Numerical simulation on impingement heat transfer characteristics in ultra fast cooling

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Abstract. Numerical simulation was performed to investigate convective heat transfer at the impinging zone of non-immersed pipe jet with Fluent in the ultra fast cooling process, and some influencing factors on heat transfer such as jet direction-to-plate angle, jet pressure (velocity), nozzle-to-impinging plate spacing (height), nozzle interval, nozzle diameter, and surface temperature of the plate were analyzed and designed. The results show that, in the ultra fast cooling process, the reasonable jet direction-to-plate angle is between 30 ° and 45 °; the economic water pressure is generally about 0.6MPa no more than 1.0MPa; since nozzle-to-impinging plate spacing (height) has little influence on the heat transfer coefficient due to the large water speed, nozzle-to-impinging plate spacing (height) is generally adjusted between 300 and 400mm; the nozzle interval should be set properly according to the dynamic relationship of the nozzle diameter and the jet acting zone, besides the maximum average heat transfer coefficient and the most efficient heat transfer will be got while nozzle interval is between 30mm and 40mm under the certain flow rate; considering the efficiency and the uniformity of heat transfer, the reasonable nozzle diameter should be set at 4mm. The model accuracy and correctness are proved by experiments under different test conditions, which show that simulation results are reliable and reasonable.

1. Introduction
The first set of ultra fast cooling (UFC) laboratory equipment in the world was developed by Hoogovens-UGB , which was installed 3 groups of headers in the cooling zone of 1.4m with water flow rate of 1000m3 h, and the cooling rate could reach 900 °C s⁻¹ for steel strip with thickness of 1.5mm. Moreover, the shape of the strips was little affected by the strong cooling due to the fact that ultra fast cooling was uniform along the strip width and length[1]. In 1990s, the ultra fast cooling system for hot steel strip was first developed by CRM of Belgium, which could realize a cooling rate up to 400°C s⁻¹ for hot steel strip with thicknesses of 4mm. JFE’s Fukuyama Works completed installation of the Super-OLAC (on-line accelerated cooling for hot strip) system, which could attain a cooling rate of 700 °C s⁻¹ for hot steel strips with thickness of 3mm. The technology applied successfully in the production of medium plate in 1980 was a typical representative of the successful applications of ultra fast cooling technology.

The greatest characteristic of ultra fast cooling is that it avoids transition boiling and film boiling and achieves a nucleate boiling, so it can nearly reach the limit of the cooling rate and achieve a higher cooling uniformity[2]. In the early 21st century, under the guidance of new generation TMCP technology [3,4], the advanced steel plate cooling devices which give priority to using jet impingement cooling technology have been developed and applied.
In order to make ultra fast cooling play an important role in hot-rolled strip steel process optimization and product development, it has a great significance to conduct a research both on heat transfer characteristics and heat transfer coefficient of impinging jet ultra fast cooling device. Jet impingement is the major heat transfer aspect of ultra fast cooling, and numerous works concerning jet impingement have been done. APS Kumar et al. calculated surface heat flux and surface temperature of strip during ultra fast cooling by inverse heat transfer method, which boundary conditions are estimated from the temperature data in the strip from two thermocouples fitted inside\[5\]. Chester et al. invested the effect of inclination angle and flow rate on the heat transfer of bottom water jet impingement on a hot strip\[6\]. Woodfield et al. conducted experiments to investigate jet impingement quenching for cylindrical block, and the effect of jet velocity, material, initial block temperature of specimen on the width of the boiling region were also analyzed\[7\]. The members of Institute of Materials Structure affiliated with Central Iron and Steel Research Institute, Ning-qi Peng et al. have established a two-dimensional numerical model of coupled thermo mechanical based on ANSYS software and analysed the influence of convective heat transfer coefficient on the properties of steel under ultra fast cooling\[8\].

Although much research has done for jet impingement, the previous work don’t conduct a comprehensive study on how the jet characteristic parameters affect heat transfer coefficient in ultra fast cooling. Numerical simulation was performed to investigate ultra fast cooling, and some influencing parameters on heat transfer characteristics such as jet direction-to-plate angle, jet pressure (velocity), nozzle-to-impinging plate spacing (height), nozzle interval, and nozzle diameter were analysed and designed.

2. Numerical solution method

2.1. Governing equations

Though Jet nozzles of the ultra fast cooling distribute along the width of the plate evenly when the steel plate is under controlled cooling, there is a certain jet angle between nozzles and the plate surface, therefore, a 3D model must be utilized to simulate the realistic flow as shown in Figure 1. The main geometric parameters (as shown in Figure 2) include nozzle diameter D, jet height H (nozzle-to-impinging plate space), jet direction-to-plate angle α and so on.

![Figure 1. 3D geometric model of ultra fast cooling](image1.png)

![Figure 2. Front view of 3D geometric model](image2.png)

In this section, the flow of the fluid is assumed unsteady and the fluid (water) is considered incompressible. For an incompressible fluid, the continuity, momentum and energy equations can are as follows:

\[
\frac{\partial U_i}{\partial x_i} = 0
\]

\[
\frac{\partial (\rho U_i)}{\partial t} + U_j \frac{\partial (\rho U_i)}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \frac{\mu}{\rho} \left( \frac{\partial U_i}{\partial x_j} + \frac{\partial U_j}{\partial x_i} \right) - \frac{\rho}{\rho^2} \partial U_i
\]

\[
\frac{\partial (\rho T)}{\partial t} + U_j \frac{\partial (\rho T)}{\partial x_j} = \frac{\partial}{\partial x_i} \left( \frac{\lambda}{c_p} \frac{\partial T}{\partial x_i} - \frac{1}{\rho} \frac{\partial}{\partial x_i} (\rho v_i) \right)
\]
Where $\rho$ is density of fluid, $\mu$ is molecular cinematic viscosity, $P$ is the mean pressure, $\lambda$ is conductivity; $c_p$ is specific heat at constant pressure; $S_{ij}$ is deformation rate tensor, $U_i$ is the mean velocity vector, $-\rho \mu_i \mu_j$ and $-\rho \mu_i v_j$ are Reynolds stress tensor and heat flux of turbulence respectively.

2.2. Turbulence modeling
Accurate modeling of turbulence phenomenon is essential in numerical simulation of jet impingement. By comparing advantages and drawbacks of different turbulence models, Zuckerman and Lior concluded that the $\kappa$-$\varepsilon$ and $\kappa$-$\omega$ based models dramatically over predicted heat transfer, whereas the $\nu^2$-$f$ model could yield accurate results [9]. Moreover, Behnia et al. concluded that the results of numerical simulation by $\nu^2$-$f$ model were in good agreements with experimental data [10]. Therefore, the $\nu^2$-$f$ model is utilized in the present study.

2.3. Mesh generation
In the cylindrical jet area, grids are divided by cooper method. In the jet impinging area of the ultra fast cooling, the size of the grids is 0.2mm (or smaller). The jet appears a phenomenon of diffusion during the jet impinging due to the gravity. Therefore in the cone-shaped area, a local grid refinement is presented (as shown in Figure 3). The computational area of jet impinging steel plate is designated larger properly. The elliptical area which is away from both sides of the geometric center of the x-axis about -25~50mm is selected to be the computational area due to the fact that nozzles are densely arranged in the cooling zone.

2.4. Boundary conditions
Based on the numerical processing assumptions, the boundary conditions are set as follows:

1) Entrance boundary conditions: the flat along the nozzle exit just as shown in Figure 2 (the entrance of calculation model) is set as the jet velocity inlet, and the inlet temperature of the water jet is 27℃.

2) The boundary conditions of the steel plate: the plate is unmovable and the initial temperature is 1000 ℃.

3) The boundary conditions of the air area: the pressure of the external node in the cylinder area is set to be the atmosphere pressure, and each component velocity of the boundary grid point is assumed to be zero at the initial state [11].

4) The outlet boundary condition: outlet is open to the atmosphere.

2.5. Numerical processing methods
The heat transfer characteristics of water jet impinging obliquely onto steel plate is studied by numerical simulation with the commercial software ANSYS Fluent. The $\nu^2$-$f$ turbulence model was coded independently as a subroutine and was compiled to Fluent code. The governing equations including momentum, energy and $\nu^2$-$f$ equations are discretized by second-order upwind scheme, and discretized equations are solved using the simple algorithm. The mixture multiphase flow model was selected, and the relaxation factors of pressure, momentum and volume fraction were set as 0.5, 0.2 and 0.4 respectively. The solution is considered convergent when the normalized energy residual reach to be less than $10^{-6}$, and residuals of other variables are less than $10^{-4}$. 
3. Results and Discussion
Considering the practical situations of ultra fast cooling in the production, the geometric boundary conditions and initial conditions of numerical model cover the applications of the steel plate under controlled cooling. The boundary conditions and initial conditions of the specific numerical model are as follows:
Jet direction-to-plate angle $\alpha$ ranges from $20^\circ$ to $90^\circ$;
Jet pressure (or velocity) $P$ ranges from 0.3 to 1.0 MPa;
Nozzle-to-impinging plate spacing (height) $H$ ranges from 300 to 600mm;
Nozzle diameter $D$ ranges from 2 to 5mm;

3.1. Effect of jet angle on heat transfer
Setting the nozzle height $H$, nozzle diameter $D$, plate temperature $T_p$, water temperature $T_W$ and jet entrance velocity $V$ 500mm, 5mm, 950$^\circ$C, 27$^\circ$C and 30m/s respectively while the jet angle varies from $20^\circ$ to $90^\circ$, the curves of convective heat transfer coefficient along $x$-axis can be attained just as shown in Figure 4(a). Figure 4(b) is the schematic diagram which shows the average heat transfer coefficient of the specified area at different jet angles.

Figure 4(a) shows that when the jet angle changes from $90^\circ$ to $20^\circ$, the symmetrical shape of the convective heat transfer coefficient gradually becomes asymmetrical shape; when the jet angle ranges from $20^\circ$ to $30^\circ$, the distribution of the heat transfer coefficient is the most flat and uniform.

Figure 4(b) shows that the average convective heat transfer coefficient presents downward trend as jet angle increases in the jet stagnation zone.

3.2. Effect of jet pressure on heat transfer
Setting the nozzle height $H$, nozzle diameter $D$, plate temperature $T_p$, water temperature $T_W$ and jet angle $\alpha$ 500mm, 5mm, 950$^\circ$C, 27$^\circ$C and 30$^\circ$ respectively while jet pressure $P$ is set as 0.3, 0.45, 0.6, 0.8, 1.0 MPa($v=24.5,30.0,34.6,40.0,44.7$m/s) respectively, the curves of convective heat transfer coefficient along $x$-axis can be attained just as shown in Figure 5(a). Most of the jet water flows to the positive $x$-axis when the water is sprayed to the steel plate obliquely except the jet angle $90^\circ$ in ultra fast cooling.
Since the water flows forward due to the jet angle, there is little water flowing to the negative $x$-axis. Therefore, the convective heat transfer coefficient here is too small to be considered. For the sake of reflecting the influence of the geometric and initial parameters on the heat transfer characteristics more obviously, this paper takes the heat transfer coefficient of $x=0.02$m (in the positive $x$ axis) to show the relationship between the heat transfer characteristics and the parameters. The curve of the convective heat transfer coefficient under different jet pressures is shown in Figure 5(b).

Figure 5(a) and Figure 5(b) show that heat transfer coefficient increases with the growing of the jet pressure in the direction of positive $x$-axis because the increasing of the jet kinetic energy and the turbulence enhance the heat transfer greatly. When the jet pressure ranges from 0.1MPa to 0.6MPa, the increase of the convective heat transfer coefficient is the fastest. In ultra fast cooling process, the economic water pressure is generally about 0.6MPa no more than 1.0MPa.
3.3. Effect of nozzle-to-impinging plate spacing (height) on heat transfer

Setting the jet angle $\alpha$, nozzle diameter $D$, plate temperature $T_p$, water temperature $T_W$ and jet entrance velocity $V$ 30°, 5mm, 950°C, 27°C and 30m s$^{-1}$ respectively while the nozzle height $H$ is set as 300, 400, 500, 600mm respectively, the curves of convective heat transfer coefficient along x-axis can be attained as shown in Figure 6(a). The curve of convective heat transfer coefficient under different nozzle heights of $x=0.02$ is just as shown in Figure 6(b).

It can be seen that convective heat transfer coefficient increases with the jet height in Figure 6(a) because gravity plays a role of accelerating the velocity of the jet water. Figure 6(b) shows that the nozzle height is almost positively correlated with convective heat transfer coefficient in the positive x-axis direction. However, if the nozzle height is too high, it will influence the arrangement of the nozzles. Considering all of the factors, the jet height should be adjusted between 300 and 400mm.

3.4. Effect of nozzle interval on heat transfer

The interference of the flows has influence on heat transfer characteristics while multiple flows work on the steel plate simultaneously, so the computational model establishes a whole jet and adds a half on both sides of the symmetrical positions just as shown in Figure 7(a). Figure 7(b) is the schematic diagram of the 3D geometric model built with Gambit pre-processing software. The jet height is 500mm and the jet angle is 30° in the model.

The flow rate of each jet is positively correlated with jet velocity and sectional area respectively and with the increasing of the nozzle diameter, the nozzle interval increases and the number of the nozzles decreases based on a certain flow rate. The relations of the nozzle interval with the average heat transfer coefficient and nozzle diameter are shown in Figure 8.

Figure 8 shows that the average heat transfer coefficient increases firstly and then decreases with the growing nozzle interval. The flow rate of each single jet increases with the growing nozzle diameter in the case of the same total flow rate of the nozzle block, but the nozzle interval should be increased according to the increasing diameter that means the average heat transfer coefficient decreases with the
increase of the acting area. The average heat transfer coefficient achieves a balance between the nozzle diameter and the square of the jet acting zone, therefore the nozzle interval must be set exactly and only by this can the device achieve the best effect of heat transfer. If nozzle interval is between 30 and 40mm and nozzle diameter is around 4mm, the average heat transfer coefficient reaches the maximum and the water jet heat transfer is the most efficient just as shown respectively in Figure 8.

Figure 7. Multi-jets geometrical sketch

Figure 8. Relationships of nozzle spacing with nozzle diameter and average heat transfer coefficient

3.5. Effect of nozzle interval on heat transfer

Setting the nozzle height H, plate temperature Tp, water temperature TW, jet angle α and entrance velocity V 500mm, 950°C, 27°C, 30° and 30m s⁻¹ respectively while nozzle diameter D is set as 2, 3, 4, 5mm respectively, the curves of convective heat transfer coefficient along x-axis can be attained just as shown in Figure 9(a). The curve of convective heat transfer coefficient under different nozzle diameters of x=0.02 is shown in Figure 9(b).

Figure 9(a) shows that the convective heat transfer coefficient grows with the increase of the nozzle diameter due to the increase of the flow rate and the turbulence of the fluid, which lead the heat transfer to be strengthened.

Figure 9(b) clearly reflects the relation of the nozzle diameter and the convective heat transfer coefficient in the positive x-axis direction. When nozzle diameter D is less than 3mm, the increment of the convective heat transfer coefficient is relatively large, and when the nozzle diameter D is more than 3mm, the increment of the heat transfer coefficient decreases. Therefore, in practical applications, the simple increase of the flow rate (or nozzle diameter) cannot achieve the desired purpose. When the nozzle diameter is between 4mm and 5mm, the distribution of the heat transfer coefficient is more flat and uniform. It can be found that the larger nozzle diameter is, the larger flow rate it has and the more it costs. It is very reasonable to set nozzle diameter around 4mm in ultra fast cooling process by fully considering the uniformity and the effect of the heat transfer, which agrees well with the results of chapter 3.4.
4. Experimental Verification

In order to demonstrate the accuracy of the heat transfer coefficient got by the numerical methods, cooling experiments have been done on the Q235 steel sample with the thickness of 14mm, length of 285mm and width of 100mm. The temperature of steel is obtained by thermocouples which are positioned at the different depths of steel sample (temperature measuring point A and B in the Figure 10). The sample is heated to around 1100 °C in the furnace and then is cooled under three different test conditions whose process parameters are listed in the Table 1. The temperature changes of point A and point B (as shown in the Figure 11) during cooling experiments are measured. The signals of thermocouples are received at 1s intervals, the error for the arrangement of thermocouples is estimated to be ±0.1mm and the error of the temperature measurement is ±0.1°C.

The heat transfer coefficients calculated by Fluent software above were used in the numerical simulation of steel temperature field, and the differences between calculated temperature and measured temperature of steel could be used to assess the validation of heat transfer coefficient got by numerical simulation.

Figure 11, Figure 12 and Figure 13 show the curves of the temperature-time history under condition 1, condition 2 and condition 3 respectively. Three figures indicate the comparison of calculated and measured values. As shown in the figures, the calculated values and measured values are in good agreement with each other, which demonstrates the validation of heat transfer coefficient got by numerical simulation. Thus it can be seen that simulation results of this paper are reliable and reasonable.

Table 1

| Process parameters of three conditions | Condition 1 | Condition 2 | Condition 3 |
|---------------------------------------|-------------|-------------|-------------|
| Jet angle (°)                         | 45          | 45          | 30          |
| Jet pressure (MPa)                    | 0.6         | 0.6         | 0.6         |
| nozzle height (mm)                    | 300         | 400         | 400         |
| nozzle diameter (mm)                  | 4           | 4           | 4           |
nozzle interval (mm) | 40 | 35 | 30
--- | --- | --- | ---

| Time/s | 0 | 2 | 4 | 6 | 8 | 10 | 12 | 14 | 16 | 18 | 20 | 22 | 24 | 26 | 28 | 30 |
| --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- | --- |
| Temperature/℃ | | | | | | | | | | | | | | | | |

Figure 11. Comparison of calculated and measured temperatures under condition 1

Figure 12. Comparison of calculated and measured temperatures under condition 2

Figure 13. Comparison of calculated and measured temperatures under condition 3

5. Summary and Conclusions
Numerical simulation and experiments were performed to investigate the jet flow and the heat transfer of ultra fast cooling, and some influencing parameters on heat transfer characteristics had been analysed. The main conclusions are as follows:

(i) Each row of high-pressure nozzles covers many dense water outlets and the space between outlets is very small in ultra fast cooling device. Tilting angle should be increased properly to reduce the setting difficulties. In addition, considering the arrangement of the nozzle interval and the distance between the roller tables, the reasonable jet angle is between 30° and 45°.

(ii) In the process of production, it is not economic to improve the heat transfer capability by simply increasing the pressure because it will greatly increase the costs and the usual practice is to minimize
the water pressure on condition that it could fully meet the needs of the cooling process. In ultra fast cooling, the economic water pressure is generally about 0.6MPa no more than 1.0MPa.

(iii) When the nozzle height is small, the heat transfer coefficient is small; but if the nozzle height is too high, it will greatly influence the arrangement of the nozzles of ultra fast cooling device. Taking all of the factors into consideration, the nozzle height should be between 300 and 400mm.

(iv) The nozzle interval should be set properly according to the dynamic relationship of the nozzle diameter and the jet acting zone, besides the maximum average heat transfer coefficient and the most efficient heat transfer will be got while nozzle interval is between 30mm and 40mm under the certain flow rate.

(v) With the increasing of the nozzle diameter, the heat transfer coefficient increases. With the nozzle diameter increasing, the larger flow rate it has and the more it costs. It is very reasonable to set nozzle diameter around 4mm in the ultra fast cooling process.

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