Heat Transfer Characteristics of a Circular Water Jet Impinging on a Moving Hot Solid

Hitoshi FUJIMOTO,1)* Katsutoshi TATEBE,2) Yamato SHIRAMASA,3) Takayuki HAMA3) and Hirohiko TAKUDA1)

1) Graduate School of Energy Science, Kyoto University, Kyoto, 606-8501 Japan.
2) Graduate School of Energy Science, Kyoto University, (Presently, Nippon Steel & Sumitomo Metal Corp.)

(Received on August 5, 2013; accepted on January 9, 2014)

The heat transfer characteristics of a circular water jet impinging on a moving hot solid were investigated experimentally. In the experiments, distilled water at room temperature was used as the test coolant. The circular jet issued from a 5-mm-diameter pipe nozzle, fell vertically downward, and impinged on a horizontal moving sheet made of 0.3-mm-thick stainless steel. The initial temperature of the sheet, the jet velocity, and the moving sheet velocity were varied systematically. The initial temperature of the moving sheet was set to 100, 150, 200, or 250°C. The mean velocity at the nozzle exit was 0.4, 0.8, or 1.2 m/s, and the moving velocity was 0.5, 1.0, or 1.5 m/s. Observations made using flash photography and thermography showed that the location of the front edge of the liquid film formed upstream of the jet impact point depends on all of these factors. The local heat flux is very small in the dry area, increases steeply near the front edge of the liquid film, and reaches a peak. If the distance between the front edge of the liquid and the jet impact point is relatively large, a second peak appears near the jet impact point. An experimental correlation was developed for predicting peak heat fluxes near the front edge of the liquid, although it has no theoretical background. The correlation agrees moderately well with the experiments.

KEY WORDS: strip cooling; infrared thermography; impinging jet; heat transfer.

1. Introduction

The impingement of liquid jets on hot materials can help achieve a high heat removal rate in the vicinity of the impact point. This cooling technique is quite common in metal manufacturing industries. In the hot rolling process, hot metal sheets are cooled by circular water jets, planar jets, or sprays. This cooling plays a role in producing the desired mechanical properties of the metals. Thus, precise control of the temperature history is highly important. In addition, the cooling equipment should be designed to prevent unwanted deformation of the metal sheets due to the local thermal stress resulting from temperature variations. In light of these considerations, the ability to predict the heat transfer rate between the water jets and hot metal sheets with high accuracy would address one of the significant issues with this cooling technique.

For many decades, extensive studies have been carried out concerning the characteristics of heat transfer between water jets and hot metals.1–12) It is well-known that the heat flux from the hot solid to the jets depends on the temperature of the solid and the existence of boiling regimes. The heat flux is also dependent on the coolant conditions, including the jet diameter and the velocity and temperature of the coolant. In addition to these parameters, the moving velocity of the solid influences the heat transfer characteristics.8–12) Chen et al.8) studied the heat transfer between an upward circular water jet and a moving metal sheet at temperatures above and below the boiling point. The researchers reported that the effect of the sheet motion was to elongate the water layer in the moving direction, resulting in heat transfer enhancement downstream of the stagnation point.

Gradeck et al.8,10) studied quenching by a subcooled water jet of a hot rotating cylinder with an initial temperature of 500–600°C. The research team showed that cooling rates are largely influenced by the moving velocity of the solid. Recently, Vakili and Gadala11) investigated boiling heat transfer on a hot moving plate, caused by multiple impinging water jets in rows. These researchers found that plate speed has some influence on the heat transfer rates in jet impingement zones. The researchers also pointed out that more experiments are needed with a wider range and variety of plate speeds. Despite the research to date, we still have insufficient knowledge of the heat transfer characteristics when metal sheets are moving.

The objective of this study was to investigate the heat transfer characteristics of a circular jet impinging on a moving hot metal sheet. In the experiments, distilled water at room temperature was used as the test coolant. The circular jet issued from a 5-mm-diameter pipe nozzle. A 0.3-mm-thick sheet made of stainless steel was adopted as the moving sheet. The initial temperature of the solid, the jet velocity, and the moving velocity of the sheet were varied systematically.

Because the heat transfer characteristics are strongly dependent on the local flow structure of the coolant, the temperature profiles of the moving sheet and the liquid flow were observed simultaneously. The liquid film that formed...
on the moving sheet was captured by flash photography, and the temperature distributions on the underside of the sheet were measured by infrared thermography. The heat transfer characteristics were evaluated by solving the heat conduction equation with a finite volume method, using the measured temperature profile as a boundary condition. The effects of the aforementioned parameters on the hydrodynamics and the heat transfer process will be discussed in detail.

2. Experimental

2.1. Experimental Setup

A schematic diagram of the experimental apparatus used in this study is shown in Fig. 1. The apparatus was composed of (1) a water supply system for making a circular jet, (2) a moving test sheet mounted on a motor actuator, and (3) observation equipment.

The water supply system consisted of a gastight water tank, a pipeline, an electromagnetic valve, a subpipeline drain, and a circular pipe nozzle. Distilled water at room temperature was used as the test liquid. The temperature, $T_f$, of the water ranged from 13–18°C. Test water was transported through the pipeline using air pressure. The electromagnetic valve switched the water supply either to the pipe nozzle or to the subpipeline for drainage purposes. Water was allowed to flow out through the drain during experiment preparations and was fed to the pipe nozzle during experiments. The water flow rate was adjusted with a pressure-regulating valve attached between an air compressor and the gastight tank. In addition, the flow rate was measured directly from the volume of water discharged during a sampling period (= 60 s).

The dimensions of the circular pipe nozzle were diameter $D = 5.0$ mm and pipe length $L = 460$ mm. The circular jet issued vertically downward from the pipe nozzle at a mean velocity of $V_0$, which was evaluated based on the volume flow rate of discharged water and the cross-section of the circular nozzle. The mean velocity, $V_0$, was set at 0.4, 0.8, or 1.2 m/s. In addition, the nozzle-to-plate spacing was set at 40 mm.

Figure 2 shows a schematic of the moving test sheet made of stainless steel (SUS430); the sheet is 60 mm wide, 220 mm long, and 0.3 mm thick. The two long sides of the sheet were bent downward at a right angle at 5 mm from the edge to prevent unwanted deformation of the test sheet during the experiments. The two short ends of the sheet were bent downward at an angle of approximately 20° starting at 30 mm from the edge, so that the water could easily flow off of the test sheet during the experiments. On the dry underside of the sheet, a thin coat of black body paint with an emissivity of $\varepsilon = 0.94$ was added to ensure the accurate measurement of temperature with the infrared thermography. In addition, a pair of K-type thermocouples with a wire diameter of 0.3 mm was spot-welded on the underside of the sheet to measure the local temperature and to calibrate the temperature measurement with the infrared thermography. The test sheet was mounted on a linear motor actuator using thermal insulators as shown in Fig. 1. The moving velocity, $V_s$, of the test sheet could be varied from 0 to 1.5 m/s. The upper limit of the sheet speed was owing to the limitation of the linear motor actuator. In the present study, $V_s$ was set to 0.5, 1.0, or 1.5 m/s.

Figure 3 depicts a schematic diagram of the infrared ther-
mography and flash photography. An infrared camera that can take thermal images with a resolution of $320 \times 240$ pixels at 60 frames per second was used to measure the temperature profile of the underside of the test sheet. The infrared camera was set to face vertically upward, with a camera-to-sheet distance of approximately 300 mm. The temperature measured with infrared thermography was calibrated by comparing it to the temperature measured at the thermocouples attached on the reverse side of the sheet. The measurement uncertainty was confirmed to be within $\pm 2^\circ C$.

A digital camera with an effective spatial resolution of $3888 \times 2592$ pixels was used to take photos of the free surface of the liquid film and the boiling phenomenon. The camera was placed above the test sheet as shown in Fig. 3. A strobe light with a light duration below $2 \mu s$ was used as the light source. The free surface images were exposed only when the strobe light was triggered.

2.2. Experimental Procedure

The metal sheet was electrically heated with a DC power supply with peak output of 3 kW ($300 \, \text{A at 10 V}$) until the sheet temperature was approximately $20^\circ C$ higher than the target temperature. Thereafter, the electric power supply to the test sheet was cut. The temperature of the test sheet began to decrease very gradually due to natural convection to the atmosphere. When the sheet temperature reached the target value, the linear motor actuator with the test sheet was activated. The test sheet was accelerated sharply until the moving speed reached a preset velocity. The distance of the initial sheet away from the test section was determined based on the acceleration time.

Immediately after the moving test sheet entered into the test section, the water jet began to collide with the sheet surface. The jet impingement formed a flowing film of water, which soon reached a moderately stable state in the vicinity of the impact point. Several preliminary experiments showed that the film flow was roughly stable when the distance, $L_s$, between the jet impact point and the leading edge of the moving sheet (see also Fig. 3) was more than approximately 75 mm.

Temperature measurements of the metal sheet were carried out by infrared thermography not on the wet surface, but on the underside. If an abrupt temperature change occurs on the wet surface, it is reflected on the underside after some delay. The delay time can be estimated using the thermal diffusivity, $a$, of the plate and the plate thickness, $d$, according to the formula $d^2/a = 0.015$ s. The plate moves 22.5 mm during the delay time when the sheet velocity is 1.5 m/s. Based on this information, the flash photography of the liquid film and the temperature measurement on the underside of the sheet were done in the region of $L_s > 110$ mm.

A small, wavy motion of the free liquid surface was observed even in the region of $L_s > 75$ mm, which probably gives rise to a small time variation in the temperature profile. Because heat is transferred within the sheet as a result of heat conduction, any sharp temperature variation occurring on the top surface of the sheet spreads to the underside. In addition, the infrared camera can take thermal images only at 60 frames per second. Thus, the measured temperature profile in the region of $L_s > 110$ mm was assumed to be steady in the present study.

To evaluate the local temperature profile as well as the heat flux on the top surface, the inverse heat conduction problem should be solved numerically using the measured temperature profile on the underside, as will be explained in a later subsection. To achieve a high-accuracy temperature evaluation on the top surface, a very thin sheet is preferable.

On the other hand, the utilization of a thin sheet presents some experimental difficulties. Because a thin sheet has a small heat capacity, a large variation in the local temperature arises in cases with small heat reductions. If the sheet is very thin, no boiling occurs as a result of the jet impingement, even at high initial temperatures of the solid. The physics of this phenomenon is different from the actual cooling. In addition, a sudden temperature change in the solid leads to local thermal stress, resulting in an undesirable local deformation of the sheet. This unwanted local deformation occurs for thinner plates at higher temperatures. To avoid these difficulties, a certain minimum sheet thickness was needed. After conducting several preliminary experiments, a 0.3-mm-thick sheet was chosen, and the initial temperature of the solid was set to be a maximum of $250^\circ C$. In the present study, the initial temperature of the moving sheet was set to 100, 150, 200, or $250^\circ C$.

Experiments were conducted 10 times for each experimental condition. The averaged local heat flux, $q_v$, at the wet surface and its error bar, $\pm \sigma$, were calculated using the standard deviation of the average value as follows:

$$q_v = \frac{1}{10} \sum_{n=1}^{10} (q_{v,n} - q_v), \quad \sigma = \sqrt{\frac{1}{10} \sum_{n=1}^{10} (q_{v,n} - q_v)^2},$$

in which the index $n$ represents the number of samples. The data reduction procedure will be explained in the next subsection.

2.3. Data Reduction

The temperature distribution of the solid surface on which the circular jet impingement was computed using the temperature profiles at the underside of the solid. The heat conduction equation inside the sheet in the three-dimensional Cartesian coordinate system is given by:

$$\rho_p c_p \left( \frac{\partial T}{\partial t} + \mathbf{v} \cdot \nabla T \right) = \lambda_p \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right) + q,$$

in which $\rho_p$, $c_p$, and $\lambda_p$ represent the material properties of the stainless steel (SUS430) and $t$, $V$, and $q$ denote the time, the moving velocity of the sheet, and the heat generation rate, respectively. The coordinates ($x$, $y$, $z$) are taken in the directions of the length ($x$), width ($y$), and thickness ($z$), as defined in Figs. 3 and 4. The origin is set at the jet impact point on the wet surface. This coordinate system does not move as the sheet moves, but is fixed in space. The material properties were assumed to be constant: the density, $\rho_p$, is 7820 kg/m$^3$; the specific heat, $c_p$, is 461 J/kg·K; and the thermal conductivity, $\lambda_p$, is 25.6 W/m·K.

The temperature profile is assumed to be steady, and no heat generation is present ($q = 0$) during the cooling tests. The following equation was used for data reduction:

$$\rho_p c_p \frac{\partial T}{\partial x} = \lambda_p \left( \frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} \right).$$
The local temperature distribution on the wet surface was evaluated using the method developed by Vader et al.\textsuperscript{13)} Equation (3) was numerically solved by a finite volume approximation. A rectangular computer domain with a length of 75.4 mm, a width of 48.6 mm, and a thickness of 0.3 mm was divided into 140 \( \times \) 90 \( \times \) 200 cells in the \( x \)-, \( y \)- and \( z \)-directions as shown in Fig. 4. The term on the left-hand side was approximated using a third-order upwind scheme and the terms on the right-hand side were done using a second-order central difference scheme. At the side boundaries in \( x \)- and \( y \)-directions, the zero temperature gradients were assumed to be:

\[
\frac{\partial T}{\partial x} \bigg|_{\text{side boundary}} = 0, \quad \frac{\partial T}{\partial y} \bigg|_{\text{side boundary}} = 0. \quad (4)
\]

At the underside boundary, the heat loss from the solid to the surrounding air could be specified in the following form,

\[
q_{\text{bottom}} = \alpha(T_{\text{bottom}} - T_{\infty}) \quad (5)
\]

in which \( T_{\text{bottom}} \) and \( T_{\infty} \) represent the local measured temperature and the atmospheric temperature. The heat transfer coefficient, \( \alpha \), for natural convection was set to 11.6 W/m\(^2\)·K. In addition, the measured temperature profile captured with the infrared camera was employed at this boundary. Because pixel noise introduced error in the measured temperature, it was necessary to apply a smoothing operation to the measured data. A smoothing method based on a least-squares method using spline functions was adopted to reduce the pixel noise.

3. Results and Discussion

3.1. Effect of Varying Sheet Temperature

Figure 5 contains photographs showing the film flows formed by the impingement of the water jet for \( V_0 = 0.4 \) m/s and \( V_s = 1.0 \) m/s. The initial temperature of the solid was varied as a parameter. In Fig. 5(a), for \( T_i = 100 \) °C, a thin liquid film is formed near the impact point. The thin film area is small on the upstream side and expands in the downstream region. A bow-shaped hydraulic jump is observed on the upstream side (see also Fig. 4). The liquid film is thick at its periphery. As expected, the area outside the hydraulic jump is dry. In Figs. 5(b)–5(d), \( T_i = 150, 200, \) and \( 250 \) °C, respectively. In these cases, the initial temperature of the solid is higher than the saturation temperature of water, and boiling vapor bubbles are seen at the liquid/solid interface. In Fig. 5(d), some dry areas appear in the thin liquid film, as indicated by the circle. These areas are formed due to the bursting of boiling vapor bubbles whose sizes are almost the same as the film thickness.

The wet area on the upstream side of the jet is smaller for a higher initial solid temperature. In other words, the front edge of the liquid film is located closer to the impact point. This phenomenon can be explained by the presence of boiling vapor bubbles: The abrupt heating of water at the front edge gives rise to the rapid formation of vapor bubbles, which are transported downstream by the sheet motion. The drag force of the bubbles acting on the liquid works in the direction of motion. Because the bubbles increase in size and number with increases in the initial temperature of the solid, the drag force increases as well. As a consequence, the front edge of the liquid is located closer to the jet impact point. The presence of boiling vapor bubbles also prevents expansion of the width of the thin liquid film. The width of the wet area for 100 °C appears to be larger than that for 200 and 250 °C.

Figure 6 depicts the temperature profiles of the solid on
the underside, as captured by the infrared camera. The experimental conditions are the same as for the previous results in Fig. 5. The color key (legend) varies, depending on the conditions; the black dot in the images represents the jet impact point on the upper surface. The low-temperature areas are observed directly downstream of the jet impact point. Their width is obviously narrower at higher initial temperatures. A comparison of Figs. 5 and 6 shows that the shape of the low-temperature areas on the underside is similar to that of the wet areas (liquid/solid contact areas) on the upper side. However, the position on the underside where the abrupt temperature decrease occurs at the center line is not the same as that of the front edge of the liquid film on the top side. The reason for this is that there is a delay because the temperature change propagates from the wet surface to the underside owing to heat conduction. The test sheet has moved some distance when the temperature change on the wet surface is reflected on the underside.

Figure 7 presents the variation in the local heat flux on the wet surface at the center line of \( y/D = 0 \). The experimental conditions are the same as for the previous results in Figs. 5 and 6. In all cases, the heat flux is very small in the upstream dry region, increases steeply at the front edge of the liquid film where the first liquid/solid direct contact occurs, and peaks in the hydraulic jump region. Because the position of the hydraulic jump shifts toward the jet impact point with increases in the temperature of the solid, the positions showing peak heat flux shift as well. The heat flux is larger for a higher temperature of the solid. In the cases where \( T_s = 150–250°C \), boiling vapor bubbles also enhance the heat transfer, and their influence increases as the temperature increases within this range. In the 100°C case, a second peak is seen near the impact point. In all cases, the heat flux decreases in the downstream region.
3.2. Effect of Varying Sheet Velocity

Figures 8(a) and 8(b) show the observed film flows for \( V_s = 0.5 \) and 1.5 m/s, respectively, under the conditions that \( V_0 = 0.8 \) m/s and \( T_s = 200^\circ C \). The wet area on the front side is smaller at the larger plate velocity; i.e., the front edge of the liquid film shifts downstream with increases in the sheet velocity. This is because there are larger viscous wall friction and drag forces between the cooling water and the boiling vapor bubbles. In the downstream areas indicated by the black squares in Fig. 8, the boiling vapor bubbles can be clearly seen for \( V_s = 1.5 \) m/s. In contrast, they are rarely observed for \( V_s = 0.5 \) m/s due to condensation of the vapor bubbles. At the lower sheet velocity, the liquid/solid contact time becomes longer and the size of the wet area on the front side is larger. As a consequence, the temperature of the solid is reduced, compared to that at the higher velocity.

Figure 9 depicts the thermal images for the same experimental conditions shown in Fig. 8. The color key is the same in both Figs. 9(a) and 9(b). The width of the low-temperature area is narrower for the larger sheet velocity. The temperature appears to be lower in the downstream region for the smaller sheet velocity. The position where the temperature first begins to decrease at the center line \((y/D = 0)\) on the underside is located more upstream for the smaller velocity, for a couple of reasons. The first is that the temperature is related to the position of the front edge of the liquid film, which in turn is affected by wall friction and drag forces. These forces are small when the velocity is low. The second is that the sheet has moved a shorter distance at the lower sheet velocity during the time available for heat conduction from the wet side to the underside.

Figure 10(a) shows the local heat flux for \( V_0 = 0.8 \) m/s and \( T_s = 200^\circ C \) when varying the sheet velocity as a parameter. The heat flux is very small in the dry region, then increases steeply near the front edge of the liquid film before reaching peak values and then decreasing in the downstream region. The position of the peak heat flux shifts downstream with increases in the sheet velocity, \( V_s \). In addition, the heat flux increases with increases in the sheet velocity. Figure 10(b) represents the local heat flux for \( V_0 = 0.8 \) m/s and \( T_s = 100^\circ C \). Here, the heat flux is also larger for a larger sheet velocity. Unlike Fig. 10(a), there are two heat flux peaks. The first set of peaks exists in the front hydraulic jump region, and the second set of peaks is located near the impact point. Such results are obtained when the distance between the front hydraulic jump and the jet impact point is relatively large, with a stable thin film region present between the two.
3.3. Effect of Varying Jet Velocity

Figures 11(a) and 11(b) are the photographs showing the film flows for \( V_0 = 0.4 \) and 1.2 m/s, respectively, under the conditions that \( V_s = 1.0 \) m/s and \( T_s = 200^\circ C \). The wet area is considerably larger for the larger jet velocity, which is associated with the greater impact inertia of the jet. Accordingly, the low-temperature area on the reverse side is wider, and the front edge of the low-temperature area shifts upstream, as shown in Fig. 12.

Figure 13(a) shows the local heat flux for \( V_s = 1.0 \) m/s and \( T_s = 200^\circ C \) when varying the mean jet velocity as a parameter. The peak value of the heat flux shifts upstream as \( V_0 \) increases. In addition, the mean jet velocity has a small effect on the peak value. Figure 13(b) represents the results for \( V_s = 1.0 \) m/s and \( T_s = 100^\circ C \). In all cases, two peaks can be seen. Although the positions of the first peak are dependent on the mean jet velocity, the peak values themselves are very similar for the following reason: The first peaks are located near the front wet edge where the apparent/bulk velocity of the liquid in the x-direction is very small. Because the sheet velocity and the initial temperature of the solid are identical in all cases, the flow and temperature profiles are probably similar in the region. As a consequence, similar peak heat fluxes were obtained.

3.4. Correlations for Predicting Peak Heat Flux

In the present study, the effects on the heat transfer characteristics of varying three parameters were investigated; namely, the initial temperature of the solid, the moving velocity of the solid, and the mean jet velocity. The authors undertook the development of a correlation for predicting the peak heat flux near the front edge of the wet area as functions of these parameters and obtained the following equation:

\[
q_{\text{peak}} = 189(T_s - T_w)_{1.00}^{0.47}V_s^{0.09}V_0^{0.99} \text{ W/m}^2
\]

Every coefficient was determined by least-squares curve fitting to the experimental results. Figure 14 shows a comparison of the predictions with the experimental results; the

![Figure 11](image1)

![Figure 12](image2)

![Figure 13](image3)

![Figure 14](image4)
predictions agree moderately well with the experimental data. The equation was built based on a trial-and-error method and has no theoretical background. However, we can examine which parameter is dominant in determining the peak heat flux. Judging from the values of the components in Eq. (6), the local heat flux is strongly dependent on the temperature difference between the initial temperature of the solid and the coolant, and is moderately dependent on the moving velocity of the sheet. In contrast, the jet velocity has little influence on the peak heat flux, as already discussed in the previous subsection.

The heat flux is considered to be dependent not only on these parameters, but also on other parameters, including the nozzle diameter, the nozzle-to-sheet distance, and so on. To obtain additional useful data, further experiments should be done under a wider variety of conditions, including much wider temperature ranges, larger jet velocities, and faster sheet velocities. These are all areas for future work.

4. Conclusions

The flow structure and heat transfer of a circular water jet impinging on a heated moving surface were investigated experimentally. The effects on the heat transfer characteristics of varying the initial temperature of the solid, the sheet velocity, and the mean jet velocity were examined. The main results obtained in this study are summarized below.

1) The flow motion of the coolant was observed by flash photography and the temperature profile of the moving solid was measured by infrared thermography. This observation technique was quite useful to understand the heat transfer characteristics.

2) The location of the front edge of the liquid film depends on the temperature of the solid, the sheet velocity, and the mean jet velocity.

3) The local heat flux at $y/D = 0$ is very small in the dry area, increases steeply near the front edge of the liquid film, and reaches a peak value. If the distance between the front edge of the liquid and the jet impact point is relatively large, a second peak appears near the jet impact point. The heat flux decreases in the downstream region.

4) An experimental correlation was developed for predicting peak heat fluxes near the front edge of the liquid, although it has no theoretical background. The correlation agrees moderately well with the experiments.

REFERENCES