Numerical Analysis for Film Boiling Heat Transfer of a Moving Hot Steel Plate

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(Received on December 20, 2010; accepted on January 27, 2011)

The cooling of the steel strip in ROT (run out table) of the hot rolling process is very important in that the ROT is the final process to control the mechanical properties of the final products. In general the products are cooled by the circular water jets which makes the comparatively thick residual water layer about 100–200 mm with the supplied flow rate on the plate. The basic phenomenon governing this cooling process is the boiling heat transfer. The boiling phenomenon is one of the most well-understood heat transfer mechanism by the past a few decades research. However, almost all the results have been reached through experimentation. This paper focuses on the boiling heat transfer on the moving hot plate with a fully numerical approach. The simulation was conducted only in a very high temperature region (over the Leidenfrost temperature) where the film boiling can be kept steadily on the plate. The film (the steam layer) was regarded as the heat resistance whose capacity (actually the steam layer thickness) varied with the plate surface temperature and the flow rate and temperature of the water jet. To fix the steam layer thickness showing a very different spatially distribution on the plate, the continuity for the temperature and heat flux at the interface between the cooling water and the steam layer is satisfied by the iterative procedure for the effective thermal conductivity of the 1st cell on the plate wall.

KEY WORDS: ROT(run out table); boiling heat transfer; film boiling; effective thermal conductivity.

1. Introduction

Strip cooling or plate cooling is one of the most important processes in steel production because it is the last process controlling the mechanical properties of products. The various cooling methods (circular impinging water jet, slit impinging water jet, mist cooling, air cooling and so on) are available with specific purposes. In this study, the type using the circular impinging water jet will be handled. Although the cooling process seems very simple and well understood, actually the process 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 in the thermal aspect, the phase-changing heat transfer (i.e. boiling) occurs between the cooling water and the hot steel plate (or strip). In the aspect of hydrodynamics, the plate (or strip) moves with high speed, thus the movement of the water remaining (residual water) on the plate are closely related with the plate motion. This means the present problem includes one more moving interface except the water-air interface. Moreover this additional moving interface is not a system boundary but an important threshold thermally coupling the plate and the cooling water. To solve the problem of the moving plate and the cooling water simultaneously is necessary because there is no information for the temperature or heat flux for the plate, which is rather a product that should be found by calculation.

As mentioned, the cooling process contains a lot of complex physics which are difficult to solve even solely. So studies of the problem have largely been restricted to experimental methods. Experiments for water cooling of a moving hot plate make it possible to obtain very useful data for example, the cooling stop temperature (or coiling temperature), average heat flux, and so on.^{1–4)} But the experiments on a small scale can not explain the real situation occurring in an actual steel plant. The strict similarity between the lab. and plant scales couldn't be achieved because not all the various phenomena are included in the process as described before but also the time scale in a lab. experiment is different from that of the real situation. The thermal energy dissipation of a sample in a lab. scale occurs too rapidly when compared with the situation in a real steel plant. Hence a lot of researchers have conducted real scale experiments.^{5–8)} These experiments produced lots of useful data with no-discrepancy in a similarity. However the lagrangian history of the temperature at a certain part of the plate couldn't be stored with time from cooling start to end because the plate moves fast and its temperature is too high. Because of these conditions, the measuring has been performed by using the pyrometer at fixed location. This is ever so much Eulerian.

In this study, the cooling process including the effects of the boiling, the two-phase free surface, and the moving plate in whole is numerically simulated. A numerical model to solve those components simultaneously will be suggested. Although the present simulation is restricted to only a very

high temperature region after FM (Finishing Mill) where the film boiling is dominant, it is possible to provide a lot of useful data, the cooling history of the plate not only at the plate surface but also at the very inner region of plate, etc.

2. Cooling Mechanism after Finishing Mill

The strip or plate cooling system after FM is schematically depicted in Fig. 1. A circular water impinging jet is usually adopted in this process because of its high cooling capacity. Recently, in the aspect of the mechanical properties of products, the requirement for the finer grain size and the control- or rapid-cooling philosophy is growing year by year. Accordingly the supplied cooling water amount is also significantly increased, thus the supplied cooling water makes a thick residual water layer on the running plate as shown in Fig. 1.

In such high flow rate conditions, the thermal interaction between the running plate and the cooling water can be classified into 4 types as shown in Fig. 2 according to the plate surface temperature, T_{wall} . First, when the surface temperature is below the vaporizing temperature, 100°C at the ambient pressure, of 1 bar, the thermal interaction between the plate and the cooling water occurs without phase change, i.e. no vaporizing. Secondly, if the surface temperature is higher than 100°C and below the Leidenfrost temperature, the nucleate boiling appears on the plate surface. If the surface temperature is over the Leidenfrost temperature as in Fig. 2(d), the plate surface is covered with a thin steam layer and the plate-water thermal interaction alters the film boiling mode. But if the momentum of the water jet is great enough to transport its cooling capacity to the plate surface against the damping of residual water layer, the spot where the single phase heat transfer occurs can exist nearly below the

Fig. 1. Schematic of traditional impinging water jet cooling system in hot rolling mill.

Fig. 2. Boiling mechanisms on hot plate surface according to surface temperature.

circular jet as shown in Fig. 2(c). The interaction mode of Fig. 2(c) can also appear when the residual water layer is thin enough. Therefore the interaction mode between the plate and the cooling water can be changed with not only the surface temperature but also the local position. Thus the plate thermally communicates with the cooling water in the boiling heat transfer and the single phase heat transfer alternatively as the plate runs.

3. Numerical Modeling

In this study, the plate cooling process in a short region after FM where the plate surface temperature is higher than the Leidenfrost temperature is dealt with numerically. Therefore the dominant communication mode between the plate and the cooling water is the film boiling (Fig. 2(c) or 2(d)). Focusing on the heat transfer problem between the plate and the cooling water, the metallurgical phase change in the plate (for example, austenite to ferrite) and the corresponding heat generation are neglected in this study.

The basic flow and temperature fields are calculated in the conditions of unsteady, turbulence, incompressible flow, using the commercial CFD code Fluent 6.3. The standard $k-\epsilon$ model is used for the turbulence modeling, all convective terms are discretized with the 2nd order linear upwind scheme, the time step is kept in 0.001 sec. during calculations and the 1st order implicit scheme is applied for unsteady formulation.

To find the water to air interface (free surface) which is formed by the circular water jet and the residual water layer, the VOF (Volume Of Fluid) technique is adopted in this study. The free surface is implicitly captured from the flow information. The modified HRIC (High Resolution Interface Calculation) scheme⁹⁾ provided in Fluent 6.3 is selected for discretization of convective terms of VOF eqn.

When film boiling is occurring on the plate surface, the steam layer (steam film) plays a roll as a severe thermal resistance between the cooling water and the plate. Generally the thickness of steam layer is too thin to be suitably allocated to a grid system. Also since the film thickness varies with a thermal condition, the fixed grid system can not be assigned and the grid layering or regeneration is required. But these grids add complexity to the problem and the solution is unstable due to the inconsistent grid spacing problem.

In this study, to simulate the cooling process including the moving boundary and the boiling heat transfer, the following iterative procedure was developed. The main idea is that the thickness of the steam layer on the plate is found during the iterative procedure for fixing the effective thermal conductivity in the 1st cell on the plate.

Fig. 3. Mathematical modeling of 1st-row cells on plate surface for effective thermal conductivity.

Figure 3 shows the grid system on the plate surface. Actually due to the film boiling on the plate, the very thin steam layer exists on the plate surface as shown in Fig. 3. To consider the thermal resistance of the steam layer without special treatments (for example, the domain decomposition or mesh regenerations), the concept of effective thermal conductivity which includes the effect of thermal resistance for the film layer is adopted to the cells in the 1st row on the plate.

In a first step, the flow and temperature fields are calculated with the initially guessed effective thermal conductivity, k_{eff} in the 1st-row cells on the plate like the flow chart in **Fig. 4**. Normally the value for water is used as an initial guess.

Second, using the wall heat flux, $\dot{q}_{wall}''$ and the wall temperature T_{wall} obtained by the previous step, the thickness of film layer, δ_{film} can be estimated with Eq. (1) describing the thermal diffusion in the steam layer.

$$\dot{q}_{wall}'' = k_{steam} \frac{T_{wall} - 100[^\circ\text{C}]}{\delta_{film}} \quad \dots\dots\dots (1)$$

Here, the constant 100[°C] on the right hand side means that the interfacial temperature between the steam layer and the liquid water is nearly the saturation temperature for atmospheric pressure. k_{steam} is the thermal conductivity of steam.

Thirdly, from continuity for the heat flux at the interface between the steam layer and liquid water, the cell-centered temperature, T_{cell} can be recalculated with the following Eq. (2).

$$\dot{q}_{int}'' (= \dot{q}_{wall}'') = k_{water} \frac{100[^\circ\text{C}] - T_{cell}}{\delta_{cell} - \delta_{film}} \quad \dots\dots\dots (2)$$

The heat flux at the interface, $\dot{q}_{int}''$ is equal to the wall heat flux, $\dot{q}_{wall}''$ given the assumption that the heat from a plate is wholly transported to the cell in the normal direction. k_{water} is the thermal conductivity of the liquid water, and δ_{cell} the distance from a plate wall to a cell-centered point as depicted in Fig. 3.

In the fourth step, using Eq. (3) the effective thermal conductivity which was used for flow field calculation in the first step can be corrected.

$$\dot{q}_{wall}'' = k_{eff, new} \frac{T_{wall} - T_{cell}}{\delta_{cell}} \quad \dots\dots\dots (3)$$

Here for $\dot{q}_{wall}''$ and T_{wall} , the values obtained from the first step are used, and T_{cell} is the recalculated value in the third step. Obtaining the new effective thermal conductivity, $k_{eff, new}$ from Eq. (3), the error analysis for the old and new values for k_{eff} is conducted. If the new value meets the convergence criterion, the solving procedure goes over the next time step. If it does not meet the criterion the procedure goes back to the first step with the corrected k_{eff} .

The terms $\dot{q}_{wall}''$ appeared in Eqs. (1–3) represent the heat flux from a plate to surroundings. They include not only conductive heat transfer but also convective heat transfers. Heat transfer, however, is a boundary or interfacial phenomenon between the system and its surroundings, thus the heat communication at a fixed position such as the interface between the plate and the cooling water can be represented with Fourier law such as Eqs. (1–3). In other words, the Eqs. (1–3) represent a resultant heat flux due to the conduction

Fig. 4. Flow chart for setting effective thermal conductivity in 1st-row cells.

and convection heat transfer. Actually the realistic thermal communication with surroundings occurs at a free surface of residual water layer. In this study the convective heat transfer between the cooling water and the surroundings is completely dealt with by considering the air domain together with the cooling water and hot plate domains as computational fields. The radiation from a plate surface is absorbed and scattered in the residual water layer. This scattering process can warm the residual water and resultantly retard the plate cooling. But since the residual water is continuously refreshed by a supplied cooling water and the temperature varying range is so narrow because the highest value is confined below 100[°C], the warming of cooling water by radiative scattering is neglected in this study. Notwithstanding, the cooling water warming is a very significant factor governing the plate cooling process, thus the radiation problem is needed to be dealt with more precisely in the future study.

In the much lower temperature region, far downstream from FM, this notion is invalid, because the film boiling is no longer dominant.

4. Results and Discussions

The numerical simulation is conducted for the steel plate 25 mm thick, the running speed 60 mpm, and the incoming temperature 850°C uniform. The thermal properties of the steel plate are assumed to be constant with temperature for convenience because the present study is focusing on the simulation model for the boiling heat transfer on the plate. The cooling water temperature is 30°C, the mass flux 20 kg/m²s. The 1.3 millions hexahedral cells are applied for the calculation.

The **Fig. 5** shows the motion of residual water as time goes after the cooling water impinges on the moving plate. A very thick residual water layer is formed. The color in the figure means the temperature. The plate is cooled along the downstream journey. The temperature of the residual water upstream is higher than downstream because of the high incoming temperature of the plate. The motion and shape of residual water becomes steady at about 2.5 sec.

Figure 6 shows the temperature distribution on the plate surface and the planes 1, 5 mm below the plate surface. The

Fig. 5. Residual water shapes and plate temperature as time goes on.

Fig. 6. Temperature distributions on (a) plate surface plane, (b) plane 1 mm below plate surface, and (c) plane 5 mm below plate surface.

plate is cooled down as it goes downstream. The spanwise distributions of temperature are clearly observed on the plate surface plane. And the hexagonal or diamond-shape cells are locally formed by the effect of jet impingement. The tails of cells are stretched out to the downstream due to the plate motion of 60 mpm. After the jet impingement zone, the surface temperature tends to recover by the heat retrieval of the plate. On more inner planes of Figs. 6(b) and 6(c), the spanwise distributions of temperature starts disappearing and the temperature recovery also vanishes. The plate temperature at 5 mm below the surface doesn't decrease in the forefront of the jet cooling zone even though the upper part is cooled down violently. This is more clearly shown in Fig. 7. This study plots the temperature profiles in the plate-depth direction along the downstream at fixed spanwise position. The surface temperature is rapidly decreasing from 0.3 m downstream where the jet cooling starts. But after 0.7 m downstream where the jet cooling finishes, the surface temperature is recovering due to the diffusion or discharge of the heat at more inner side of the plate. It is possible to check that the temperature at the deeper position than 1 mm is still decreasing in Fig. 7.

The streamwise temperature change is plotted in Fig. 8. The data is captured from 1 mm below the plate surface. Firstly the plate is slowly cooled down during passing through the residual water cooling zone (first residual water

Fig. 7. Temperature profiles in plate thickness direction at various streamwise positions.

Fig. 8. Cooling history at position 1 mm below plate surface along the downstream.

Fig. 9. Surface heat flux distribution along the downstream.

zone) where the jet cooling does not start yet. In the impinging jet cooling zone, the plate is cooled rapidly. After the zone, the plate experiences the residual water cooling (second residual water zone) again. But in this zone the plate temperature near the surface is retrieved as shown in Fig. 8. In Fig. 9, the surface heat flux is depicted along the stream-wise direction. The heat flux is violently fluctuated by passing through the impinging jets. The average heat flux is about 2.7 MW. Although the value seems big, it is reasonable since the present case is confined in the very high temperature region about 1 m from the FM exit. In actual as appeared in the figure, the fluctuating heat flux is slightly decreasing along the downstream path, the first residual water cooling region has a 5 times higher heat flux than the second residual water cooling region.

5. Conclusions

For the cooling process in a moving hot plate, an overall simulation technique coupling the plate motion, the free surface of residual water, and the boiling heat transfer was sug-

gested. Although the present work is limited only for the very high temperature region nearby FM exit, since the plate surface temperature doesn't decrease simply by the repetitive cooling and heat retrieval in the plate, the available region of the present work is expected to be comparatively wide. By considering the thermal interaction between the cooling water and the plate as a part of solutions, the cooling rate which has been regarded as a boundary condition based on experiments for the simulation of plate cooling was directly found. The temperature distributions in the span-wise and plate-thickness directions could be obtained. After finishing the rapid cooling by the impinging jet, the plate's heat retrieval was also found during the simulation. As well, the cooling history of the plate which was very difficult to measure by experiments in the real plant could be analyzed. In the future, to achieve greater completeness for the related physics, the effects of the metallurgical phase change and the corresponding heat generation will be added to the present work.

Acknowledgement

This research was supported by Basic Science Research Program through the National Research Foundation of Korea (NRF) funded by the Ministry of Education, Science and Technology (the Grant 2010-0024619).

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