table 1 Calibration data of the thermal camera

Temperature of calibrator, °C	Indicated temperature of thermal camera, °C
100.0	99.1
149.9	146.0
199.5	195.4
249.8	244.3
299.7	292.6
349.5	342.2
399.3	391.0
449.5	441.7
500.0	491.8

2.2 Data reduction for evaluating surface heat flux

The surface heat flux in the jet impingement region was evaluated by numerically solving the heat conduction equation using the measured temperature profiles [18, 19]. The basic idea of the solution method is mentioned. The transient heat conduction equation in a three-dimensional coordinate system fixed in space is given by

$$\rho c \left(\frac{\partial T}{\partial t} + V_s \frac{\partial T}{\partial x} \right) = \lambda \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right), \quad (1)$$

where t is the time, and (x, y, z) is the coordinates in the solid's moving, width, and thickness directions, as shown in Fig. 1. T , ρ , λ , and c are the local solid temperature, density, thermal conductivity, and heat capacity, respectively.

When the moving test sheet entered the cooling section, the test solid was significantly cooled in the jet impact region. Low-temperature area was expanded downstream, obeying Eq.(1). Subsequently, the temperature distribution of the solid became quasi-steady in the jet impact region, as shown in Fig. 1, and the following steady-heat conduction equation is true,

$$V_s \frac{\partial T}{\partial x} = \frac{\lambda}{\rho c} \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right). \quad (2)$$

The time required to achieve the quasi-steady state temperature profile is the sum of the two transient times. The one is the hydrodynamic time to stably form the radial spreading flow along the test surface due to jet impingement. High-speed videography revealed that the time looked very short (see **Fig. 3**). The other is the thermal time due to heat conduction after the stable jet impingement on the moving test sheet begins. The time can be calculated by the exact solution of the transient heat conduction equation [18] and depends on the thickness of the test sheet and the size of the analysis domain. In the present analysis, the sum of the two transient times was approximately 50 ms or less. The corresponding moving length of the test sheet was 220 mm for the moving velocity of $V_s = 4.4$ m/s, whose distance was shorter than the length of the test sheet (= 330 mm). Therefore, the quasi-steady temperature profile of the test surface can be captured, and the heat conduction analysis using Eq. (2) is possible.

The surface heat flux was evaluated using the method reported by Hiakata and Fujimoto [18]. **Fig. 4** shows a schematic of the analysis domain composed of numerous small rectangular cells. Eq. (2) was discretized with a finite difference method in the following form:

$$V_s \frac{-2T_{i-1,j,k} - 3T_{i,j,k} + 6T_{i+1,j,k} - T_{i+2,j,k}}{6\Delta x} = \frac{\lambda}{\rho c} \left(\frac{T_{i+1,j,k} + T_{i-1,j,k} - 2T_{i,j,k}}{\Delta x^2} + \frac{T_{i,j+1,k} + T_{i,j-1,k} - 2T_{i,j,k}}{\Delta y^2} + \frac{T_{i,j,k+1} + T_{i,j,k-1} - 2T_{i,j,k}}{\Delta z^2} \right) \quad (3)$$

in which (i, j, k) and $(\Delta x, \Delta y, \Delta z)$ are the cell numbers and sizes, respectively. The second-order central differencing scheme was used in the second derivative terms. Also, the first derivative term adopted a third-order downwind UTOPIA scheme. By rewriting Eq.(3), we obtain

$$T_{i,j,k-1} = \Delta z^2 V_s \frac{\rho c}{\lambda} \frac{-2T_{i-1,j,k} - 3T_{i,j,k} + 6T_{i+1,j,k} - T_{i+2,j,k}}{6\Delta x} - \Delta z^2 \left(\frac{T_{i+1,j,k} + T_{i-1,j,k} - 2T_{i,j,k}}{\Delta x^2} + \frac{T_{i,j+1,k} + T_{i,j-1,k} - 2T_{i,j,k}}{\Delta y^2} + \frac{T_{i,j,k+1} - 2T_{i,j,k}}{\Delta z^2} \right). \quad (4)$$

Boundary conditions have also been presented in Fig. 4. At the side boundaries in the y -direction, the zero-gradient condition was imposed. At the upstream boundary in the x -direction, the temperature equals the initial value, whereas the zero-gradient condition was imposed at the downstream boundary. The temperature, $T_{Z=H}$, was given at the top surface of the test sheet ($z=H$). Moreover, the following boundary condition was true,

$$\alpha_{top} (T_{Z=H} - T_{air}) = -\lambda \frac{\partial T}{\partial z} \Big|_{z=H} \quad (5)$$

in which T_{air} and α_{top} are the temperatures of surrounding air and heat transfer coefficient. The heat transfer coefficient was set to 50, considering the forced convection along the flat plate surface and thermal radiation [22].

Inverse analysis was conducted in the following manner. First, using Eq. (5), a finite difference approximation can calculate the temperatures at a small distance from the top surface ($z=H$). As the temperatures at the top surface and the first grid points away from the surface are known, the

temperatures at the second grid points can be calculated by Eq. (4). Thereafter, the temperature of the solid is evaluated using Eq. (4) in a similar manner from the top surface to the bottom surface ($z=0$). The surface heat flux profile at the bottom surface can be evaluated using Fourier's law using the temperature profile near the bottom surface.

The analysis domain with a length of 95 mm, width of 50 mm, and thickness of 0.3 mm was divided into $170 \times 90 \times 200$ cells in the x -, y - and z -directions. The mesh sizes in the x and y directions corresponded to the pixel size of a thermal camera, and several grid sensitivity tests determined the size in the thickness direction ($= 0.0015$ mm).

The accuracy of the evaluation method was thoroughly investigated in our previous study [18] through the following approach. First, we created a test temperature profile on the uncooled surface using the exact solution of the heat conduction equation. Subsequently, inverse analysis was conducted using the test profile. As a result, the evaluated surface heat flux closely matched the given profile with high accuracy, and the estimation error caused by the finite difference approximation was minimal. Second, the evaluation method underwent testing using another temperature profile, generated by adding the exact solution to random noise. The evaluation method was found to be highly sensitive to random error noise, necessitating the use of data smoothing operations employing B-spline functions in the reduction of the measured temperature profile. However, this smoothing operation induced some local deviation between the evaluated surface heat flux and the given profiles. The maximum value of this local deviation was a few MW/m^2 . A similar magnitude of local evaluation error was considered unavoidable in the present analysis. Additionally, the evaluation error is significantly larger than the error arising from the inaccuracies in the heat transfer rate at the top surface in Eq. (5).

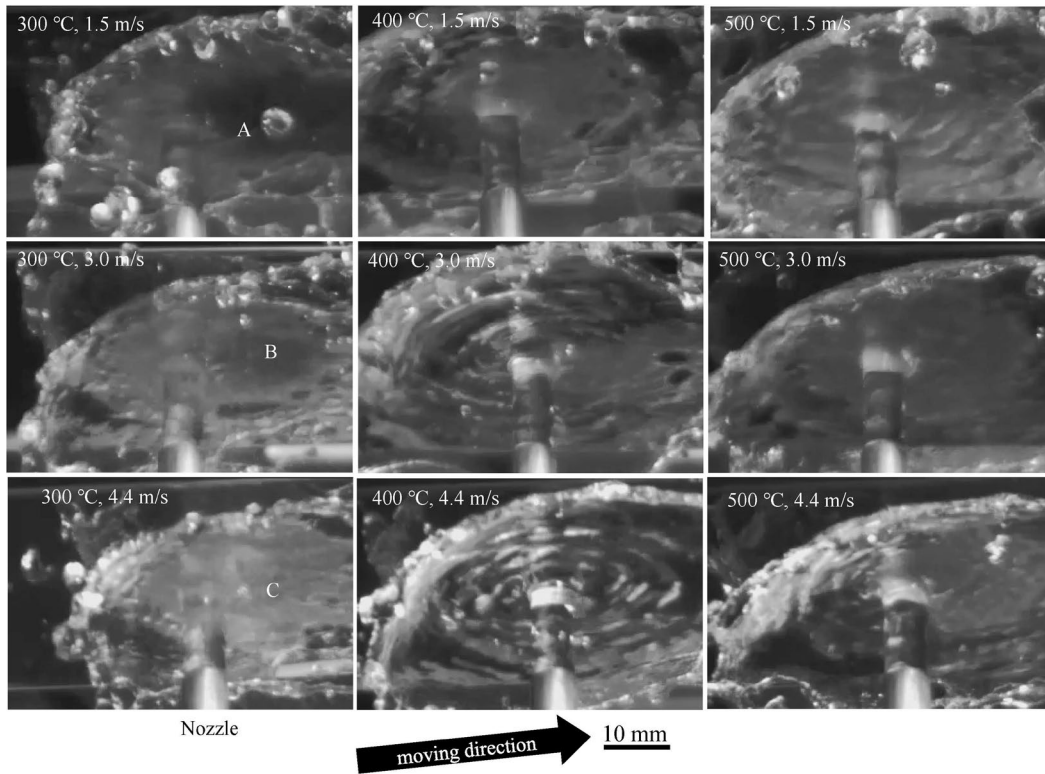


Fig. 3 Instantaneous images of water flow for $Q = 3.0$ L/min and $V_s = 3.0$ m/s.

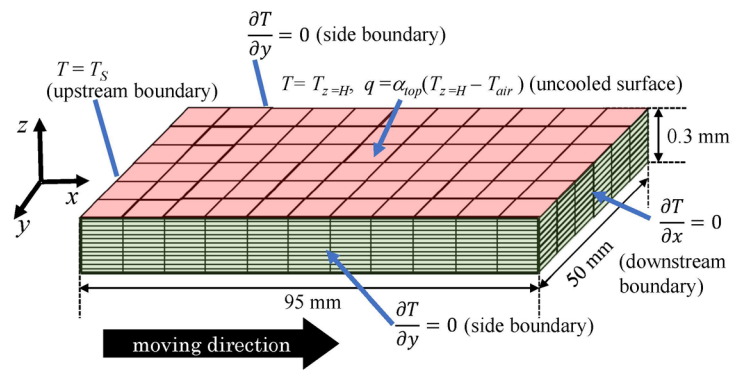


Fig. 4 Schematic of analysis domain.

3. Results and discussion

3.1 Effect of varying the initial temperature of the test sheet and moving velocity on the hydrodynamics and heat transfer characteristics of impinging jets

Fig. 3 shows the instantaneous images of water flow motion near the jet impact point under a water flow rate $Q = 3.0$ L/min, moving velocity of $V_s = 1.5, 3.0,$ and 4.4 m/s, and an initial temperature of test sheet of $T_s = 300, 400,$ and 500 °C. In all cases, circular water film was formed along the solid surface from the jet impingement point. The water film was left due to gravity at the edge. In addition, high-speed videography revealed that the radially spreading film flow became stable immediately after the test sheet entered the test section, suggesting that steady heat conduction was allowable.

The impact of varying the temperature of the solid on flow boiling phenomena is described, with reference to the results for $V_s = 4.4$ m/s. At $T_s = 300$ °C, nucleate boiling occurred. However, due to low video resolution, the boiling vapor bubbles could not be directly observed in the images. The presence of vapor bubbles and/or small droplets was indirectly confirmed from the cloudy (hazy) water film where numerous small bubbles/droplets scattered the observation light. At $T_s = 400$ °C, the water film was transparent, while at $T_s = 500$ °C, the water film was slightly cloudy. Given the initial temperature of the test sheet, film boiling occurred, and a vapor layer was present between the solid surface and water. The boiling mode for each result will be discussed in section 4.2 using the relation between the initial temperature and the heat flux.

Some influence of the moving velocity on the boiling phenomena was observed in the results at $T_s = 300$ °C. The water film appeared cloudy upstream of the jet impingement point, irrespective of the moving velocity of the solid, indicating nucleation boiling. The downstream areas marked by the symbols “A,” “B,” and “C” were darker for smaller moving velocity. In particular, the area “A” for $V_s = 1.5$ m/s was extremely dark, suggesting the presence of fewer vapor bubbles.

Fig. 5 shows the measured temperature profiles of the test sheet on the uncooled surface (top surface) under the same experimental conditions as in the previous figure. The black circles plotted in each image represent the estimated jet impact points. For $T_s = 300^\circ\text{C}$, a linear low-temperature zone was elongated from the jet impact point in the moving direction. The temperature in the zone decreased as the moving velocity V_s decreased, considering the contact time of the water jet with the local point on the test sheet was longer. The results align with the observation that nucleate boiling was less pronounced on the downstream side of the jet impact point for the smaller moving velocity, as illustrated in Fig. 3. Furthermore, the width of the low-temperature zone was narrower for the larger moving velocity, indicating that the effective cooling width due to jet impingement did not coincide with the size of the spreading circular water film in Fig. 3.

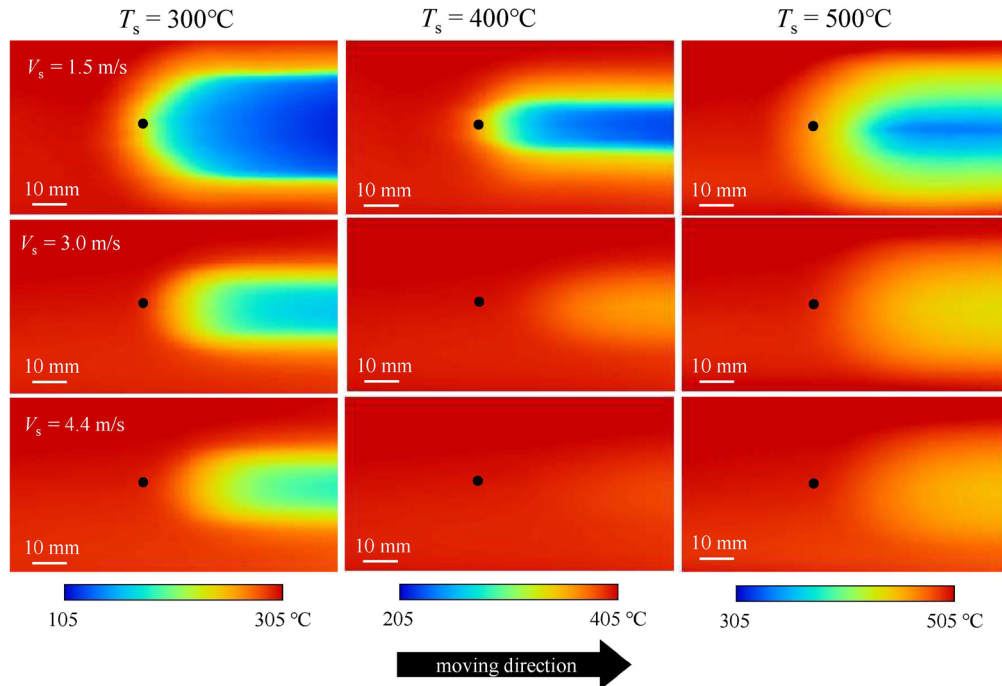


Fig. 5 Measured temperature profiles for $Q = 3.0$ L/min on the uncooled surface (top surface) for

$$Q = 3.0 \text{ L/min.}$$

The upstream edge of the low-temperature zone shifted downstream as the moving velocity increased, considering the temperature drop on the cooled surface propagated to the reverse side owing to heat conduction. The test sheet moved a certain distance when the temperature drop was reflected on the reverse side with some delay. The distance was larger for a greater moving velocity.

The initial temperature of the solid influenced the measured temperature profile. For $V_s = 3.0$ and 4.4 m/s, the temperature drop was large at $T_s = 300$ °C in nucleate boiling, as shown in Fig. 3, which decreased at $T_s = 400$ °C before slightly increasing at $T_s = 500$ °C in film boiling. For $V_s = 1.5$ m/s, the low-temperature zone was narrow at $T_s = 400$ °C compared to when $T_s = 300$ and 500 °C.

Fig. 6 shows the estimated surface heat flux on the cooled surface (bottom surface) under the same experimental conditions as in the previous figure. The shape and size of the high heat flux region exhibited rough line symmetry in the width direction but asymmetry in the moving direction. They depended on the initial temperature of the solid and the moving velocity. Table 2 lists the locations indicating the highest heat flux (maximum heat flux) point in each analysis concerning the jet impingement point. The location of the peak point also depended on the parameters.

Table 2 Locations of peak heat flux point with respect to the center of jet impingement

	$T_s = 300$ °C	$T_s = 400$ °C	$T_s = 500$ °C
$V_s = 1.5$ m/s	(-5.6 mm, 1.7 mm)	(-2.8 mm, 0.6 mm)	(8.3 mm, -0.6 mm)
$V_s = 3.0$ m/s	(0.0 mm, 2.8 mm)	(12 mm, -0.6 mm)	(1.7 mm, 0.6 mm)
$V_s = 4.4$ m/s	(4.5 mm, 3.9 mm)	(15 mm, -0.6 mm)	(0.6 mm, 0.6 mm)

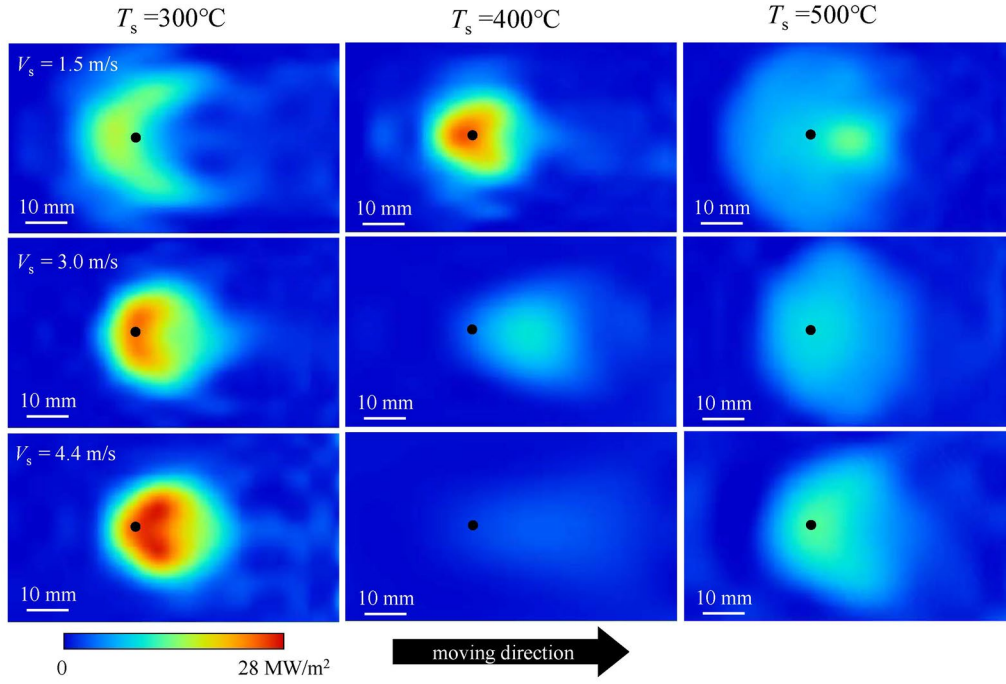


Fig. 6 Estimated surface heat flux for $Q = 3.0$ L/min on the cooled surface (bottom surface) for $Q = 3.0$ L/min.

A bow-shaped high heat flux region was observed for $T_s = 300$ °C and $V_s = 1.5$ m/s, whereas circular regions were observed for $V_s = 3.0$ and 4.4 m/s. The location of the peak point shifted downstream with an increase in the moving velocity. The mechanism of these results is discussed herein. The impact inertia of the jet produced some wet areas showing nucleate boiling around the impact point. The shape of the wet area was circular in cases of the static solid but not in the present cases. This is because of the large viscous wall friction due to the movement of the test sheet, which depends on the moving velocity and acts in the moving direction. Wet area was assumed to be present in the upstream region to the impact point at $V_s = 1.5$ m/s but not at $V_s = 3.0$ and 4.4 m/s. This can be confirmed indirectly from the fact that the low-temperature zone was narrower for a larger moving velocity, as shown in Fig. 5. In addition, the surface heat flux is extremely high where the cooling water is in first contact with a dry

moving solid. Therefore, a bow-shaped high heat flux region is formed on the upstream side at $V_s = 1.5$ m/s. Circular regions were observed near the impact point at $V_s = 3.0$ and 4.4 m/s, where heat flux was achieved on the upstream half side.

For $T_s = 400$ °C, high heat flux was present in the jet impact region at $V_s = 1.5$ m/s, whereas the heat flux was considerably small for $V_s = 3.0$ and 4.4 m/s. This experimental condition is close to the critical boundary of the surface temperature, at which some wet area appeared around the jet impact point. Since the contact time of the water jet with the local point on the test sheet was shorter for larger moving velocity, the decrease in the local surface temperature in the moving direction is smaller. Consequently, nucleate boiling was observed at $V_s = 1.5$ m/s, and the peak point was present in the jet impact region. In contrast, a very small heat flux was obtained for $V_s = 4.4$ m/s.

For $T_s = 500$ °C and $V_s = 4.4$ m/s, a relatively high heat flux region was observed around the jet impact point. The value is higher than that for $T_s = 400$ °C and $V_s = 4.4$ m/s because the temperature difference between the solid and coolant was larger. Similar results were obtained for $V_s = 3.0$ m/s. For $V_s = 1.5$ m/s, high heat flux appeared at a downstream point to the jet impact point. As the surface temperature decreases in the moving direction during the jet impingement, the chance of water/solid contact increased.

The heat flux profile obtained in this study was asymmetric. Therefore, using heat transfer data for static solids is unsuitable for predicting the cooling characteristics in actual ROT cooling.

3.2 Maximum heat flux under various experimental conditions

Fig. 7 shows the maximum heat flux under various experimental conditions. In this study, multiple runs were conducted under each cooling condition. The results are shown as the arithmetic mean values, with error bars representing the standard deviation. The initial temperature of test sheet ranged $T_s = 200\text{--}590\text{ }^{\circ}\text{C}$ with the interval of $50\text{ }^{\circ}\text{C}$ (except for $590\text{ }^{\circ}\text{C}$) for $V_s = 1.5$ and 3.0 m/s , and $100\text{--}590\text{ }^{\circ}\text{C}$ for $V_s = 4.4\text{ m/s}$. The maximum heat flux increased with the initial temperature of the solid, thereby achieving a maximum value before decreasing and reaching a minimum and then increasing again. The relationship between the maximum heat flux and the initial temperature of the solid was similar to a typical pool boiling curve [23], but the present heat flux is much higher. The boiling mode at the peak point was determined using the chart. Non-boiling was observed at low temperatures of the solid, after which nucleate boiling occurred in a relatively low-temperature range, less than the local peak. The boundary between non- and weak-nucleate boiling cannot be correctly reviewed using the present experimental technique. Transition boiling occurred in the temperature range between the local peak and valley values, and film boiling occurred in a range higher than the valley value. The nucleate-to-transition and transition-to-film boiling boundaries are called the critical heat flux (CHF) and minimum heat flux (MHF) points, respectively. As the experiments were conducted at intervals of $50\text{ }^{\circ}\text{C}$, we could not precisely determine the CHF and MHF points. Nevertheless, for convenience, we considered the CHF and MHF points at temperatures that showed a peak and valley, respectively. In addition, the accurate inverse analysis was difficult when the variation in the measured temperature profile was small, similar to the result for $V_s = 4.4\text{ m/s}$ and $T_s = 400\text{ }^{\circ}\text{C}$, as shown in Fig. 6. Several data points indicated

by ovals in the figure fulfilled such a condition for inverse analysis, and hence, the estimation accuracy is poor.

The water flow rate (or jet impact velocity) significantly affects the flow boiling and heat transfer characteristics. However, in this study, it could only be set in a small range owing to the experimental restrictions. Experiments with large water flow rates were impractical owing to the unwanted buckling of the test sheet occurring easily and the heat capacity of the test sheet becoming exceedingly small. We observed a small effect caused by varying the water flow rate on the maximum heat flux in Fig. 7, where the maximum heat flux values at the CHF and MHF points were slightly higher for larger water flow rates.

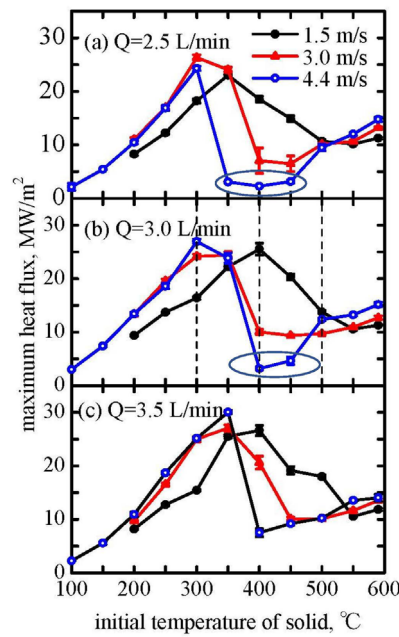


Fig. 7 Variation of maximum heat flux with the initial temperature of the solid.

Conversely, the moving velocity of the test sheet influenced the maximum heat flux, particularly in the transition boiling regime. Both the CHF and MHF points shifted to a higher initial temperature range

for lower moving velocities, thereby indicating that the boiling mode cannot be determined by the initial temperature of the solid alone.

Regarding the maximum heat flux chart, we determined the boiling mode at the peak points of the results, as shown in Figs. 5 and 6. The boiling mode was weak nucleate boiling at $T_s = 300\text{ °C}$ for $V_s = 1.5\text{ m/s}$ and strong nucleate boiling close to CHF for $V_s = 3.0$ and 4.4 m/s . The initial temperature $T_s = 400\text{ °C}$ was close to the CHF point for $V_s = 1.5\text{ m/s}$ and near the MHF point for $V_s = 3.0$ and 4.4 m/s . At $T_s = 500\text{ °C}$, the boiling mode was transition boiling for $V_s = 1.5\text{ m/s}$. For $V_s = 3.0$ and 4.4 m/s , the boiling mode was film boiling.

3.3. Relation between maximum heat flux and local temperature

The chart of the maximum heat flux to the initial temperature, as shown in Fig. 7, is confusing for understanding the boiling mode because the maximum heat flux significantly depends on the initial temperature of the solid and moving velocity; hence, a better data reduction method was sought. Consequently, we found that the local surface temperature of the solid at peak points instead of the initial temperature can correlate well with the experimental data. Fig. 8 shows the correlation between the maximum heat flux and local surface temperature estimated by inverse analysis. Unlike the case of Fig. 7, the data were not shown by the arithmetic mean but by each experimental value. This is because the maximum heat flux and estimated local temperature are conditionally scattered in each run, even under the same conditions. We conducted additional experiments with the following conditions: nozzle diameter $d = 10.5\text{ mm}$, water flow rate $Q = 6.7\text{ L/min}$, corresponding jet impact velocity $v_i = 1.14\text{ m/s}$, moving velocity $V_s = 3.0$ or 4.4 m/s , and initial temperature of test sheet $T_s = 250\text{--}550\text{ °C}$. The jet

Reynolds number was $Re_j = 12800$. Longer and wider test sheets were used ($440 \times 70 \times 0.3$ mm) compared to $d = 7.0$ mm. Note that the unwanted buckling of the test sheet occurred for the case of $(T_s, V_s) = (350 \text{ }^\circ\text{C}, 3.0 \text{ m/s})$. The results are shown in Fig. 8.

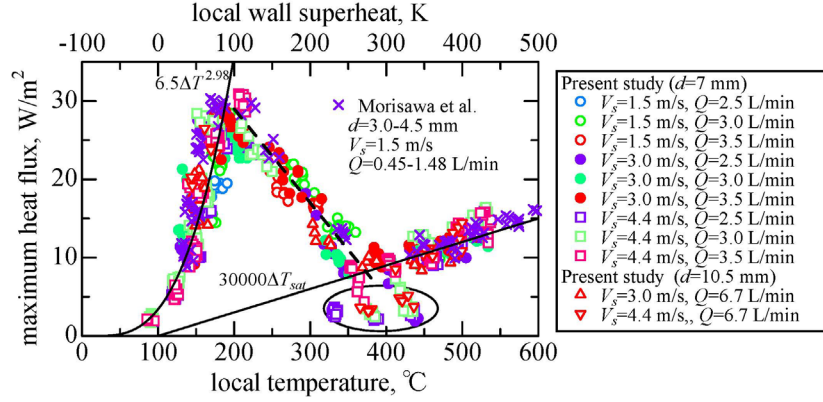


Fig. 8 Relation between the maximum heat flux and local temperature.

In our previous study [21], the boiling heat transfer characteristics of an upward circular jet impinging on a moving solid were investigated on the conditions that $d = 3.0-4.5$ mm, $Q = 0.45-1.48$ L/min, $v_i = 0.97-1.49$ m/s, $V_s = 0.5-1.5$ m/s, and $T_s = 300-700$ $^\circ C$. The original experimental data for $V_s = 1.5$ m/s have also been added to Fig. 8 for reference. All the data plotted in Fig. 8 were obtained under the condition that the water temperature was 17 $^\circ C$, and the test material was stainless steel (SUS430) with 0.3 mm thickness, the moving velocity of the solid was faster than the jet impact velocity.

Although various data obtained under different experimental conditions are plotted in one figure, a strong correlation was observed between the maximum heat flux and local temperature. Maximum heat flux increased as the local temperature increased to approximately 190 $^\circ C$, thereby attaining a peak, before steeply decreasing to 190–370 $^\circ C$ and increasing again. The maximum heat flux was modestly

scattered, particularly near the CHF point, owing to the low repeatability of the experiments caused by strong nucleate boiling and the influence of various parameters, including the jet impact velocity. The limit of superheat of water at atmospheric pressure, which is approximately 300 °C [24, 25], is present in the transition boiling regime. Around the MHF point, the data indicated by an oval were less accurate, considering the variation in the measured temperature profile was similar to that for $V_s = 4.4$ m/s and $T_s = 400$ °C in Fig. 6. The data scatter was relatively small in the film-boiling regime, which indicated that the local temperature could roughly determine the maximum heat flux. To the best of our knowledge, this has not yet been reported in the literature.

3.4. Empirical correlations between maximum heat flux and local temperature

We attempted to build empirical correlations capable of predicting the maximum heat flux in nucleate, transition, and film boiling regimes as shown in Fig. 8, referring to the prior studies on static solids. In fully developed nucleate boiling for the cooling of a static solid, the stagnation heat flux was often correlated by the function as $q = C\Delta T_{sat}^m$ where q and ΔT_{sat} are the stagnation heat flux and the local wall superheat [3], [26-28]. C and m are fitting parameters to the experimental data. By using a similar function form, the following correlation was built,

$$q_{max} = 6.5\Delta T^{2.98} \quad (6)$$

in which ΔT is the difference between the local solid and water temperatures ($=17$ °C). To fit some data below the boiling temperature of the water, ΔT was employed instead of the wall superheat.

Liu and Wang [29] built a semi-empirical correlation for predicting the stagnation heat flux for a static solid in the film boiling regime as $q = 1.41 Re_j^{0.5} Pr^{0.166} (\lambda_l \lambda_v \Delta T \Delta T_{sat})^{0.5} / d_j$ in which Pr , λ_l , λ_v , and

d_j are the Prandtl number, thermal conductivities of liquid and vapor, and the jet diameter, respectively. However, such formulation was unavailable in the present data because the maximum heat flux showed little jet Reynolds number dependence. Moreover, the correlation of Liu and Wang was significantly smaller than the present experimental data. In the present study, the following simple correlation based on Newton's law of cooling was built since the maximum heat flux increased linearly with the local temperature of the solid as,

$$q_{max} = 30000\Delta T_{sat}. \quad (7)$$

In transition boiling, the variation of maximum heat flux is correlated by the following linear function fitting to the data,

$$q_{max} = -120000\Delta T_{sat} + 41000000. \quad (8)$$

Fig. 9 shows the comparison between these correlations and the measured data. In (a), Eq. (6) and the measured data were compared under conditions where the local surface temperatures were less than 190 °C, which is an intersection of the lines of Eqs. (6) and (8). Eq. (6) was in modest agreement with the measured data. In (b), Eq. (7) and the measured data are presented in the range of local surface temperatures above 373 °C, which is an intersection of the lines of Eq. (7) and (8). Except for the data indicated by a circle in Figure 8, the correlation can predict the maximum heat flux roughly within $\pm 20\%$ in the film boiling regime. Additionally, Eq. (8) fits the experiments in the transition boiling regime roughly within $\pm 20\%$, as shown in Fig. (c).

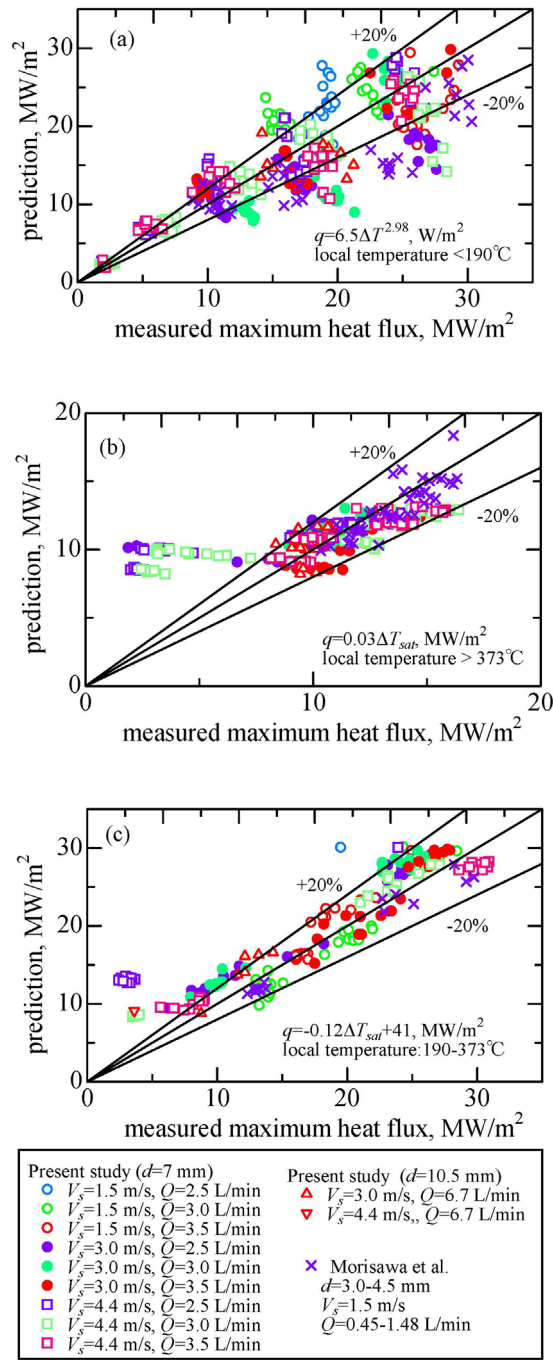


Fig. 9 Comparison of correlations with experimental data.

Eqs. (6)–(8) were fitting correlations without any theoretical backgrounds but agreed moderately well with the present experimental data. Correlations based on “local surface temperature” would be

useful forms to predict the surface heat flux in industrial applications. However, a significant gap exists between the present experiment and actual ROT cooling. Further experimental studies are required to confirm the applicability of these correlations from an industrial perspective, posing a significant challenge for future research.

4. Conclusions

This study investigated the heat transfer characteristics of an upward single water jet impinging on a moving hot solid through laboratory-scale experiments. The experiments involved varying the initial temperature of the solid, the moving velocity of the solid, and the water flow rate. The key findings of this study are summarized as follows:

- (1) The low-temperature region of the solid extended from the jet impact point on the uncooled surface. The temperature in this zone decreased as the moving velocity decreased, indicating that the contact time of the water jet with the local point on the test sheet increased. In nucleate boiling, the width of the zone was narrower for larger moving velocities, suggesting that the effective cooling area in the jet impingement region did not align with the size of the spreading circular water film.
- (2) In nucleate boiling, the surface heat flux was significantly high in the region where the cooling water first contacted a dry moving solid. The shape and size of the high surface heat flux region depended not only on the initial temperature of the solid but also on the moving velocity. The temperature of the solid surface decreased in the moving direction, influencing the heat flux profile, resulting in an asymmetric heat flux distribution in the moving direction.

- (3) The maximum heat flux curve with respect to the initial temperatures of the solid resembled a typical pool boiling curve, albeit with much higher heat flux values. The initial temperatures of the solid indicating the critical heat flux (CHF) and maximum heat flux (MHF) points were influenced by the velocity of the solid.
- (4) The chart of the maximum heat flux to its local temperature showed an S-shaped curve, which indicated that the local temperature is a suitable parameter for correlating the heat transfer characteristics. The maximum heat flux reached a peak at approximately 190 °C and reached a valley at approximately 370 °C.
- (5) Three empirical correlations in nucleate, transition, and film boiling regimes were established as functions of the local temperature alone. The equations for transition and film boiling could predict the present results approximately within $\pm 20\%$. While a considerable gap exists between the present experiment and actual ROT cooling, the correlations based on local temperature have the potential to simulate the temperature profile in a commercial mill. Further experimental studies are essential to confirm the applicability of these correlations from an industrial perspective, presenting a significant challenge for future research.

Declaration of generative AI in scientific writing

The authors declare that they did not use any AI technologies in writing.

Author contributions

Keisuke Nagata: Investigation, Visualization, Writing-Original Draft

Takumi Hirao: Investigation, Visualization

Kengo Terasaki: Investigation, Visualization

Tokuma Hikata: Methodology, Validation

Hitoshi Fujimoto: Supervision, Writing-Review and Editing, Conceptualization

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