Spray and jet cooling in steel rolling

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Prediction and control of roll and strip cooling are necessary in modern steel mills because they not only affect the process efficiency but also strongly influence the quality of rolled products. In this article, relationships among metallurgy, heat transfer, and control of the cooling system in steel rolling are first discussed. Heat transfer characteristics associated with the water spray and jet cooling used in rolling processes are then studied. The effects of important convective heat transfer parameters on cooling performance for both stationary and moving surfaces are examined. Results indicate that local heat fluxes up to $20 \times 10^6\ W/m^2$ are observed in the nucleate boiling regime. The present results are compared with typical boiling heat transfer studies in terms of heat fluxes, heat transfer coefficients, spray rate, and cooling efficiency. The effect of surface motion is found to increase the cooling efficiency of roll and strip cooling. Finally, implementation of the present finding in roll and strip cooling to thermomechanical processing in steel rolling is proposed.

Keywords: boiling; cooling; finite difference; heat transfer measurement; rolling; steel; water spray; water jet

Introduction

In metal rolling processes, the roll is used as a tool to deform the strip at very high speeds. As shown in Figure 1, heat transfer phenomena involved in the process are complicated by the interaction between the roll and strip, the heat removed from the roll by the coolant, and the heat removed from the strip by the cooling systems at the runout table. These processes occur at high pressure, high velocity, and high heat flux; these make the prediction and control of the roll and strip temperatures an extremely challenging heat transfer problem. In this article, the strip is used as a general term for the rolled product, such as sheet, strip, plate, and slab.

Knowledge of spray and jet cooling of the roll and strip can contribute to insights about the thermomechanical processing of the workpiece and lubricant behavior and can lead to the desired mechanical properties and surface condition of the rolled strip. Also, proper cooling can lead not only to extended roll life by minimizing the thermally induced stresses but also to improvements in the strip shape quality by controlling the thermal crown (Tseng et al. 1989; Bennon 1985). Recently, spray and jet cooling has also been applied to interstage cooling, as well as cooling of coils (Mazur et al. 1989). Therefore, adequate cooling of the roll and rolled strip is of critical concern to mill designers and process engineers. In order to study the influence of cooling practices on the roll and strip, a good understanding of the heat transfer aspects of the process is essential.

In spite of the great practical importance of this subject and of the many cooling systems being employed in steel mills, the fundamental heat transfer phenomena of roll and strip cooling still are not fully understood. It is also to be noted that many mathematical models have been developed to simulate the thermal behavior of the rolling process. The fundamental law governing the energy conservation is believed to be well understood. However, the difficulty in most models is in accurate representation of the boundary condition. It is found that the equivalent heat transfer coefficient of cooling is the most important input information in these models.

In this article, the importance of roll and strip cooling on the thermomechanical processing is first discussed. A review of the recent heat transfer research of the cooling technology in the steel rolling process is then presented. The basic heat transfer modes involved in the roll and strip cooling are included in the review. Combined experimental and numerical methods for determining heat transfer characteristics for various cooling conditions are then reported, and typical results are presented. Based on the present study, suggestions regarding design and control of cooling systems are then given. Finally, areas where further work is needed are identified.

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Thermomechanical processing and microstructures

Originally, rolling processes were merely used to attain the desired dimensions of the workpiece. However, since the 1950s, it has been realized that the physical properties of the product can be markedly affected by changing the finishing temperature. A procedure to achieve the desired finishing temperature within the austenite range, which is about 200 °C lower than the normal or conventional finishing temperature, is referred to as "controlled rolling." Typical experimental data indicate that an increase of 100 °C in the finishing temperature raises the transition temperature by about 4 °C in the Charpy test (Irvine et al. 1970; Heedman et al. 1981; Polukhin et al. 1986) and 5 °C in tear tests (Roberts 1983). In general, a lower finishing temperature raises the yield strength of the rolled strips, but has no major effect on the ultimate strength or elongation (Heedman et al. 1981; Roberts 1983).

Recently, a technique known as "controlled cooling" using the accelerated cooling concept has been developed to lower the strip temperature of the runout table. Because the basic rolling practice remains unchanged as in the controlled rolling,

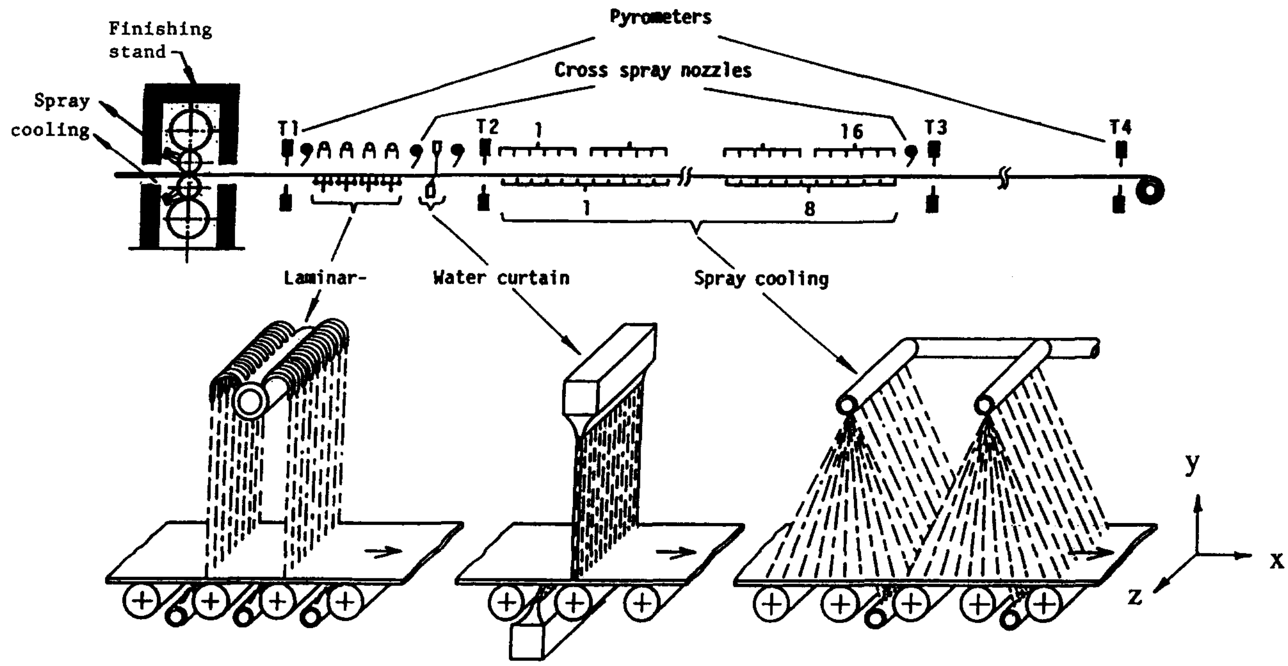

Figure 1 Cooling systems at the runout table in a typical hot rolling mill

the controlled cooling technique has gained a rapid acceptance by the rolling industry since its inception in the 1960s. The term "ThermoMechanical Processing" (TMP) has been applied to such practices.

Controlled cooling or TMP not only eliminates the troublesome differences in strength among samples taken from the head, body, and tail of a coil but also increases the strength and fracture resistance of steel strips. The result is a continuous process for producing steel strips that have properties in the as-rolled condition that equal or exceed those of regular heat-treated strips. Moreover, TMP or controlled cooling has also the advantage of reducing the alloy content required to achieve a specified property level; in general, the lower alloy content improves the weldability of the steel. Therefore, steel strips are not only cheaper to make but also cheaper to fabricate in subsequent operations.

The role for controlled cooling is to produce a high-density nucleation site for ferrite (α) grains in the austenite (γ) matrix during transformation by controlling the hot rolling conditions, so as to refine the structure after transformation. The effect of controlled rolling and accelerated cooling on microstructures of the rolled strips can be interpreted using the continuous cooling diagram as shown in Figure 2. As shown, the controlled cooling process changes the ordinary α-pearlite structure to the α-bainite structure with very fine α grains.

The accelerated cooling process is characterized by a $\gamma \rightarrow \alpha$

Notation

A	Cooling surface area
B	Strip or plate width, m
C_p	Specific heat of the strip or roll
d	Diameter of the circular (round) nozzle
e	Heat generation rate due to deformation and/or phase transformation, W/m^3
h	Local heat transfer coefficient, $h = q/(T_s - T_c)$, W/(m$^2\cdot$K)
$\bar{h}$	Average heat transfer coefficient, $h = q/(T_s - T_c)$, W(m$^2\cdot$K)
H	Strip or plate thickness
k	Thermal conductivity of the strip, W(m$\cdot$K)
Δl	Length of the cooling area at the runout table
m_s	Production rate of the rolled piece (kg/s)
m	Mass flow rate of coolant (kg/s)
q	Local heat flux, $q = h(T_s - T_c)$, W/m^2
$\bar{q}$	Average heat flux, $q = h(T_s - T_c)$, W/m^2
Q	Overall heat transfer rate, W
r	Distance in the radial direction (Figure 5)
S	Slot width of the planar (slot) jet (Figure 14b)
S_L	Distance between planar jets (Figure 14a)
S_n	Distance between tubes of circular (round) jets (Figure 14a)
S_p	Distance between rows of circular (round) jets (Figure 14a)
t	Time
Δt	Time required for the strip to travel through the cooling area
T	Strip or roll temperature
T_c	Coolant temperature
T_s	Strip or roll surface temperature
T_{sat}	Saturation temperature of coolant
u	Strip moving speed (m/s)
x, y, z	Eulerian coordinates, defined in Figure 1
W	Coolant spray rate or flux, kg/(m$^2\cdot$s)
V_j	Jet or spray velocity of coolant (m/s)

Greek symbols

α	Thermal diffusivity of the strip or roll, m^2/s
η	Cooling efficiency, kJ/(kg of coolant)
ρ	Density of the strip or roll

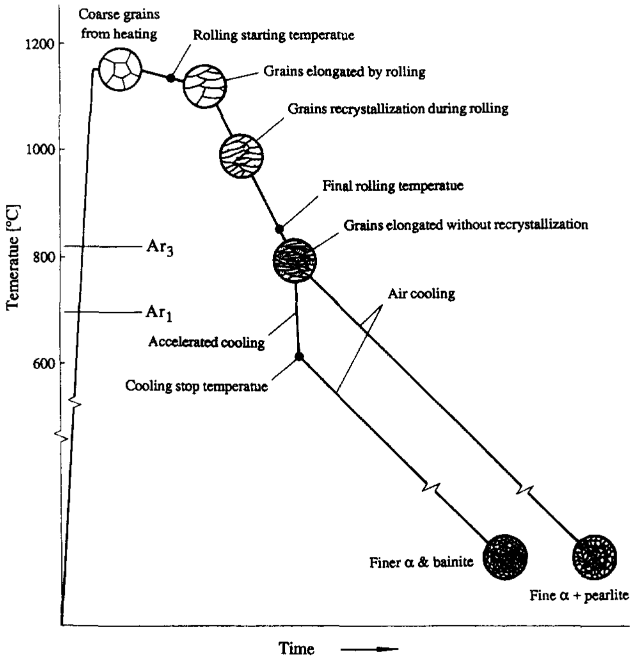

Figure 2 Schematic illustration of controlled rolling with and without accelerated cooling

transformation in a starting temperature just above the A_{r3} temperature (Roberts 1983). In accelerated cooling, the steel may be strengthened through the suppression of the process of static recovery and recrystallization of deformed α. Thus, the deformed α and the nondeformed α on the substructure of bainite is finally formed. Distl, Kaspar, and Zeislmair (1987) studied three high-strength low-alloy (HSLA) steels with a finish temperature at 820 °C and observed that the yield and ultimate tensile strength increased by 50–80 MPa at a cooling rate of 15 °C/s compared with the plates that merely come from controlled rolling. The transition temperature in Charpy testing is slightly improved by accelerated cooling. Tamura et al. (1988) obtained similar findings for two other steels. These findings concluded that the yield and ultimate tensile strengths were elevated by 30–60 MPa at a cooling rate of 10 °C/s while the accelerated cooling, starting at 800 °C, was stopped at 600 °C, which was followed by air-cooling. In fact, a large amount of research has been reported in controlled rolling and cooling (Roberts 1983; Distl et al. 1987; Tamura et al. 1988).

The major differences for processing parameters between controlled cooling and direct quenching are the cooling rate and the temperature range of cooling. Controlled cooling, in principle, tends to choose the optimum cooling rate applicable to a wide variety of as-hot-rolled steel products without any heat treatment after cooling, whereas direct quenching aims to obtain low-temperature transformed products such as martensite or bainite, which necessitates the subsequent tempering treatment. In direct quenching, the water cooling is also stopped at a specific temperature, then followed by air-cooling. The cooling rate for direct quenching can be as high as 50 °C/s for a 20-mm steel (Ohtomo et al. 1988). Both controlled cooling and direct quenching are being widely adopted in the plate, pipe, and bar mills. Recently, Kawano et al. (1990) reviewed the cooling technology developed in Japan and concluded that control of the coiling temperature and cooling rate is a cost-effective way to control the microstructural transformation and thus to improve mechanical properties of hot rolled high-strength steel.

Cooling and control systems

Various on-line cooling systems have been developed for both the controlled rolling and controlled cooling processes (Figure 1). For cooling and lubricating of the rolls, water spray is generally used. The spray is also applied for descaling purposes, i.e., removal of fine oxide particles that are generated in the hot rolling of strip. Numerous types and sizes of spray nozzles are commercially available. For roll-cooling applications, a flat, fan-shaped spray pattern is usually preferred (Tseng et al. 1989).

In runout table cooling, water is applied to both the top and bottom surfaces. Water to the bottom surface falls away immediately and has a short residence time. Water on the top surface has a significant residence time and, consequently, is more effective. Thus, more flow is needed from the bottom (about 1.5 times that from the top surface) in order to maintain symmetric cooling. Water spray is often used to cool the bottom surface and to sweep the top surface for the purpose of pyrometry measurement.

Recently, a laminar-flow system shown in Figure 1 has been developed for the runout table cooling. It consists of staggered arrays of round nozzles that create low-pressure, laminar-flow jet streams. The laminar flow is desired because it can penetrate the vapor film on the cooled surface and increase the coolant residence time. Since splashing is minimized and contact of water with the strip is improved, the laminar-flow system has higher cooling efficiency than the water-spray system (Southwick 1986; Ruddle et al. 1988).

More recently, a planar jet called the *water-curtain system* has been adopted for accelerated cooling applications. It spans the entire width of the strip, as shown in Figure 1. In addition to the advantage of laminar jet cooling, it also provides uniform cooling in the transverse (lateral) direction of the strip. As reported by Koring (1985), water-curtain jets are about 50 percent more efficient than spray nozzles in removing heat from the strip. Consequently, the water-curtain system has been adopted by many remodeled and newly built steel mills.

As demonstrated in Figure 2, the microstructures and mechanical properties of the rolled strips are strongly dependent on the finishing and coiling temperatures as well as the cooling rate at the runout table, i.e., the time–temperature profiles. Therefore, adequate control of roll and strip cooling requires the achievement of the following objectives:

(1) to track the desired time–temperature profiles;
(2) to achieve the desired finishing and coiling temperatures;
(3) to minimize temperature gradients in the strip (in the lateral and depthwise directions); and
(4) to minimize the coolant consumptions.

While the strip finishing temperature is controlled by roll cooling (Tamura et al. 1988), the coiling temperature and heat removal rate are controlled by accelerated cooling at the runout table. Much development has been focused on the control of coiling temperatures using feedforward and feedback control (Hinrichsen 1976; Moffat et al. 1985). Since heat transfer behavior is not well defined in controlled cooling, the heat removal rate was estimated in feedforward control and compensated in feedback control. To account for the great uncertainty in cooling efficiency, Groch et al. (1990) further improved the automatic temperature-control system by including an adaptive algorithm, which can estimate the cooling efficiency according to on-line temperature measurements coupled with a thermal model. Recently, Chen et al. (1990) modeled an optimal feedforward control system by considering the temperature gradient in the depthwise direction, as well as coolant consumptions.

In order to produce high-quality rolled steel, it is important to accurately predict, monitor, and control rolling processes. As shown in Figure 3, a successful development of controlled rolling and controlled cooling requires the integration of thermal, metallurgical, and control engineering. Knowledge of roll and strip cooling is critical in determining the thermal models for feedforward, feedback, and statistical process control strategies.

The typical operating ranges used for cooling control in hot rolling of steel can be summarized as follows:

- finishing strip temperature: 700–1100 °C;
- coiling strip temperature: 400–700 °C;
- cooling rate in controlled cooling: 5–150 °C/s;
- cooling efficiency: 10–400 kJ/kg;
- heat fluxes: 10^5–10^7 W/m^2;
- heat transfer coefficients: 10^3–10^5 W/m^2 – K
- strip speed: 5–15 m/s.

Previous work in spray and jet cooling

In roll and strip cooling, heat transfer occurs as a combination of three modes of heat transfer, i.e., radiation, convection, and conduction. As shown in Figure 4, heat transfer mechanisms include the heat generated in the strip by deformation, and the heat created at the roll/strip interface by friction. Heat transfer mechanisms at the roll include the heat removed by the coolant (the heat removed by the water film due to the water spread and surface rotation), the heat removed by the ambient, and the heat conduction to the backup roll. The mechanisms at the strip include the heat generation due to the phase transformation (e.g., from austenite to ferrite or pearlite), the laminar jet cooling (cooling due to the runoff liquid and vapor on the strip surface), spray cooling (or water curtain) from the bottom surface, the conduction to the rollers, and the heat removal by the ambient via convection and radiation.

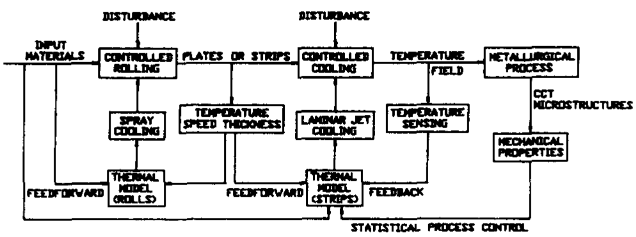

Figure 3 Processing diagram of controlled rolling and controlled cooling in steel rolling

Based on the studies summarized by Roberts (1988), Spalding and Afgan (1981), and Tseng et al. (1987), the surface temperature, heat density, heat flux, and heat transfer coefficient associated with each heat transfer mode are given in Table 1. The use of spray in roll cooling or of jet in strip cooling is a complicated process involving *spray or jet* impingement on a *moving* surface (roll or strip). While the roll or strip speed often exceeds the coolant velocity and strongly influences the flow pattern developed, the surface temperature is often higher than the saturation temperature of the coolant, and *boiling* occurs. The boiling phenomenon may be further complicated by its dependence on surface temperature, and can be identified as nucleate, transition, or film boiling. Surface conditions, such as roughness and oxidation, also affect heat transfer behavior. Since their effects on heat transfer have become major research subjects as discussed by Roberts (1983, 1988), they are beyond the scope of this article. Furthermore, the heat transfer characteristics may be also affected by many other parameters, including *thermophysical* properties of the coolant, *geometric* parameters of the cooling system, and *operating* parameters in rolling.

For a given cooling system such as water spray, circular jets, or planar jets as shown in Figure 1, the heat transfer behavior is dependent on the following parameters:

(1) the spray rate or jet velocity (the effect of the Reynolds number);

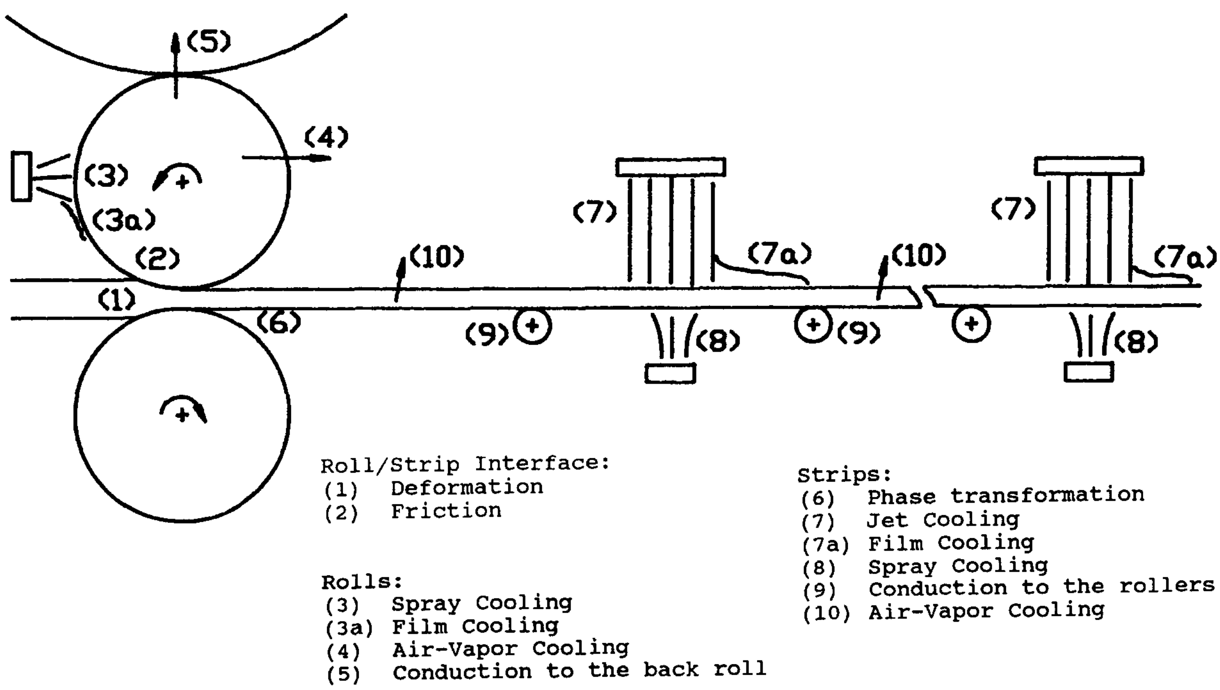

Figure 4 Heat transfer characteristics in roll, strip, and roll/strip interface

Table 1 Heat transfer characteristics in rolls and strips

	Heat transfer mode	Surface temperature (°C)	Heat density (W/m^3)	Heat flux (W/m^2)	Heat transfer coefficient (W/m^2 K)
Roll/strip interface	(1) Deformation	900–1100	10^{10}–10^{11}	—	—
	(2) Friction	500–1000	—	10^7–10^8	10^5–10^6
Rolls	(3) Spray cooling	80–200	—	10^6–2×10^7	5×10^3–10^5
	(3a) Film cooling	80–150	—	10^5–10^6	10^2–10^4
	(4) Air-vapor cooling	150–550	—	10^3–10^4	10^1–10^2
	(5) Conduction to the back roll	100–300	—	10^3–10^4	10^2–10^3
Strips	(6) Phase transformation	900–1100	10^8–10^9	—	—
	(7) Jet cooling	100–800	—	10^6–2×10^7	10^3–10^5
	(7a) Film cooling	400–1000	—	10^5–10^6	10^5–5×10^3
	(8) Spray cooling	400–1000	—	10^6–2×10^7	10^3–2×10^4
	(9) Conduction to the rollers	400–1000	—	10^3–10^4	10^2–10^3
	(10) Air-vapor cooling	400–1000	—	10^4–10^5	10^1–10^2

(2) the water temperature (the effect of subcooling);
(3) the surface temperature (the effect of superheating, $T_s - T_{sat}$); and
(4) the speed of surface motion (the effects of the roll rotating speed and the strip moving speed), where the effect of thermophysical properties on heat transfer is included in the water and metal temperatures.

In spray cooling, the spray rate (W) is often used as a parameter in determining the average heat flux. The correlations summarized by Morales, Lopez, and Olivares (1990) indicate that average values of q are proportional to W^n where $n = 0.5$–0.8. In jet-impingement cooling, heat transfer in single phase and film boiling is proportional to V_j^n where $n = 0.50$–0.65. In nucleate boiling, heat transfer is not appreciably affected by the jet velocity, although the critical heat flux is proportional to $V_j^{1/3}$ (Katto 1985). In the case of a high-pressure spray, it has been found that high pressure and high jet velocity deteriorate cooling performance, due to water bouncing off the cooled surface and the inherent instability of the spray.

The effect of the coolant temperature may be represented by the degree of subcooling ($T_{sat} - T_c$). Several steel mills have reported that lowering the coolant temperature significantly increased the heat removal rate at the runout table (Moffat et al. 1985; Groch et al. 1990). In transient heat transfer studies, Ishigai et al. (1978) using a planar jet and Ochi et al. (1984) using a circular jet reported that increasing subcooling from 5 to 55 K increased local heat fluxes by more than one order of magnitude in the transition and film boiling regimes ($T_s - T_{sat} = 200$–800 K).

The effect of surface temperature is characterized by the degree of superheating, $T_s - T_{sat}$, which determines the boiling regime. Typical studies concerning boiling heat transfer include critical heat flux in nucleate boiling by Katto (1985), transition boiling by Kalinin, Berlin, and Kostiouk (1987), and spray film boiling by Deb and Yao (1989). The effect of increasing subcooling and impacting velocity was found to shift the boiling curves (see Figure 13) to higher temperatures and higher heat fluxes. Among the four parameters mentioned above, the effect of surface motion was the least studied until recently. Data from Monde and Katto (1978) and Ishigai, Nakanishi, and Ochi (1978) are for cooling of stationary surfaces using circular jets and planar jets, respectively.

The authors (Tseng et al. 1987) reviewed studies for single-phase jet impingement on rotating surfaces and found that increasing rotating speed beyond jet velocity generally enhances heat transfer. In an experimental study of heat transfer on a moving plate, Zumbrunnen et al. (1990) reported that peak h values occur upstream of the stagnation point due to nucleate boiling, but only one moving speed was considered. The effect of surface motion was found to enhance heat transfer in studies by Roberts (1988) and Chen, Kothari, and Tseng (1991) for spray cooling on rolls and jet cooling on strips, respectively; both studies include cases with and without boiling.

In order to increase the understanding of roll and strip cooling, the effects of surface motion on temperature distribution and on heat transfer of the roll and strip are studied further and presented in the next two sections, respectively. As mentioned earlier, the controlled rolling technique is mainly manipulated by the roll cooling, while the controlled cooling is regulated by both the roll and the strip cooling.

Further study of spray (roll) cooling

A combined experimental and numerical approach has been developed to study the local heat transfer of water spray on a rotating cylinder to simulate the roll cooling. As mentioned earlier, for cooling of the rolls, water spray is generally used. In the present study, an elliptical nozzle with a hydraulic diameter of 3.9 mm is used. Ten 0.127-mm (0.005-in.) type-K (chromel-alumel) thermocouples were installed on the roll surface. Based on the heat transfer coefficient, which is on the order of 100 kW/m^2-K, the corresponding thermocouple response time was found to be of the order of 1 ms. A detailed description of the experimental apparatus is given by Tseng et al. (1989).

The surface temperatures at several axial and circumferential locations were recorded using a PC/AT-based data acquisition system. The sampling rate was adjusted for each rotating speed to give 300 readings per cycle. The roll surface temperature (T_s) versus the angular position (θ) is recorded and used as the boundary condition for a numerical model previously developed to calculate the heat transfer coefficient and heat flux.

Numerical model for data reduction

The finite-difference technique is used to develop the numerical model. The basic governing equation for the model is

$$\rho C_p\left(\frac{\partial T}{\partial t} + \omega\frac{\partial T}{\partial \theta}\right) = \frac{1}{r}\frac{\partial}{\partial r}\left(kr\frac{\partial T}{\partial r}\right) + \frac{1}{r^2}\frac{\partial}{\partial \theta}\left(k\frac{\partial T}{\partial \theta}\right) + \frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right) \qquad (1)$$

where T is the temperature; r, θ, and z are the radial, circumferential, and axial directions, respectively, of a cylindrical coordinate with respect to a Eulerian reference frame; t is time; ω is angular velocity; and ρ, k, and C_p are density, thermal conductivity, and specific heat of the roll, respectively. Since the temperature also decreases in each of the subsequent cycles, the term $\partial T/\partial t$ is included. This equation is subject to the boundary condition at the roll surface ($r = R$),

$$T(t_i, R, \theta, z_j) = T_{ij}(\theta), \qquad 0 \leqslant \theta \leqslant 2\pi \qquad (2)$$

where $T_{ij}(\theta)$ is the modified measurement temperature at a specific discrete time, t_i, and axial location, z_j.

The corresponding local heat transfer coefficient (h) or heat flux (q) of the cooling can be found from the computed temperature distributions:

$$q(t_i, \theta, z_j) = -k\partial T(t_i, R, \theta, z_j)/\partial r \qquad 0 \leqslant \theta \leqslant 2\pi \qquad (3)$$

and

$$h(t_i, \theta, z_j) = [k\partial T(t_i, R, \theta, z_j)/\partial r]/[T_{ref} - T(t_i, R, \theta, z_j)] \qquad 0 \leqslant \theta \leqslant 2\pi \qquad (4)$$

where T_{ref} is the reference temperature that may be the coolant temperature (in the spray region) or ambient temperature (in the air cooling region), and $\partial T(t_i, R, \theta, z_j)/\partial r$ is computed from the finite-difference model. Results presented in this article are based on the measurement data at one axial location; consequently, only a two-dimensional (2-D) model is needed in computation.

It is to be noted that, as shown in Figure 3, thermal models are very crucial in the prediction and control of rolling processes. When the thermal models are coupled with stress models, metallurgical models, and roll crown and flatness models, they may be used to construct a complete mathematical model for the process control in steel rolling. However, in modeling, the outer boundary conditions that require distributions of convective heat transfer usually are not well known. In boiling heat transfer, where h is a function of the surface temperature (T_s), $q(T_s)$ is more often used in describing convective heat transfer behavior.

Results

The results for the case of the axial position of $z/d = 0$ and $\omega = 58$ rpm are reported in a form of the roll surface temperature (T_s) versus the angular position (θ), as shown in Figure 5. As shown, the surface temperature drops sharply within the spray region; outside the region, it rises again in each cycle. However, the peak temperatures gradually decrease from cycle to cycle. The corresponding roll and spray Reynolds numbers are 27,500 and 35,300, respectively. The nozzle used is Type H1/4U-2540 Vee Jet Nozzle by Spray Systems of Wheaton, IL. It is a typical type designed for steel rolling mills and has an elliptic opening with a major axis of 4.45 mm and a minor axis of 3.43 mm. It is located 165 mm away from the roll surface, providing a fan-shaped spray impinging perpendicularly on the surface; the area directly covered by the spray for the case considered is about 100 mm wide and 7 mm thick for a flow pressure at 500 kPa. The water spray diverges from a small opening to an area 60 times larger, thereby increasing the surface area available for cooling.

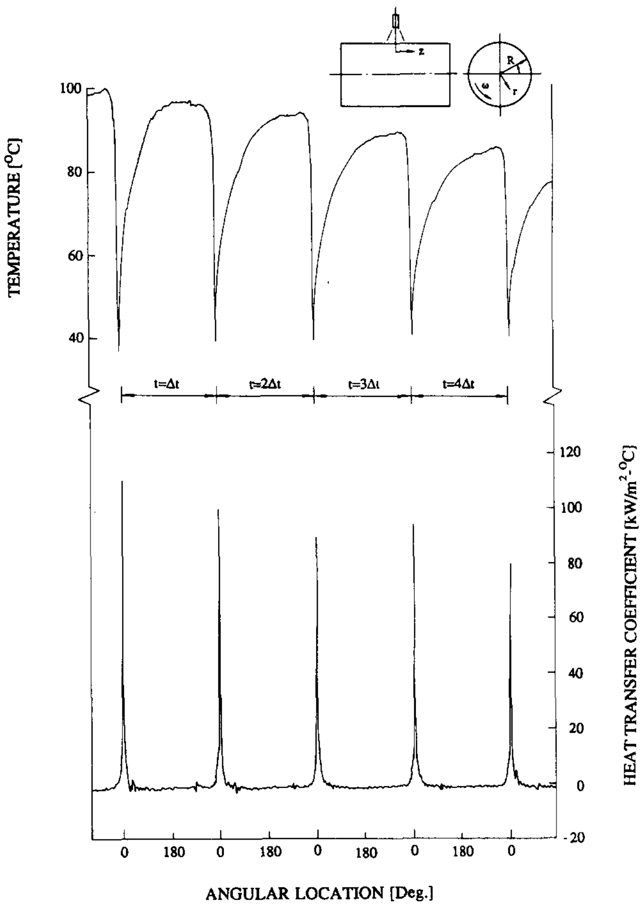

Figure 5 Measured surface temperature (T_s) and calculated local heat transfer coefficient (h) for $z/d = 0$ and $\omega = 58$ rpm for the case of surface temperature below boiling point

As shown in Figure 5, the peak heat transfer coefficients vary from 80 to 110 kW/m^2-K under the impingement center. The variation of the peak values may be due to the inherent instability of spray from the nozzle. The nozzle flow instability caused by supply pressure fluctuations as well as trapped air bubbles creates a waving motion of the spray sheet (flow). If the spray flow does not impinge on the roller, this instability would lead to the atomization of the liquid sheet, as required for other applications, such as spraying pesticides, paint, etc. Therefore, the spray tip is not stable, and its impingement location and intensity on the roll surface fluctuate as a result of this instability. Variations of heat transfer at the thermocouple position occur because the instantaneous impingement position and intensity change from cycle to cycle. Also, the huge divergence of the spray flow pattern may also elevate the variation of heat transfer.

In the case of spray on a stationary surface, the area directly under the spray (about 7 mm thick, as mentioned earlier) covers an angle about 3°. The heat transfer coefficients for each cycle indicate that intensive cooling spreads out an angle about 50° due to surface motion; outside the cooling region (outside the 50° cooling zone), heat transfer that involves only the convection between the surrounding air and roll was neglected in the computation, since its coefficient is several orders of magnitude smaller than that of the water impingement. Also, outside the cooling region, the cooling may continue for a total angle of 180° due to water film staying on the roll surface.

For the case of higher surface temperatures (say, 30 °C higher than the water saturation temperature), the results are shown in Figure 6 for $\omega = 58$ rpm at $z/d = 0$. As shown, the surface temperature under impingement drops more than that of Figure 5, and recovers fully in each cycle. The full recovery is mainly due to the fact that the inner region is still relatively hot and heat is transferred from there to the surface. As a result, it causes the surface temperature to rise. It is to be noted that, from the numerical results, the temperature in the inner region at that moment can be found to be about 120 to 130 °C. When the surface temperature rises to about 100 °C, extra evaporation (or latent) heat is needed to boil off the residual water adhering to the surface. However, the inner region is not hot enough, and therefore cannot transfer enough heat to further raise the temperature before the next impingement (cycle) comes. This phenomenon can be observed much more clearly in Figure 7 for the case in which the inner temperature is much higher.

The peak heat transfer coefficients in Figure 6 vary from 150 to 200 kW/m^2-K under the impingement center. Again, the variation is believed to be mainly caused by the transient and instability behavior of the spray. As shown, tne surface temperatures drop to the neighborhood 50 °C. The peak heat flux is found to be 10×10^6 W/m^2 and as much as twice as high as that of Figure 5.

At even higher surface temperatures, boiling was clearly observed. Figure 7 shows the roll surface temperature and heat transfer coefficient for the same rotating speed under the same location. The surface temperature drop is more than that in Figures 5 and 6 because the initial roll surface temperature is increased a great deal. The peak heat transfer coefficients vary from 140 to 200 kW/m^2-K; the variation is much larger than that shown in either Figure 5 or Figure 6. In the first six cycles, it occurs at the roll surface temperature in the range of 100 to 190 °C, which is in the nucleate boiling regime. Although in the last few cycles the surface temperature drops at the end below 100 °C briefly, it does not affect the intensive boiling that occurs on the surface. It is believed that the increase of the variation of peak h is contributed by the unstable phenomena of nucleate boiling. Because of the combination of the instabilities of water spray and nucleate boiling, the variation of peak h should be higher than that of Figure 5 or Figure 6. The peak heat flux in the present case is about 20×10^6 W/m^2, which is twice as high as that of Figure 6 and four times as high as that of Figure 5.

It is understood that, away from the center of the nozzle, the planar condition cannot be assumed and the temperature drop is greatly reduced, as reported by Tseng et al. (1990). At $z/d = 6.65$ or $z = 25.4$ mm, the peak heat transfer coefficient can drop 50 percent in comparison with the value at the center for $\omega = 58$ rpm.

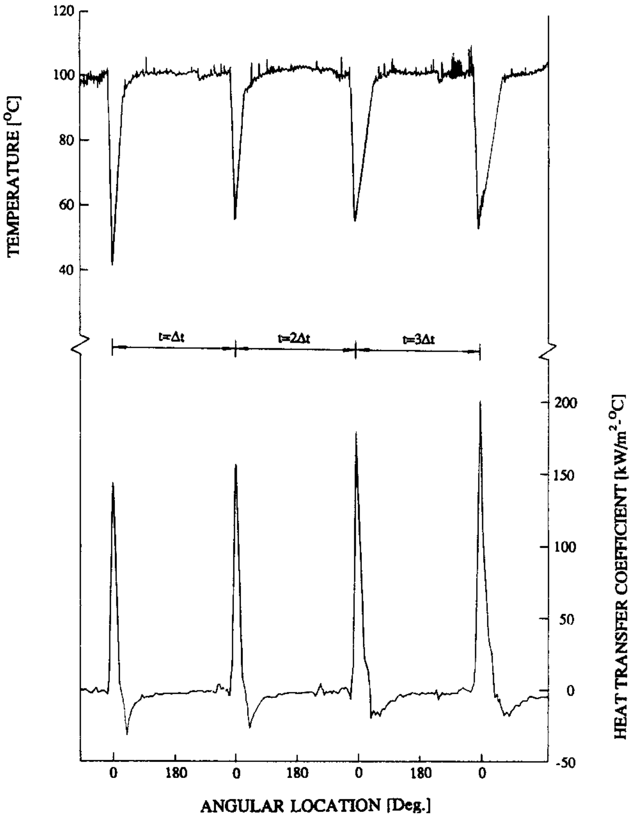

Figure 6 Measured surface temperature (T_s) and calculated local heat transfer coefficient (h) for $z/d = 0$ and $\omega = 58$ rpm for the case of surface temperature near boiling point

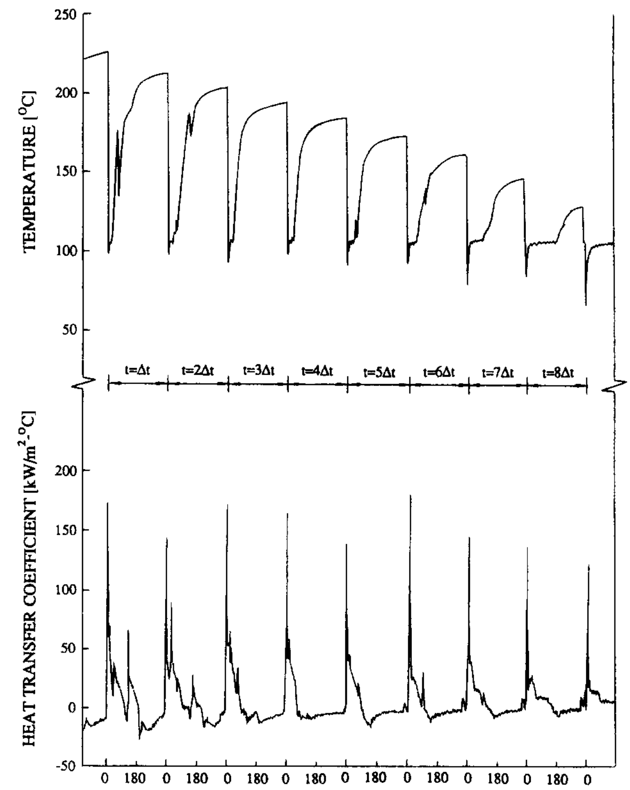

Figure 7 Measured surface temperature (T_s) and calculated local heat transfer coefficient (h) for $z/d = 0$ and $\omega = 58$ rpm for the case of surface temperature above boiling point

Further study of jet (strip) cooling

As shown in Figure 1, we consider strips at the runout table cooled by water spray, arrays of circular (round) jets, or an array of planar (slot) jets. In this section, the local heat transfer for a water jet on a moving plate to simulate the strip cooling is studied. The method developed for studying strip cooling is similar to that for roll cooling. In this study, six type-K coaxial film thermocouples at different lateral positions (z) were used to measure the instantaneous strip surface temperatures. The measured surface temperatures were then used as the boundary condition for the numerical model for data reduction. A detailed description of the experimental apparatus is provided by Chen et al. (1991).

Numerical model for data reduction

Using an Eulerian coordinate system, the heat transfer equation to simulate the steel plate used in the jet cooling experiments is

$$\rho C_p\left(\frac{\partial T}{\partial t} + u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y}\right) = \frac{\partial}{\partial x}\left(k\frac{\partial T}{\partial x}\right) + \frac{\partial}{\partial y}\left(k\frac{\partial T}{\partial y}\right) + \frac{\partial}{\partial z}\left(k\frac{\partial T}{\partial z}\right) + e \tag{5}$$

where u is the strip moving speed in the x-direction; v is the strip speed in the y-direction (important in the roll/strip interface, but negligible at the runout table); and e is the heat generation rate due to phase transformation. The term $\partial T/\partial t$ is often considered negligible for cooling of moving strips at constant speed. Also, in the present study, the heat conduction in the moving direction is neglected due to the high Peclet number. This equation is subject to the boundary condition at the strip surface:

$$T(t_i, x, H, z_j) = T_{ij}(x) \tag{6}$$

where $T_{ij}(x)$ is the measurement temperature at a specific discrete time, t_i, and transverse location, z_j. With a more general boundary condition, the present model can also be used to simulate the thermal behavior of the strip at the runout table:

$$k\partial T_s/\partial n = h(T_{ref} - T_s) + q_f \tag{7}$$

where $\partial/\partial n$ represents differentiation along the normal of the strip boundary (positive outwards); subscript s represents the surface value; and q_f is the heat flux due to radiation.

While Equations 5 to 7 are often used for predicting temperature distributions in the strip, they are too complex for the design of on-line control system. As a result, only the coiling temperature has been controlled in most accelerated cooling systems in hot strip mills (Hinrichsen 1976; Moffat et al. 1985; Groch et al. 1990). In this respect, a simpler model than Equations 5 to 7 based on a lumped system approach may be adopted:

$$Q = m_s\int C_p(T)dT \approx m_sC_p(T_1 - T_2) \tag{8a}$$

and

$$m_s = \rho uHB \tag{8b}$$

where Q is the heat removal rate; m_s is the steel production rate (kg/s); C_p is the specific heat of steel; T_1 and T_2 are the steel bulk temperatures at the finishing and coiling temperatures at the runout table, respectively; ρ is the strip density; u is the strip moving speed; H is the strip thickness; and B is the strip width. The heat removal rate can also be defined as

$$Q = m\eta = WA\eta \tag{9}$$

where m is the coolant mass flow rate (kg/s); A is the cooling surface area; W is the coolant spray rate or water flux (kg/m^2 s); and η is the cooling efficiency (kJ/kg), defined as the heat removed per unit mass of coolant. The heat removal rate may also be obtained by integrating local heat fluxes over the cooling surface,

$$Q = \iint q dA = \bar{q}A \tag{10}$$

where q and $\bar{q}$ are local and average heat fluxes, respectively. Combining Equations 9 and 10, one obtains

$$\eta = \bar{q}/W \tag{11}$$

The coiling temperature may be calculated using Equations 8 to 10. Since the cooling rate significantly effects the microstructure of the rolled steel (Figure 2), it is often used to correlate mechanical properties of the rolled product. Thus, it is necessary to estimate the cooling rate by combining Equations 8 to 10:

$$\frac{\Delta T}{\Delta t} = \frac{2\bar{q}}{\rho c_p H} \tag{12}$$

or

$$\frac{\Delta T}{\Delta l} = \frac{2\bar{q}}{\rho u c_p H} \tag{13}$$

where Δl is the length of the cooling area at the runout table equal to $A/2B$; Δt is the time required for the strip to travel through Δl; and $\Delta T = T_1 - T_2$.

Results

Figure 8 shows the measured surface temperatures at the center of the circular jet ($z/d = 0$) versus the Eulerian position x. It is shown that the surface temperature decreases sharply when it enters the cooling area. The temperature reaches a minimum before it recovers when the heating (conduction) from within the strip overcomes the convective cooling. The location of minimum temperature is slightly behind the stagnation point (jet center). After the strip leaves the cooling area, the temperature continues to recover until it enters the next cooling stage. It is also seen that a higher strip speed results in a higher surface temperature at the stagnation region under jet impingement. This is attributed to the fact that less residence

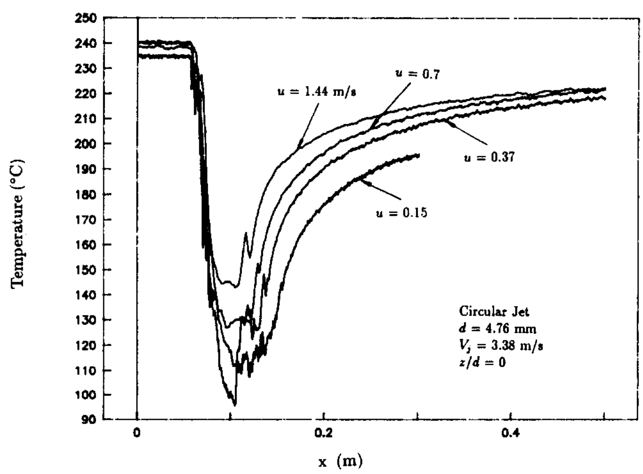

Figure 8 Measured plate surface temperature (T_s) versus Eulerian coordinate (x)

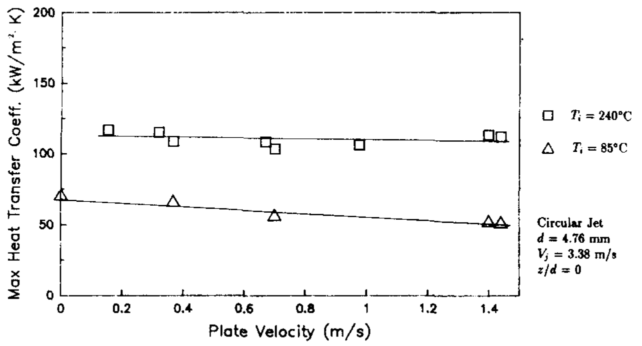

Figure 9 Maximum heat transfer coefficient (h) versus strip speed (u)

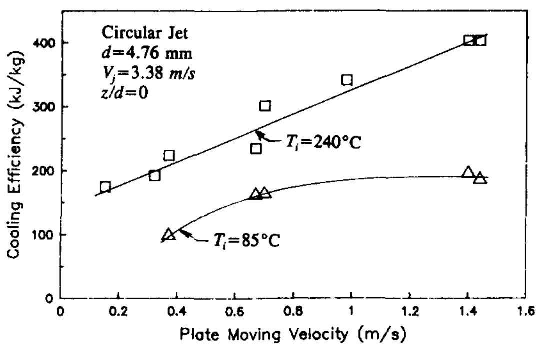

Figure 10 Cooling efficiency (η) versus strip speed (u)

time is experienced in the cooling zone for the higher moving speed.

Figure 9 shows the peak heat transfer coefficient, which occurs at the stagnation point, as a function of strip speed. In the case of $T_i = 240\,°C$, the peak h values are approximately constant at 110 kW/m^2-K; these occurred at local surface temperatures in the range of 110–180 °C, which are in the nucleate boiling regime. In the nucleate boiling regime, the heat transfer is dominated by the nucleation of vapor bubbles. Thus, boundary conditions such as moving surfaces and varying surface temperatures have negligible effects on heat transfer.

In the case of $T_i = 85\,°C$ where no boiling occurs, as the strip speed is increased, the peak h is decreased from 70 kW/m^2-K ($u = 0$) to 50 kW/m^2-K ($u = 1.44$ m/s). Although this may be partially due to the uncertainty of the experimental results, it is consistent with the numerical analysis carried out by Zumbrunnen (1990), who indicated that the heat transfer in single-phase forced convection is decreased as a result of decreasing surface-temperature boundary conditions.

Distributions of q and h have been reported by Chen, Kothari, and Tseng (1991). The cooling area covers 12 to 20 nozzle diameters in the moving (x) direction and 8 to 15 nozzle diameters in the transverse (z) direction. Despite different thermocouples being used by an earlier study with 0.0127-mm (0.0005-in.) thermocouples (Chen et al. 1991) and by the present study with coaxial film thermocouples fabricated by Medtherm, a consistent maximum heat flux of 17×10^6 W/m^2 has been reported.

Using Equations 9 and 10, the cooling efficiency can be calculated; it is shown in Figure 10. The cooling efficiency increases as the strip moving speed increases for both initial temperatures. This increase is attributed to the higher strip surface temperature (Figure 8), and hence the larger temperature difference between the strip surface and the coolant, rather than the higher heat transfer coefficients (Figure 9).

Assessment of cooling performance

While it is difficult to compare the cooling performances of various studies, due to a number of heat transfer parameters mentioned before, some comparisons such as cooling efficiency and heat fluxes are possible. The present results for a circular jet on a moving strip are shown in Figure 11. In the figure, the cooling efficiency versus contact time (between the steel surface and cooling area) is compared with that of Roberts (1988) for water spray on rolls. It is shown that a decrease in contact time results in an increase in the cooling efficiency. As shown in Figure 9, this effect is due more to the larger temperature difference between metal surface and the coolant (less surface temperature drop for shorter contact time) rather than higher heat transfer coefficients.

In a hot rolling mill, the surface temperature in the cooling area is very difficult to measure, because the surface is covered by water and vapor. Therefore, the strip surface temperature can only be measured at several discrete locations, such as the finishing mill exit and the coil temperatures. Therefore, it is often convenient to use the initial surface temperature as the heat transfer parameter in investigations.

Figure 12 shows the effect of initial strip surface temperature on the cooling efficiency. It is shown that as the surface

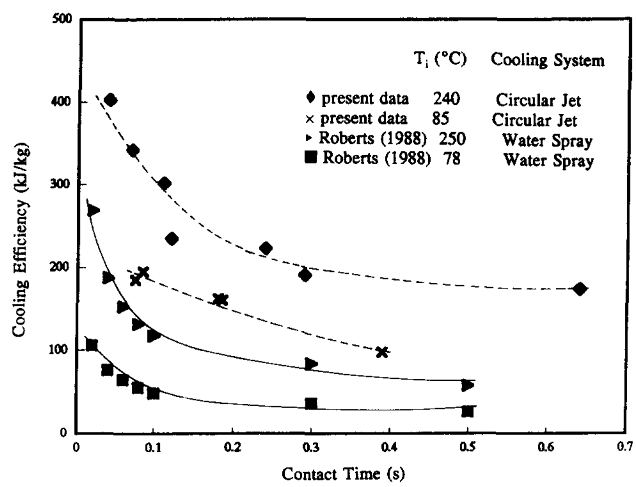

Figure 11 Cooling efficiency (η) versus contact time

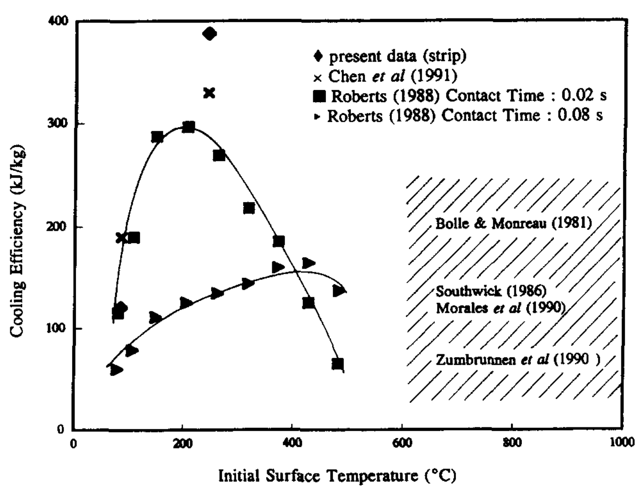

Figure 12 Cooling efficiency (η) versus initial surface temperature (T_i)

Table 2 Cooling performance from selected studies

Authors or editors	Surface (cooling)	q (W/m^2)	h (W/m^2 K)	W (kg/m^2 s)	η (kJ/kg)	T_i (°C)
Bolle & Monreau (1981)	Plate (spray)	$2-5 \times 10^5$	400–2,000	1–10	100–250	600–1000
Southwick (1986)	Strip/plate (spray/jet)	$10^5-3 \times 10^6$	500–5,000	2–20	10–150	700–900
Roberts (1988)	Roll (spray)	$3 \times 10^6-2 \times 10^7$	5,000–100,000	30–140	50–300	75–500
Morales et al. (1990)	Plate (spray)	$10^5-5 \times 10^6$	500–10,000	2–50	10–150	600–1200
Zumbrunnen et al. (1990)	Plate (slot jet)	$5 \times 10^6-10^7$	40,000–70,000	130	60	644
Chen et al. (1991)	Plate (round jet)	10^6-10^7	10,000–200,000	3.71	388	240

temperature increases, the cooling efficiency increases initially, reaches a maximum, and then decreases (Roberts 1988). The variation in efficiency is associated with an increasing heat transfer in nucleate boiling (where surface temperature is in the range of 150–200°C) and a decreasing heat transfer in the transition boiling regime, respectively. The present data show a cooling efficiency of up to 400 kJ/kg in the nucleate boiling regime. As the contact time increases, the surface temperature drops more, resulting in a shift of maximum efficiency to a higher initial surface temperature. If the surface temperature in the cooling area is used, the maximum cooling efficiency should occur in the nucleate boiling regime ($T_s = 150-200$°C). A cooling efficiency of up to 1,500 kJ/kg was reported by Labeish (1989) when water spray on a hot metal at 150–200°C was used.

Table 2 summarizes cooling parameters including heat fluxes, heat transfer coefficients, spray rates, cooling efficiency, and surface temperatures. The discrepancy in heat transfer is attributed to

(1) whether local or average values in q and h are used;
(2) whether single phase, nucleate, or film boiling are encountered (see Figure 13 for the comparison); and
(3) transient behavior and sensor response.

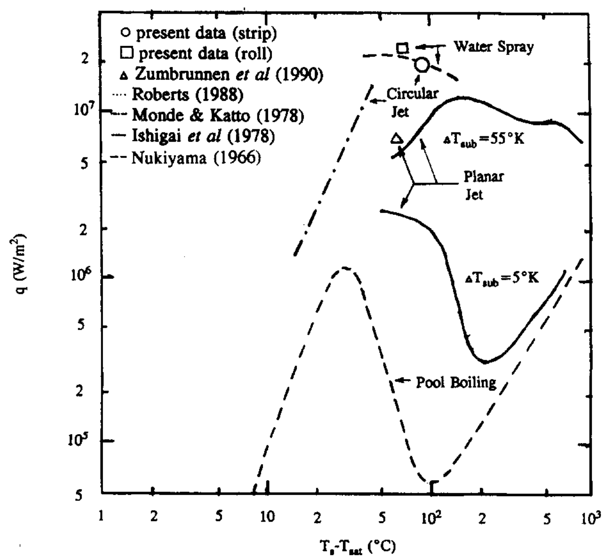

Figure 13 Boiling heat transfer on stationary and moving surfaces

The cooling efficiency in a wide range of 10–400 kJ/kg has been reported, because a number of heat transfer parameters are involved. Generally speaking, spray cooling is used in roll cooling and descaling, but jet cooling is more effective in strip cooling for material-properties control, in which a relatively high cooling rate is needed. Local heat transfer coefficients up to 200 kW/m^2-K are reported by Chen, Kothari, and Tseng (1991) using jet cooling on a moving plate and the present study of spray cooling on a roll. For boiling heat transfer, it is recommended that q rather than h be used in describing cooling characteristics.

Figure 13 shows heat fluxes versus surface temperatures in the boiling regimes. The pool boiling curve (Nukiyama 1966) is shown as a reference. Critical heat fluxes at a steady state from a circular jet on a uniformly heated disk, up to 15×10^6 W/m (Bennon 1985), were reported and correlated by Katto and Yokoya (1988); a physical model describing the heat transfer mechanism was later given by Kandula (1990). Ishigai, Nakanishi, and Ochi (1978) showed that an increase in subcooling from 5 to 50 K not only substantially enhance the heat transfer, but also shifted the heat flux curve to a higher temperature at $T_w - T_{sat} = 100$°C. In cases of cooling on moving surfaces, heat fluxes are generally higher than those for stationary surfaces. In water spray cooling, heat fluxes as high as 20×10^6 W/m^2 have been reported by Roberts (1988) and the present study. In circular jet cooling of moving strips, heat fluxes of up to 17×10^6 W/m^2 are reported by Chen, Kothari, and Tseng (1991) and the present study. A peak transient heat flux as high as 10^8 W/m^2 in nucleate boiling was reported by Labeish (1989).

Implementation in cooling system design and control

According to the studies mentioned above, some guidelines regarding the cooling system design may be established. As shown in Figures 5 to 8, the surface temperature recovers along the moving direction beyond the impingement region. This information may be used to consider where the next spray or jet may be located. For example, if maximizing heat transfer is needed, the cooling jets may be arranged so that the surface temperature is kept in the nucleate boiling regime. However, this is quite difficult at the runout table in hot rolling because the strip surface temperature, except in the stagnation region, is always in the film boiling regime. Therefore, the jet arrangement should be primarily based on distributions of convective heat transfer.

Local heat transfer distributions in q and h are also important in designing cooling systems. In roll cooling, they can be used to calculate thermal expansion and induced thermal stresses, which provide valuable information for the prediction and control of thermal crown and flatness (Tseng et al. 1989; Bennon 1985). In strip cooling, local q and h may be used to calculate the temperature profile in space and time (Chen et al. 1990) for better control. Equation 13 indicates that for a given finishing and coiling temperature, the runout table length (Δl) may be reduced by increasing the average heat flux (heat intensity). The runout table length is also linearly proportional to ρ, u, C_p, and H of the strip, provided that the time is sufficient for the temperature in the depthwise direction to reach a uniform value.

When the circular jet system is considered, staggered arrays of jets should be used to minimize the temperature variation in the lateral (z) direction. From a previous study (Chen et al. 1991) and the present work, it has been shown that the cooling area covers 12 to 20 diameters in the moving direction and 8 to 15 diameters in the lateral direction. Higher moving speeds shorten the cooling length in both moving and lateral directions. It is suggested that the distance for staggered arrangement for circular jets should be about 3 to 5 nozzle diameters between tubes (S_n) and about 10 to 20 nozzle diameters between rows. Nozzle diameters are in the range of 10–20 mm.

In the case of planar jets, the cooling zone covers about 20 slot widths according to an experimental study performed by Zumbrunnen, Incropera, and Viskanta (1990). Therefore, the distance between planar (slot) jets is recommended to be about 20 slot widths (S), as shown in Figure 14b. The typical slot width is in the range of 10–20 mm.

When the coiling temperature is to be controlled, Equations 7 to 10, coupled with average q and h, or cooling efficiency may be used to predict the coolant flow rate required. Feedback control can then be used to compensate for the errors. Both on–off control and variable flow control have been adopted in steel mills. The cooling rate that is often used to predict and control the microstructures of the strip can be calculated using Equation 12, provided that the runout table length and the moving speed are known. For control of temperature profiles in time and in the strip, the partial differential equations with local distributions of convective boundary conditions must be used (Chen et al. 1990). Further, adaptive control may be used to estimate the actual cooling parameters such as q, h, and η_c. Recent developments in adaptive control of controlled cooling have been given by Yahiro et al. (1991) of Kawasaki Steel, and developments in robust control for accelerated cooling have been reported by Yousuff et al. (1991) of Drexel University.

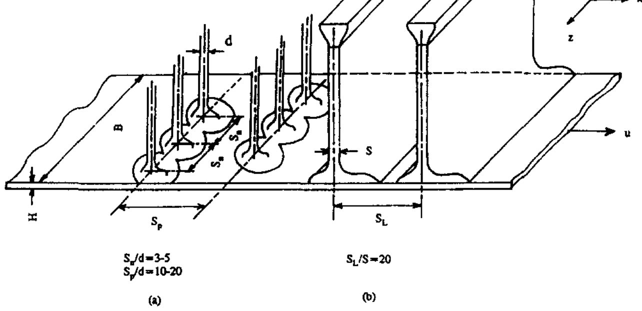

Figure 14 Suggested cooling system arrangement for (a) staggered arrays of circular jets and (b) array of planar jets

Conclusions

The importance of roll and strip cooling on thermomechanical processing in steel rolling has been reported. Prediction and control of roll and strip cooling are necessary in modern steel mills, because they not only affect the process efficiency but also strongly influence the quality of rolled products. Metallurgical, thermal, and control aspects of controlled rolling and controlled cooling are given.

Current steel mills are built with incomplete databases in heat transfer associated with spray and jet cooling. Based on the present study and other recent research efforts, the heat transfer coefficient is found to be affected by many parameters, including surface temperature, jet velocity and temperature, jet geometry and arrangement, and strip speed. The effect of surface temperature on boiling was well documented. Increasing coolant subcooling (decreasing jet temperature) was found to push the boiling curve to higher heat fluxes at higher surface temperatures. Increasing jet velocity increases heat transfer so long as it stays in the laminar regime. The effect of surface motion on heat transfer has not been studied until recently.

Two types of cooling—spray cooling on a roll and jet-impingement cooling on a moving strip—are studied. Cooling characteristics including heat fluxes, heat transfer coefficients, spray rates, cooling efficiency, and surface temperatures are presented and compared. Results indicate that the effect of surface motion substantially influences local heat transfer behavior. When present studies are compared to data in the literature, it is found that heat fluxes (heat transfer coefficients) range from average values of 2×10^5 W/m^2 (400 W/m^2-K) in film boiling to peak local values of 20×10^6 W/m^2 (200 kW/m^2-K) in nucleate boiling. These correspond to cooling efficiencies of 50–400 kJ/kg at $T_i = 100$–$500\,^\circ$C and 10–250 kJ/kg at $T_i = 600$–$1000\,^\circ$C. These large variations are primarily due to the boiling regime and whether the average or local values are reported.

Based on the thermal models and heat transfer studies, a cooling system arrangement is suggested (Figure 14) and a control strategy is given. For controlled cooling at the runout table, the planar jet system is preferred. If circular jets are to be considered, a staggered arrangement should be used. Water sprays are preferred in providing descaling and sweeping.

In the steel industry, the basis for using jet and spray cooling has been primarily the trial-and-error approach or plant experience. The available heat transfer data in the literature, on the other hand, is obtained in a well-controlled laboratory environment, where only one parameter is considered at a time. In the future, research is needed to consider interactions among multiple jets as well as the combined effect of all major heat transfer parameters. With the present incomplete database, it is suggested that heat transfer correlations obtained in the laboratory be adjusted in the actual industrial situation, i.e., using model-based parameter estimation.

Furthermore, the successful operation of steel mills requires the integration of heat transfer with metallurgy and control technologies. Reliable sensors in the laboratory and steel mills must be developed to detect surface as well as internal strip temperatures. Future development should also include on-line parameter estimation, adaptive control, and statistical process control (SPC) using intelligent sensors to measure strip thickness, profile, and internal temperatures.

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