Numerical Analysis for Jet Impingement and Heat Transfer Law of Self-Excited Pulsed Nozzle

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The nozzle is the key component of ultra-fast cooling equipment in hot-rolling steel industry, which is crucial for improving the cooling performance. In this paper, in order to optimize the TMCP ultra-fast cooling technology, a self-excited pulsed nozzle was applied into the ultra-fast cooling equipment, and its cooling performance of jet impingement was studied. The flow states and heat transfer characteristics on the surface of the 840°C steel plate, which were impinged by the conventional cylindrical convergent nozzle and self-excited pulsed nozzle formed by adding a Helmholtz oscillating chamber, were simulated by using ANSYS-Fluent under the same inlet pressure, respectively. The maximum jet velocities, dynamic pressures, outlet flow rates of the two nozzles, the temperatures and heat fluxes of the plate surface were monitored. The results showed that, under the pressure of 0.8 MPa, the average outlet flow of the self-excited pulse nozzle was lower than that of the cylindrical convergent nozzle, whilst the self-excited pulse nozzle had higher instantaneous outlet velocity and dynamic pressure. The self-excited pulsed jet could increase turbulence intensity and heat flux on the plate surface. Compared with continuous jet impingement, the self-excited pulsed jet impingement had a better heat transfer effect with lower energy input. The results of the study can provide data support for nozzle designing and better application of ultra-fast cooling equipment.

KEY WORDS: ultra-fast cooling; self-excited jet; jet impingement; heat transfer; turbulence intensity.

1. Introduction

Steel has many advantages, such as abundant resources, low cost, stable performance, and recyclability. It is used as an important structural material in different walks of life. The new generation of controlled rolling and controlled cooling technology based on ultra-fast cooling (UFC) is one of the most effective technologies to produce high-quality hot-rolled products.^{1–4)} The UFC equipment impinges cooling water of 0.4–0.8 MPa onto plate surface of high-temperature through densely arranged nozzles to make steel quickly pass through austenite phase zone, and “freeze” hardened austenite to the dynamic transformation temperature of austenite to ferrite.^{5–8)} Increasing the cooling ability of UFC equipment can improve production efficiency and make the production of high-quality steel products possible. Lots of research had been done to improve the heat transfer performance of jet impingement. Ai *et al.*⁹⁾ found that reducing nozzle diameter could contribute to the improvement of heat transfer performance at the same outlet flow. Chester and Hauksson *et al.*^{10,11)} found that increasing flow rate and velocity could improve the heat transfer

ability of the hot plate surface. Modak *et al.*¹²⁾ found that the use of CuO nanofluids could enhance efficiency of jet impingement cooling. When the volume fraction of CuO nanoparticles was 0.15% and 0.60%, the cooling efficiency of CuO nanoparticles was 32.6% and 112.8% higher than that of pure water, respectively.

In the early 1980s, self-excited pulsed jet was discovered, which was developed based on transient flow, acoustic and fluid resonance theory. It had attracted much attention due to its high instantaneous pressure and cavitation effect. It achieved excellent impact performance without external excitation source.^{13,14)} Morel¹⁵⁾ carried out jet impingement experiments on Helmholtz oscillating nozzle. It was found that the amplitude of outlet velocity was increased by 60% when jet velocity frequency was close to the natural frequency of the oscillating chamber. Huang *et al.*^{16,17)} applied a Helmholtz injector in oil and gas exploitation and found that the erosion rate of the self-excited oscillating jet was higher than that of the continuous jet. Li *et al.*¹⁸⁾ found that self-excited pulsed jet produced by the Helmholtz nozzle had greater pressure fluctuation and impact pressure peak than continuous jet produced by 120° conical nozzle. Zhou *et al.*¹⁹⁾ found that the peak pressure of the self-excited pulsed jet was 2.5 times higher than that of the ordinary continuous jet.

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Ten and Povey²⁰⁾ found that the heat transfer capacity of the pulsed jet was 20% higher than that of continuous jet at the same average mass flow rate when they applied a self-excited pulsed oscillator to the cooling process of a gas turbine. Gao and Zeng²¹⁾ found that the convective heat transfer coefficient (HTC) in a tube can be increased by 10%–40% after using self-excited pulsed jet. Lee *et al.*²²⁾ studied the effect of pulsed jet on the fluid mixing and heat transfer in the corrugated pipe. Lee and Nishimura *et al.*^{22,23)} studied the heat transfer characteristics of the pulsed jet in the groove channel, and the heat transfer effect can be enhanced by pulsed jet was proved. Haneda and Tsuchiya²⁴⁾ found that flow oscillation can enhance the local heat transfer and make Nusselt number become uniform over the stagnation region on the target plate. Baffigi and Bartoli²⁵⁾ found that ultrasonic wave can enhance the heat transfer from cylinder to distilled water.

Some studies^{26–30)} showed that the turbulence intensity on the target plate was a significant factor affecting the cooling performance of jet impingement. Improving the turbulence intensity can effectively improve the cooling ability of jet impingement. However, whether self-excited pulsed jet can enhance the turbulence intensity on the impacted surface and improve the heat transfer ability of high-temperature steel materials has not been reported. In this paper, the flow characteristics of cooling water and the heat transfer performance on the plate surface were compared between self-excited pulsed jet and conventional continuous jet, influence laws of the two nozzles were obtained, which would provide references for the development and application of UFC equipment.

2. The Heat Transfer Process of Jet Impingement and the Physical Model

2.1. The Heat Transfer Process of Jet Impingement

Free jet impingement cooling is one of the important cooling methods in ultra-fast cooling technology. The left half of Fig. 1 shows the flow process of cooling water during free jet impingement. When cooling water jet impacts

on the plate surface, a stagnation zone is formed at first. It is a transition zone where high-pressure water flows from axial to radial direction along the plate surface. Wall jet zone is formed when cooling water develops along the radial direction. Part of kinetic energy is converted into potential energy, which makes liquid level rise and produces hydraulic jump. The continuous impact of cooling water makes wall jet zone expand gradually, and pushes the hydraulic jump to move along radial direction until the whole plate surface is wetted. The right half of Fig. 1 shows the heat transfer mechanism between cooling water and hot surface during jet impingement. When cooling water impinged onto plate surface, it could go through the following fourth stages: bubble generation and growth, steam film formation, steam film rupture, and bubble separation. Stagnation zone has maximum static pressure, bubbles produced by boiling are broken before they grow up, so stable vapor films cannot be formed. Outside the stagnation zone, the farther distance from the stagnation zone is, the stronger the bubble growth ability is, and the easier it is to form stable steam films. The ability to sweep vapor film largely depends on radial flow ability of the cooling water.

2.2. Composition of Physical Model

Figure 2(a) shows the internal structure of the self-excited pulsed nozzle for research, which was composed of upstream nozzle and downstream nozzle. The inlet diameter (d_0), convergent angle (θ_1), and outlet diameter (d_1) of the upstream nozzle were 6 mm, 30° , and 2 mm, respectively. A circular arc transition with radius of 1.5 mm was adopted at the entrance of the upstream nozzle. The downstream nozzle had an outlet diameter d_2 of 2.4 mm, d_2/d_1 of 1.2, and a cone angle of the lower collision wall θ_2 of 120° . The diameter of the oscillating chamber was D of 16 mm and the length of the oscillating chamber was L of 3 mm. Figure 2(b) shows a conventional cylindrical convergent nozzle compared with the above self-excited pulse nozzle, whose structural parameters were identical to the upstream nozzle of the self-excited pulsed nozzle.

The high-temperature steel plate was 45 steel (Chinese industrial standard), with a size of 150 mm $\times$ 30 mm $\times$ 20 mm. The composition of 45 steel used in research were C-(0.42–0.5)-Si-(0.17–0.37)-Mn-(0.5–0.8)-0.25Cr wt%. Specific heat capacity and heat conductivity of 45 steel at

Fig. 1. Schematic diagram of jet impingement process.

Fig. 2. Internal structure of the two nozzles. (a) Self-excited pulsed nozzle. (b) Cylindrical convergent nozzle.

Table 1. Conductivity and specific heat of 45 steel.

Temperature °C	Specific heat J/(kg·K)	Conductivity W/(m·K)
20	472	58
100	480	50.7
200	498	48.1
300	524	45.7
400	560	41.7
500	615	38.3
600	700	33.9
700	854	30.1
755	1 064	25.1
800	806	27.5
900	637	26.2

different temperatures are listed in **Table 1**. During the simulation, thermophysical parameters were obtained by piecewise linear fitting according to the data in Table 1. In the simulation, the distance from the nozzle outlet to the plate surface was H of 200 mm.

3. Numerical Analysis

3.1. Simulation Method

Three-dimensional numerical simulation was selected to calculate flow profiles and heat transfer characteristics with ANSYS-Fluent. When simulating flow profiles in the two nozzles, inlet pressure was selected as 0.8 MPa, which was a common pressure of the front section in ultra-fast cooling equipment. In the specified pressure, the flow status in nozzles is turbulent with Reynolds numbers more than 4 000, and high simulation accuracy can be achieved by using the standard k - ε two-equation model. So the standard k - ε model in Reynolds Average Navier-Stokes (RANS) model was used to simulate flow profiles in the cylindrical convergent nozzle. However, due to the existence of unstable vortices in the self-excited pulsed nozzle, RANS is not suitable to solve large-scale eddy effects in the process of imbalance and instability accurately. By contrast, Large Eddy Simulation (LES) model in turbulence model can decompose variables into filtered parts and small-scale vortex parts, and solve vortex effects at all scales in a certain range. So LES model was selected to calculate flow profiles in the self-excited pulsed nozzle. The boundary conditions of the two nozzles were set as pressure-inlet and pressure-outlet. Water was set as the working fluid. The temperature of the water was 297 K. The total calculation time was 2 seconds.

Jet impingement cooling is a typical two-phase thermal fluid-solid coupling problem with water and air. The VOF model in multiphase flow can effectively solve the unsteady problem in which there was no large mixing between two phases. And the precision of the explicit algorithm was higher than that of the implicit algorithm in VOF model while calculating the cooling process. So explicit algorithm in VOF model was applied to simultaneously calculate both fluid and temperature fields on the plate surface. Explicit algorithm of VOF model cannot be simultaneously used

with LES model in calculating the self-excited heat transfer process. In order to maintain the pulsing effect of the self-excited pulsed jet, outlet velocity of the nozzle calculated by LES model was taken as velocity-inlet of the self-excited pulsed jet heat transfer process while calculating. For continuous jet, the inlet boundary condition of heat transfer process was still defined as pressure-inlet.

Fluid-solid interface was used as a coupling wall, the standard k - ε model was selected as turbulence model, and evaporation-condensation mechanism was activated to calculate the vaporization process. Pressure-velocity coupling method was used to solve parameters, and pressure-outlet was selected as the outlet of the model. The working fluid was water with temperature of 297 K. Time step was $1e^{-4}$ seconds. Total calculation time was 10 seconds. In the evaporation-condensation mechanism, the evaporation frequency and condensation frequency were assumed to be 0.1 and the saturation temperature was assumed to be 373.15 K. In addition, the following assumptions were made: initial temperature distribution of the steel plate was uniform, the heat transfer of steel plate was isotropic, radiation heat transfer of steel plate was ignored and there is no oxide scale on the plate surface.

3.2. Governing Equations of Turbulence Models

The k and ε equations of the standard k - ε model in the RANS model are:

$$\frac{\partial(\rho k)}{\partial t} + \frac{\partial(\rho k u_i)}{\partial x_i} = \frac{\partial}{\partial x_i} \left[\left(\mu + \frac{u_t}{\sigma_k} \right) \frac{\partial k}{\partial x_i} \right] + G_k - \rho \varepsilon \quad \dots (1)$$

$$\begin{aligned} \frac{\partial(\rho \varepsilon)}{\partial t} + \frac{\partial(\rho \varepsilon u_i)}{\partial x_i} = & \frac{\partial}{\partial x_i} \left[\left(\nu + \frac{u_t}{\sigma_\varepsilon} \right) \frac{\partial \varepsilon}{\partial x_i} \right] + C_{1\varepsilon} \frac{\varepsilon}{k} G_k - C_{2\varepsilon} \rho \frac{\varepsilon^2}{k} \quad \dots (2) \end{aligned}$$

Where k is turbulent kinetic energy, ε is turbulent dissipation rate; u_i and u_j are velocity components in x and y directions, respectively; x_i and x_j are coordinate positions in x and y directions, respectively; ν is kinematic viscosity of flow field and ρ is density of liquid; $C_{1\varepsilon}$, $C_{2\varepsilon}$, σ_k , σ_ε are 1.44, 1.92, 1.0 and 1.3, respectively.

In addition, u_t is the turbulent viscosity coefficient and G_k is the turbulent kinetic energy generated by the average velocity gradient, they can be further written as Eqs. (3) and (4):

$$u_t = \rho C_\mu \frac{k^2}{\varepsilon} \quad \dots (3)$$

$$G_k = -\rho \overline{u_i' u_j'} \frac{\partial u_j}{\partial x_i} \quad \dots (4)$$

Where $-\overline{\rho u_i' u_j'}$ is Reynolds stress and $C_\mu=0.09$.

The governing equation of the LES model is the spatially filtered N-S equation:

$$\frac{\partial \rho}{\partial t} + \frac{\partial(\overline{\rho u_i})}{\partial x_i} = 0 \quad \dots (5)$$

$$\frac{\partial(\rho \bar{u}_i)}{\partial t} + \frac{\partial(\rho \bar{u}_i \bar{u}_j)}{\partial x_j} = \frac{\partial}{\partial x_j} \left(\mu \frac{\partial \bar{u}_i}{\partial x_j} \right) - \frac{\partial \bar{P}}{\partial x_i} - \frac{\partial \tau_{ij}}{\partial x_j} \dots (6)$$

Where $\bar{u}_i$, $\bar{u}_j$ are filtered velocity in x and y directions, respectively; $\bar{P}$ is fluid pressure; τ_{ij} is subgrid-scale stresses, whose main function is to filter out momentum transport between small-scale vortices and large-scale vortices.³¹⁾

However, filtered subgrid-scale stress tensor τ_{ij} is unknown. In order to make Eq. (6) closed, subgrid-scale model needs to be established. At present, the most widely used subgrid-scale model is Smagorinsky-Lilly model can be seen in Eq. (7):

$$\tau_{ij} = \frac{1}{3} \tau_{kk} \delta_{ij} - 2\mu_t \bar{S}_{ij} \dots (7)$$

For incompressible flow, τ_{kk} can be neglected, it can be defined as $\tau_{kk} = \gamma M_{sgs}^2 \bar{P}$; δ_{ij} is Kroenecker delta; the eddy-viscosity μ_t and rate of strain tensor $\bar{S}_{ij}$ can be written as:

$$\mu_t = \rho L_s^2 |\bar{S}| \dots (8)$$

$$\bar{S}_{ij} = \frac{1}{2} \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i} \right) \dots (9)$$

The mixing length for subgrid scales L_s can be defined as $L_s = \min(\kappa d_w, C_s \sqrt{V})$, where κ is VonKarman constant; d_w is the distance from the calculation unit to the closest wall; V is volume of calculation unit; $C_s=0.1$. The rate of strain tensor $|\bar{S}|$ can be defined as $|\bar{S}| = \sqrt{2\bar{S}_{ij}\bar{S}_{ij}}$. It can be seen that τ_{ij} is determined by rate of strain tensor for the resolved scale due to the small change of L_s .

4. Results and Discussion

4.1. Maximum Dynamic Pressure, Outlet Flow and Maximum Flow of the Two Nozzles

Figure 3 shows the dynamic pressure inside the two nozzles within 1 second after jet started. It can be seen that the average dynamic pressure in the self-excited pulsed one was 1.15 times than that of the cylindrical convergent one.

Fig. 3. Maximum dynamic pressure comparisons of the two nozzles under the same inlet pressure.

The oscillation of the pressure caused the change of the velocity at the nozzle outlet.

Figure 4 shows the curves of the maximum jet velocities within 1 second after jet started. It is clearly observed that the jet velocity of the self-excited pulsed nozzle was pulsating, and the peak velocity was 48 m/s, which was significantly higher than that of the cylindrical convergent one, but the average velocity of the self-excited pulsed nozzle was only 36.4 m/s, which was lower than the 39.3 m/s of the cylindrical convergent one.

Figure 5 shows the radial distributions of the average jet velocity at the outlet of the two nozzles. It is clearly shown that velocities at the outlet of the cylindrical convergent nozzle were the same substantially. As for the self-excited pulsed nozzle, the maximum velocity was at the center of the outlet. With the increase of the distance from the center, the velocity decreased gradually, the velocity deviation from the cylindrical convergent nozzle increased gradually.

When the inlet pressures were 0.8 MPa, the average outlet flow rates of the two nozzles are shown in Table 2.

Fig. 4. Maximum jet velocities under the same inlet pressure.

Fig. 5. The radial distribution of average velocity at nozzle outlets.

The average flow of the self-excited pulsed nozzle was only 84.6% of that of the cylinder convergent one.

4.2. Comparison of Flow Characteristics on the Steel Plate Surface

The increase of turbulence intensity and shear force on the plate surface was helpful to promote the exchange of momentum and heat in the boundary layer, and to remove the stagnant water and steam film on the plate surface, so as to enhance the heat transfer capability of jet impingement. Where turbulence intensity is defined as the ratio of the root-mean-square of the velocity fluctuations to the mean flow velocity. Due to the pulsating characteristics of the self-excited pulsed jet, the turbulence intensity and shear force on the coupling surface changed with time. While for the continuous jet, the above two variables were approximately constant. **Figure 6** shows the dynamic changes of

Table 2. The average outlet flow of the two nozzles.

Nozzle parameters	Self-excited pulsed nozzle	Cylindrical convergent nozzle
Nozzle inlet pressure P_i /MPa	0.8	0.8
Flow Q /L/min	6.02	7.12

Fig. 6. Comparison of turbulence intensity and shear force. (Online version in color.)

turbulence intensity and shear force at 5 mm away from the jet center within 5 seconds after jets stability.

Figure 7 shows the turbulence intensity and shear force distributions of the two jets on the coupling surface, where the data of the self-excited pulsed jet was the average value in 20 consecutive time steps. With the increase of the distance from the jet center, turbulence intensities and shear forces of the two jets showed a trend of increasing rapidly to the peak value and then decreasing gradually. The turbulence intensity of the self-excited pulsed jet was obviously higher than that of the continuous one in the impacted zone, while it was almost the same in the wall jet zone. The shear force of the continuous jet in the stagnation zone was larger than that of the self-excited pulsed one, while the other regions were the opposite.

4.3. Influence of Flow Characteristics on Heat Transfer Coefficient

Figure 8 shows the heat transfer coefficient (HTC) distributions of the two jets on the plate surface, where the data of the self-excited pulsed jet was the average value in 20 consecutive time steps. It shows that the self-excited pulsed jet has better heat transfer performance than the continuous one. On the whole plate surface, the HTC of self-excited

Fig. 8. Comparison of heat transfer coefficient on the coupling surface. (Online version in color.)

Fig. 7. Comparison of turbulence intensities and shear forces on the coupling surface. (a) turbulence intensity. (b) shear force. (Online version in color.)

Table 3. Data of turbulence intensity, shear force and heat transfer coefficient on the plate surface.

Distance from the impingement point (mm)	Self-excited nozzle			Cylindrical convergent nozzle		
	Turbulence intensity (%)	Shear force (Pa)	Heat transfer coefficient ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$)	Turbulence intensity (%)	Shear force (Pa)	Heat transfer coefficient ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$)
0	2.16	4.43	71 041.10	1.17	26.04	61 047.33
5	1.58	58.37	46 289.80	1.24	35.33	32 507.24
10	0.84	25.01	21 605.30	0.73	3.52	15 215.45
15	0.48	12.03	12 341.90	0.46	2.93	9 152.38
20	0.35	8.10	9 184.98	0.37	2.54	7 120.47
25	0.27	5.49	5 654.17	0.31	2.54	4 772.24
30	0.23	4.43	4 582.21	0.28	2.54	4 077.14

pulsed jet is significantly greater than that of continuous jet. Moreover, the HTC distribution of the self-excited pulsed jet in the stagnation zone is more uniform.

Table 3 shows the data of turbulence intensity, shear force and HTC of the two jets at different radial distance from the jet center line. **Table 4** shows the comparison of these data. It can be found that the HTC of self-excited pulsed jet is significantly higher than that of continuous jet within 30 mm from the jet center, which is up to 12.4%–42.4%. At the jet center, the turbulence intensity of self-excited pulsed jet is 85.09% higher than that of continuous jet, while the shear force is 83% lower than that of continuous jet, resulting in the growth rate of HTC by only 16.37%; at 5 mm away from the jet center, the growth rate of turbulence intensity and shear force are 27.14% and 65.23% respectively, and HTC is 42.4%; at 10 mm away from impingement center, the above values are 15.18%, 614.68% and 42%, respectively. When the radial distance from the jet center exceeds 15 mm, the turbulence intensity of self-excited pulsed jet is lower than that of continuous jet, but the HTC of self-excited pulsed jet is still higher due to the huge growth rate of shear force. It can be concluded that HTC is mainly determined by turbulence intensity and shear force. Near the jet center, the effect of turbulence intensity on the HTC is greater than that of shear force, while away from the jet center, shear force has more influence.

4.4. Comparison of Heat Transfer Performance

Three temperature and heat flux monitoring points were set at the jet center (point 1), 10 mm (point 2) and 20 mm (point 3) away from the jet center on the plate surface. **Figure 9** shows the temperature curves of the two jets under the inlet pressure of 0.8 MPa. It can be seen that the cooling rate at the jet center was the fastest, and the farther away from the point, the slower the cooling rate was. **Table 5** shows the average cooling rate of the above three monitoring points within 2 seconds from jet beginning. It is clearly observed from the Table 5 that cooling rate of the self-excited pulsed jet was slightly higher than that of the continuous jet at each monitoring point, and turbulence intensity of self-excited pulsed jet on the coupling surface was stronger, which weakened the thickness of temperature boundary layer and improved the heat transfer efficiency. Although the shear force of the continuous jet formed by the cylindrical convergent nozzle was greater at point 1, it

Table 4. Comparison of flow and heat transfer characteristics of the two jets.

Distance from the impingement point (mm)	Growth rate of the self-excited nozzle relative to the cylindrical convergent nozzle (%)		
	Turbulence intensity (%)	Shear force (Pa)	Heat transfer coefficient ($\text{W}\cdot\text{m}^{-2}\cdot\text{K}^{-1}$)
0	85.09	−83.00	16.37
5	27.14	65.23	42.40
10	15.18	614.68	42.00
15	5.69	314.93	34.85
20	−4.06	223.82	28.99
25	−10.43	119.65	18.48
30	−17.77	77.11	12.38

Table 5. Comparison of the average cooling rates at the monitoring points.

Monitoring point	Self-excited nozzle	Cylindrical convergent nozzle
point 1	315.6	298.9
point 2	257.9	235.7
point 3	162.2	146.9

Fig. 9. Comparison of temperature drop curves at the monitoring points. (Online version in color.)

Fig. 10. Comparison of temperature and heat flux at point 1. (Online version in color.)

Table 6. Data of q_{MHF} and t_{MHF} at the three locations.

Monitoring point	Self-excited nozzle		Cylindrical convergent nozzle	
	q_{MHF} (MW·m ⁻²)	t_{MHF} (s)	q_{MHF} (MW·m ⁻²)	t_{MHF} (s)
point 1	9.26	0.34	8.95	0.3
point 2	6.23	0.54	5.75	0.45
point 3	3.39	1.05	3.16	1

did not increase its cooling rate. It can be concluded that the effect of turbulence intensity on heat transfer performance was greater than that of shear force.

Figure 10 shows the temperature and heat flux curves at point 1. After jet started, the film boiling regime and transition boiling regime ended instantaneously, and the heat flux increased to the maximum value rapidly, and then the cooling process transited to the nucleate boiling regime, the heat flux decreased gradually. The temperature of point 1 kept decreasing during the whole cooling process. The times to reach the maximum heat flux (t_{MHF}) were almost the same, and change trends of the temperature and the heat flux were basically the same.

Table 6 shows the maximum heat flux (q_{MHF}) and time to reach the maximum heat flux (t_{MHF}) at three monitoring points on the steel plate surface. The maximum heat fluxes of the self-excited pulsed jet at three monitoring points were 9.26 MW·m⁻², 6.23 MW·m⁻² and 3.39 MW·m⁻², while the corresponding values of the continuous jet were 8.95 MW·m⁻², 5.75 MW·m⁻², and 3.16 MW·m⁻², respectively. It can be seen from Table 6 that q_{MHF} of the self-excited pulsed jet were 0.31 MW·m⁻², 0.48 MW·m⁻² and 0.23 MW·m⁻² higher than those of the continuous jet at three monitoring points, the relative improvement rates were 3.5%, 8.5%, and 7.3%, respectively. Figure 11 shows the trends of q_{MHF} and t_{MHF} . It can be seen that with the increase of the distance from the impingement point, q_{MHF} decreased and t_{MHF} increased, the jet form had little effect on the change rates of the two variables.

Fig. 11. Comparison of q_{MHF} and t_{MHF} at the three locations.

5. Conclusions

In this paper, the cylindrical convergent nozzle used in the ultra-fast cooling equipment and the self-excited pulsed nozzle formed by adding an oscillating chamber were studied. The jet impingement characteristics and the heat transfer performance of the two nozzles were compared by simulation. The main conclusions of the study are as follows:

(1) The pressure oscillation inside the self-excited pulsed nozzle made its instantaneous maximum outlet velocity higher than that of the cylindrical convergent one, but the radial distribution of the average velocity at the self-excited pulsed nozzle outlet was uneven, resulting in its average outlet flow rate lower than that of the cylindrical convergent one.

(2) The turbulent intensity and shear force on the plate surface was changing when the self-excited pulsed nozzle impacted the plate. The pulsating characteristic of the self-excited pulsed jet can enhance the perturbations of the velocity boundary layer, resulting in a significant increase in the turbulence intensity of the plate surface.

(3) When water impinging on steel plate, HTC is mainly determined by turbulence intensity and shear force on the plate surface. Near the jet center, the effect of turbulence intensity on the HTC is greater than that of shear force, while away from the jet center, shear force has more influence.

(4) The self-excited pulsed nozzle had a better cooling performance of jet impingement than the original nozzle. The maximum heat fluxes of the self-excited pulsed jet at the three monitoring points were 3.5%, 8.5%, and 7.3% higher than those of continuous jet, while the flow consumed was only 84.6% of that of the continuous jet.

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Nomenclature

C_s : Samagorinsky constant
 D : diameter of the oscillation chamber
 d_0 : inlet diameter of the upstream nozzle

d_1 : outlet diameter of the upstream nozzle
 d_2 : outlet diameter of the downstream nozzle
 d_w : distance from the calculation unit to the closest wall
 G_k : turbulent kinetic energy generated by the average velocity gradient
 H : distance from the nozzle outlet to the plate surface
 HTC : heat transfer coefficient
 k : turbulent kinetic energy
 L_s : mixing length for subgrid scales
 M_{sgs} : subgrid-scale Mach number
 P_i : nozzle inlet pressure
 $\bar{P}$: fluid pressure
 Q : nozzle flow
 q_{MHF} : maximum heat flux
 R : transition arc radius of the upstream nozzle
 $|\bar{S}|$: rate of strain tensor
 $\bar{S}_{ij}$: rate of strain tensor
 t_{MHF} : time to reach maximum heat flux
 t : time
 T_m : surface temperature at maximum heat flux
 ν_t : turbulent viscosity coefficient
 u_i, u_j : velocity dimensionless
 $\bar{u}_i, \bar{u}_j$: filtered velocity dimensionless
 V : volume of calculation unit
 x_i, x_j : spatial dimensionless
 θ_1 : convergent angle of the upstream nozzle
 θ_2 : angle of the lower collision wall
 σ_k : turbulent Prandtl number of the k equation
 σ_ε : turbulent Prandtl number of the ε equation
 ν : kinematic viscosity
 μ_t : eddy-viscosity
 ρ : density of liquid
 γ : water viscosity coefficient
 τ_{ij} : filtered subgrid-scale stress tensor
 κ : VonKarman constant

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