Fully Numerical Analysis for Effects of Cooling Water Flow Rate and Plate Running Speed on Steel Plate Cooling in Very High Temperature Region

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The steel plate cooling process in hot rolling work has a variety of parameters influencing the cooling aspect of the process and the mechanical properties of products; i.e. the cooling water flow rate, the plate running speed, the cooling water temperature, the injection method of cooling water and so on. The present study deals with the cooling water flow rate and the plate running speed as vital parameters in the cooling process. The process is simulated by a fully numerical approach for a cooling type of impinging water jet on the moving plate. The previous achievement by Park1) of numerical analysis for the cooling process including the free surface capturing, the numerical modeling of boiling heat transfer, and the treatment of the moving plate is successfully adopted in the present study. The residual water height, the impinging pressure, the residual water shape, and the cooling history including the average heat flux are suggested for various cooling water flow rates and plate running speeds. The thickness distributions of the vapor film layer on the running plate are graphically presented too.

KEY WORDS: ROT(Run Out Table); boiling heat transfer; film boiling; effective thermal conductivity; cooling water flow rate; plate running speed.

1. Introduction

Plate cooling process using the circular impinging water jet will be dealt with in this study. Although the cooling process seems very simple and well understood, the process actually includes a lot of complex physical components. First, the process contains the multi phase flow problem having the freely changing interface between the liquid water and air. Moreover regarding the thermal aspect, the phase-changing heat transfer (i.e. boiling) occurs between the cooling water and the hot steel plate. In the aspect of hydrodynamics, the plate moves at a high speed, thus the movement of the water remaining (residual water) on the plate are closely related with the plate motion. These barriers have made the experimental study flourish rather than the numerical one. A lot of researchers have thought that the experiment is the only alternative for investigating all the related phenomena simultaneously.

Experiments for water cooling of a moving hot plate make it possible to obtain very useful data such as the cooling stop temperature, the average heat flux, and so on.2–5) As discussed in previous work,1) since the process is closely connected with the thermal-fluidic behavior of the residual water accompanying with the impinging jet motion, the plate running motion, the heat transfer between the hot steel plate and the cooling water, and the complex effect of these factors, the perfect similarity between the laboratory scale and the real scale cannot be achieved. Hence a lot of researchers have conducted real scale experiments6–9) not to be retained on a laboratory scale. These experiments have produced lots of useful data with no-discrepancy in a similarity. In real scale experiments, however the testing expense is too high as well as a lot of uncertainties such as the climate conditions (ambient temperature, humidity, etc.) and the working conditions of upstream processes (refining, heating furnace, roughing mill, finishing mill, etc.) are solved together in the final resultant products. Thus the exclusive effect of each parameter (cooling water temperature, flow rate, plate running speed, and so on) is hard to be analyzed quantitatively and qualitatively using experimental methods.

In this study, thanks to the recent innovative growth of computing equipments and the tremendous development of the related numerical modeling1), the effect of each cooling parameter will be investigated fully numerically to exclude the ambiguous results mixed with other uncertain factors under fixing the conditions of other parameters.

In this study, the two most important factors, i.e. the supplying cooling water flow rate and the plate running speed, among the various influencing parameters will be handled exclusively. All the cases are numerically simulated on the basis of the computational model by Park1) for the steel plate cooling in hot rolling process. The qualitative influences of each parameter will be discussed in detail as well the complex interconnecting results will be handled. The results for the hydrodynamic characteristics such as the height of the
residual water and the impinging pressure as well as the heat transfer feature such as the cooling history, etc. will be presented and discussed reciprocally as the above two factors change.

2. Cooling Mechanism after Finishing Mill

Many cooling mechanisms such as spray cooling, slit jet cooling, and so on, have been used for real hot rolling plants. The circular impinging water jet type (shown in Fig. 1), one of them, is the type to be largely adopted in most of the real plants at the present time. That has a special advantage to the plate cooling since it can maximize its cooling performance to extract the heat from the hot moving plate due to the impinging effect of the water jet against the moving plate.

Recently, the supplying amount of the cooling water tends to increase continuously in connection with the request for the development of the new steel products having better mechanical properties or a specific quality, for example, the fining grain size, the phase transforming, and etc. All these pursue the diversifying and upgrading of steel products through the process control only without changing any hardware specifications such as the length of the Run Out Table. Most new facilities constructed after late 1990, have a cooling water supplying capacity over 20 kg/m²·sec.

Accordingly the amount of the residual water (the water remaining on a moving plate not yet drained out through the side exhausting path) is greatly increased. This can improve the plate cooling capacity somewhat. But it is expected that the increased supplying flow rate thickens the residual water layer and retards the supply of the cold cooling water to the hot steel plate close at hand. The circular jet type can concentrate its momentum on a small spot, and thus can transport the coldness of the cooling water to the cooling object (moving hot steel plate) more closely. Actually the attempts to keep the laminar state of the water jet longer and stable, were preexisted. They usually manipulated the nozzle shape such as the length of the straight region and the angle of the cone area in the nozzle to retard the transition to turbulence. Furthermore, the convective heat transfer between the hot steel plate and the residual water layer has a considerable contribution to the whole heat transfer. The convective heat transfer is composed of contributions from not only the shear flow from the running plate at high speed but also the mixing and circulation by the impingement of the jet. Also the circular impinging water jet has an effect reducing the representative temperature of the cooling water through mixing the residual water by initiating the large scale current in the residual water layer.

As mentioned before, it is not true that the increase of the flow rate always contributes to the enhancement of the cooling performance linearly. It has been empirically known that the cooling performance seems to be saturated with increasing the supplying amount of the cooling water. In this paper, these saturation phenomena of the cooling capacity are simulated along the various cooling water flow rate and the plate running speeds. The change of the cooling water flow rate, the corresponding change of the cooling performance and the intermediate parameters such as the residual water thickness or the impinging pressure are reciprocally related.

3. Numerical Modeling

First, to simulate the free surface motion of the residual water layer, VOF (Volume Of Fluid) method is applied in this study. VOF is a fully Eulerian scheme to capture the free surface position on a fixed mesh system and has been successfully applied to various problems including gas-liquid interface. The transport equation for VOF function (gas volume fraction in a computing cell) is discretized by HRIC (High Resolution Interface Calculation) scheme. Although the scheme having a much higher accuracy such as PLIC (Piecewise Linear Interface Calculation) scheme is applicable, the present problem does not need such a high resolution procedure which accompanies the high computing cost, because the main physics for the plate cooling occurs near the plate surface, i.e. the interface between the running plate and the residual water.

Second, since the plate running motion is a vital factor of the cooling process as mentioned before, the inflow of a hot plate is numerically modeled equally to the real situation. The plate flows in with high temperature and moves out by being cooled by the cooling water. Although in fact, the temperature has some distribution in plate depth at the inlet stage, for the simplicity it is assumed that the plate comes in with a uniform temperature profile.

Next, the heat transfer between the plate and the cooling water is modeled by the film boiling phenomenon. This was established by Park for the first time and available in a very high temperature zone after the finishing mill. The main idea is that a very thin steam layer is in existence between the plate and the residual water. The steam layer plays a role as a kind of thermal resistance. The steam layer thickness depends on the thermal situation on the plate and calculated through the iterative solving procedure for the overall thermal balance of the plate and the cooling water. The thin steam layer was not applied directly in real simulations because the layer is too thin to guarantee the accuracy of the numerical analysis if the mesh system is allocated for those thin regions. Thus the concept for the effective thermal conductivity in a first grid cell for the residual water on the plate was developed. Actually the steam vaporized on the plate is not restricted in the steam layer, and escapes from the layer in the form of bubbles. But in this study, those emissions, the vaporized mass, and the loss of the residual water by vaporization are not considered in the calculation because the vaporized mass is actually too small when compared with the mass of the supplied cooling water so as to be neglected, despite of the big volumetric amount. The details for the effective thermal conductivity and the
iterative procedure for the steam layer thickness will not be handled in this paper. More detailed information is available in the reference.\textsuperscript{1)}

The only one period in a span-wise direction as shown in Fig. 2, the 6 times repeated length between nozzles in a plate moving direction, and the entering and exiting regions where the nozzle is not located, are selected for whole calculating domain. The distance from nozzle to plate is 150 mm and the nozzle diameter is 5 mm. The 18 000 cells are assigned for the surface that parallel with the plate upper surface, the 12 nodes for the plate thickness, and the 60 nodes vertically for the air-water region upon the plate. Total 130 000 cells involve in a calculation. The length of the calculating domain including the entering and exiting region is 1 meter. The plate thickness is 12.5 mm. The lower surface of the plate is regarded as the symmetry plane under the assumption that the same cooling process occurs on the opposite side too. But in the real hot rolling process, the upper and lower parts of a plate have a little different cooling condition. In this study, for simplicity the same condition is applied for both upper and lower surfaces. So actually the plate of 25 mm thickness is dealt with in this study.

To solve the basic flow and heat transfer, the commercial CFD program, Fluent 6.3.26 was basically used. The UDF (user defined function) program was coded to realize the above concepts. To calculate the domain with one meter long, the Zeon E5620/2.4GHz-12M processor was used, and 8 cores involved in a calculation. The computing time of about 96 hours was taken to simulate for each case for 4 seconds calculation in a real time scale.

4. Results and Discussions

First of all, the level of residual water layer was predicted for the various cooling water flow rates and the plate running speeds. The plate speed was tested in the range of from 40 to 70 mpm. The mass flux of supplied cooling water ranged from 10 to 45 kg/m²s. The level of residual water layer is nearly linear to the supplied cooling water flow rate as shown in Fig. 3. It is revealed that the plate running speed is not a vital factor governing the level of residual water layer. Thus the level can be represented as the following linear regression equation with the supplied cooling water mass flux.

\[
\text{Level} [\text{cm}] = 0.0396 \cdot \text{mass flux} [\text{kg/m}^2\text{s}] + 2.1256 \quad \ldots \quad (1)
\]

where, the mass flux, \( \text{mass flux} \) was calculated with based on the plate surface area covered by the residual water not a nozzle exit area.

Figure 4 shows the distribution of the impinging pressure on the plate surface in the running direction. The graph is captured from the meridian line arraying the supplying nozzle. The peaks are detected at 6 points directly below the supplying nozzles. From a result, the plate running speed is also not a vital factor influencing the impinging pressure. Figure 4 shows only the case of mass flux 16.3 kg/m²s because the other cases have a quite similar pattern of the 6 peaks and low pressure region. But the pressure at peaks is greatly changed with the supplied cooling water flow rate as shown in Fig. 5. The figure shows the magnified view around the first peak on the plate. It is detected that the locations of peak points move slightly in each case not to be fixed directly under the nozzle. This is a kind of instantaneous matter caused by the cross flow within the residual water layer. At different time, the peaks can also change to other locations. The high impinging pressure has two meanings; first, that the impinging jet can break or suppress the Leidenfrost steam layer, and second, the jet transmits its coldness to the hot plate more impactful. Figure 6 repre-
sents the relation between the maximum impinging pressure and the supplying mass flux. Although actually the impinging pressure is also influenced by the nozzle exit diameter, in this study the nozzle exit diameter was fixed as 5 mm in order to reduce the number of parameters and investigate the effects of the bulk amount of supplied cooling water intensively. Anyway, the maximum impinging pressure can be expressed as following second order polynomial of the mass flux.

\[ P_{\text{max}} = 0.0268 M^{0.25} + 0.1291 M^{0.122} \]

where \( P_{\text{max}} \) [kPa], \( M \) [kg/m²s].

Figure 7 shows the shapes of residual water layer for various supplied cooling water flow rates. As the supplied amount of cooling water increases, the undulation of free surface becomes remarkable. The level of residual water right below the nozzle area becomes 2–5 times of one for no-jet-cooling area according to the supplied mass flux. At low cooling water mass flux, 10.9 kg/m²s, the levels of residual water at both the nozzle area and the no-jet area are nearly same. As well as, in this paper, the Leidenfrost steam layer has been visualized as shown in Fig. 8. The thickness of Leidenfrost steam layer is found during the iterative procedure for the effective thermal conductivity of 1st row cells on plate surface. The iterative procedure is originally based on the continuity for the temperature and the heat flux at the interface between the steam layer and the residual water (for
more detailed information, see Ref. 1). Figure 8 shows the thickness distributions of Leidenfrost steam layers for various cooling water mass flow rates. The thickness has a range about from 2 to 80 μm. The present estimated values for the steam layer thickness are regarded quite reasonable on the basis of the Leidenfrost’s finding that the thickness of steam layer is about from 50 to 200 μm under the non-impinging condition. As shown in Fig. 6, since the impinging pressure is proportional to the square of mass flux, as increasing the mass flux, the thickness of the steam layer becomes smaller and smaller. Moreover, the area of the extremely thin steam layer is enlarged as the cooling water flow rate increases as shown in Fig. 8. For example, for the case of mass flux 43.6 kg/m²s, the most area right under the nozzle has nearly zero thickness of steam layer. Of course the thickness of the steam layer is determined by not only the impinging condition (impinging pressure) but also the resultant temperature on the plate surface by the cooling process. That is the zero thickness of steam layer means that the temperature on the plate surface goes below the saturation temperature. Thus, although the heat transfer is expected to be boosted tremendously in the single phase heat transfer mode (zero thickness of steam layer), actually since the temperature difference between the plate surface and the cooling water becomes small, the heat transfer is reduced by the no-phase-change heat transfer mode.

In Fig. 9, when the film boiling phenomenon is numerically modeled or not, the cooling histories are compared with each other. In no-boiling case, the Leidenfrost steam layer is neglected, thus the thermal resistance between the plate and the cooling water is forcibly removed. The case of no-boiling has a larger heat transfer performance and shows little difference at the heat retrieval region. The no-boiling case merely shows the heat retrieval. This is caused by neglecting the effect of the Leidenfrost steam layer which has absolutely different values for the nozzle area and the no-jet area. In fact there are two main factors of the heat transfer (convection of impinging jet and heat resistance of Leidenfrost steam layer). It is a result from omitting one of them.

Figure 10 shows the changing history of the temperature at 1 mm under the plate surface for various cooling water flow rates. As expected, the trivial results are obtained. The bigger mass flux is, the larger temperature drop is. In Fig. 11, the same quantity is plotted for various plate running speeds. The case of more slowly moving shows a greater temperature drop because of the increase of absolute time being exposed in jet cooling region.

From the above results for various plate speeds and cooling water mass flow rates, the averaged heat fluxes are plotted like Fig. 12 (Figure 13 is a kind of map of heat flux representing the same physics as Fig. 12.). It is revealed that the heat flux tends to saturate as the mass flux increases, which it is mentioned shortly in the beginning part of this paper. It is considered to be closely related to the saturation of the thickness of Leidenfrost steam layer as described before. Also of significance is that the averaged heat flux increases as the plate running speed increases. Although the total heat transfer amount (i.e. the temperature drop) decreases as the running speed increases, the cooling intensity per unit time is bigger under high running speed conditions because the thermal communications between the plate and the cooling water occurs in a much higher temperature mood under the high running speed conditions. The similar findings to this have been reported severally.
5. Conclusions

The cooling process for hot running steel plate has been simulated for various cooling water flow rates and plate running speeds. From the simulations, the interconnecting parameters between hydrodynamic and thermal features, such as the residual water level and the impinging pressure were found to be a function of cooling water mass flux. Also the heat flux was estimated with variations of cooling water flow rate and plate running speed. The phenomenon of heat flux saturation as an increase in cooling water flow rate was fully numerically reproduced in this study. To explain the related thermal-fluidic physics, the Leidenfrost steam layers have been visualized and compared for various cases for cooling water flow rate and plate running speed. The increase of heat flux along the plate running speed has also been calculated in the present numerical simulation. The variation of heat flux for various cooling water flow rates and plate running speeds has been graphically mapped.

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