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Turbulent impinging jet heat transfer enhancement due to intermittent pulsation

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ABSTRACT

A numerical study was carried out of heat transfer under a pulsating turbulent slot impinging jet. The jet velocity was varied in an intermittent (on–off) fashion. The effects of the time-mean jet Reynolds number, temperature difference between the jet flow and the impinging surface, nozzle-to-target distance as well as the frequency on heat and mass transfer were examined. The numerical results indicate significant heat transfer enhancement due to intermittent pulsation of the jet flow over a wide range of conditions for both cooling and heating cases. Simulations of the flow and temperature fields show that the instantaneous heat transfer rate on the target surface is highly dependent on the hydrodynamic and thermal boundary layer development with time.

1. Introduction

The efficiency of a convective heat transfer process is often limited by a very thin near-stagnant fluid layer held by viscosity along the target surface. Jet impingement provides an effective way to enhance heat and mass transfer due to its thin hydrodynamic and thermal boundary layers in the stagnation region. Also, the impinging jet configuration makes it possible to adjust the location of interest and the heat transfer rate by varying a number of flow and geometric parameters such as jet Reynolds number, nozzle shape, jet array assembly, nozzle-to-plate spacing etc. Hence the jet impingement technique is commonly employed for various cooling, heating and drying applications in many industrial applications ranging from thermal drying of continuous sheets of materials (e.g., textiles, films, papers, veneer, lumber, etc.) and foodstuffs, electronic component and gas turbine blade cooling, manufacture of printed wiring boards and metal sheets, printing processes, deicing or aircraft wings as well as tempering of glass and nonferrous metal sheets. Due to its importance, jet impingement heat transfer has been the topic of numerous investigations over the past three decades [1–4].

With rapid miniaturization of Micro-Electro-Mechanical Systems (MEMS), heat dissipation is becoming a major obstacle to further developments in micro-electronics industries. As flow pulsation is widely believed to increase heat transfer due to its features such as flow instability, higher turbulence levels, chaotic mixing and nonlinear dynamic response of the hydrodynamic and thermal boundary layers [7,14], pulsing impinging jets have been studied recently to evaluate their heat transfer enhancement potential [5–11]. However, current experimental and numerical results show conflicting results; both enhancement and no enhancement of heat and mass transfer have been reported [12]. Our previous numerical investigations on both laminar and turbulent impinging jets indicate that no noticeable enhancement by sinusoidal pulsation can be observed [9,12]. Unlike *sinusoidal* pulsation, *intermittent* pulsation referring to square-pulse waveform has been shown to display significant heat transfer enhancement [13–16]. Zumbrunnen and Aziz [13] firstly reported substantial heat transfer enhancements by the unsubmerged intermittent jet flow in 1993. Sheriff and Zumbrunnen [14] tested the cooling performance of a pulsing water impinging jet associated with both sinusoidal and square-pulse waveforms, and indicated that the reduction of local heat transfer by the sinusoidal pulsation decreases markedly from the stagnation point, while the square-wave pulsation can increase the time-averaged Nusselt number by up to 33%. Behera et al.'s numerical results [15] show that the sinusoidal pulsed impinging jet shows some enhancement over a steady impinging jet of the same Reynolds number only at

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Nomenclature

Latin Symbols

c	constant
c_p	specific heat (J kg^{-1} K^{-1})
f	frequency of pulsation (Hz)
g	gravity acceleration (m s^{-2})
H	nozzle-to-plate distance (m)
k	thermal conductivity (W m^{-1} K^{-1})
L	jet length (m)
n	integer ($\geq$0)
Nu	Nusselt number (Eq. (5))
P	static pressure (Pa)
Pr	Prandtl number, $\mu c_p/k$
q	heat flux density (W m^{-2})
Re	Reynolds number, $\rho u_{\text{jet}} w/\mu$
St	Strouhal number, fw/u_{jet}
t	time (s)
T	temperature (K)
ΔT	temperature difference (K)
u	velocity (m/s)
w	slot width (m)
x, y	Cartesian coordinates (m)

Greek symbols

ρ	density (kg m^{-3})
μ	dynamic viscosity (kg m^{-1} s^{-1})

Subscripts

$'$	fluctuation
avg	average with time
f	film temperature
i, j	indices in Einstein summation convention (Eq. (1)–(3))
jet	jet flow
on	open state
off	closed state
s	summation

large pulsation amplitude; the maximum enhancements are 5% and 10% in the stagnation zone and wall jet zone, respectively, However, they reported that the time-averaged Nusselt number increases up to 12% in the stagnation zone and 35% in the wall jet zone for the corresponding intermittent pulsed impinging jet. Recently, Zhou et al. conducted an experimental study of the heat transfer for jet impingement with sinusoidal and rectangular jet velocity pulsations on smooth as well as non-smooth surfaces [16]. For a smooth surface, the sinusoidal pulsation enhanced the average Nusselt number by up to 10% while the rectangular pulsation increased the Nusselt number by up to 40%. For a non-smooth surface, the advantage of rectangular pulsation was clearly seen as the decrease and significant enhancement were reported for sinusoidal and rectangular pulsations, respectively. However, the reason for the difference in behavior of smooth and rough surfaces is not yet clear.

Further studies are therefore needed to examine the effect of the intermittent pulsation on heat and mass transfer in impinging jets. In the current study, the heat and mass transfer characteristics of a two dimensional turbulent slot air impinging jet subjected to intermittent pulsation is investigated numerically for both cooling and heating cases. In drying of tissue paper, for example, the hot jet may be a couple of hundred degrees higher in temperature than the wet sheet being dried. Also cryogenic impinging jets are being considered for rapid freezing of fish and marine products. Therefore, the temperature difference between the jet and target can be varied to study the effect of temperature-dependent physical properties. In both cases, the heat transfer results are expected to be influenced by the temperature difference as well. The effect of jet Reynolds numbers, frequency and time ratio (on/off ratio) of the intermittent pulsation, temperature differences and geometrical configuration on heat transfer is discussed.

2. Numerical model

A two dimensional symmetric impinging jet configuration as shown in Fig. 1(a) was modeled. The air jet is vertically injected from the slot nozzle. The fluid was assumed to be incompressible and Newtonian fluid with temperature-dependent fluid properties. The intermittent pulsation with a uniform velocity profile (Fig. 2) was applied at the inlet of the impinging jet as described below:

$$u_{\text{jet}} = \begin{cases} u_{\text{on}}, & \text{for } nt_s \leq t \leq nt_s + t_{\text{on}} \\ u_{\text{off}}, & \text{for } nt_s + t_{\text{on}} \leq t \leq (n+1)t_s \end{cases} \tag{1}$$

where $u_{\text{off}} = 0$, $t_s = t_{\text{on}} + t_{\text{off}}$ and $n = 0, 1, 2\ldots$ in the current study. In following calculations, the time in the open (or on) state (t_{on}) and time in closed (or off) state (t_{off}) can be different from each other. Numerical simulation of the flow and thermal fields in a semi-confined turbulent impinging jet requires the solution of the continuity equation (2), the Navier–Stokes equations (3) and the energy equation (4) along with appropriate turbulence model equations:

$$\frac{\partial u_i}{\partial x_i} = 0 \tag{2}$$

$$\rho\frac{\partial u_i}{\partial t} + \rho\frac{\partial(u_i u_j)}{\partial x_j} = -\frac{\partial P}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j} + \frac{\partial u_j}{\partial x_i}\right) - \rho\overline{u_i' u_j'}\right] + \rho g_i \tag{3}$$

$$\rho c_p\left[\frac{\partial T}{\partial t} + \frac{\partial(u_i T)}{\partial x_i}\right] = \frac{\partial}{\partial x_i}\left(k\frac{\partial T}{\partial x_i} - \rho c_p\overline{u_i' T'}\right) \tag{4}$$

The computational domain and relevant boundary conditions are shown in Fig. 1(b). Due to geometric and flow symmetry, only

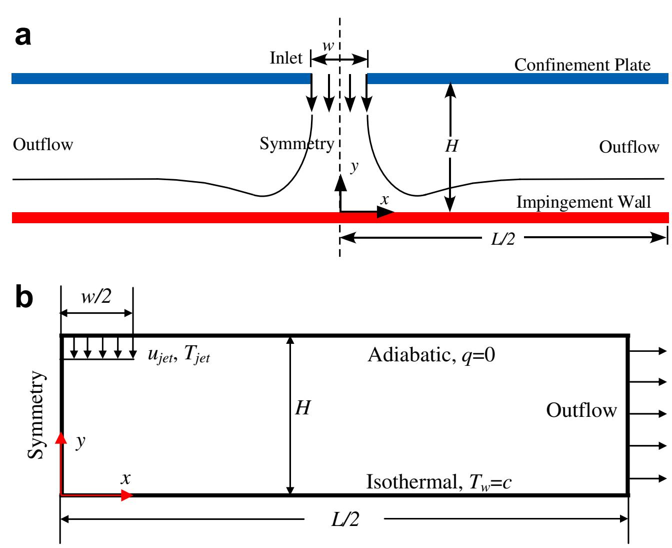

Fig. 1. (a) Physical model of the single pulsed slot impinging jet simulated, (b) computational domain and the relevant boundary conditions.

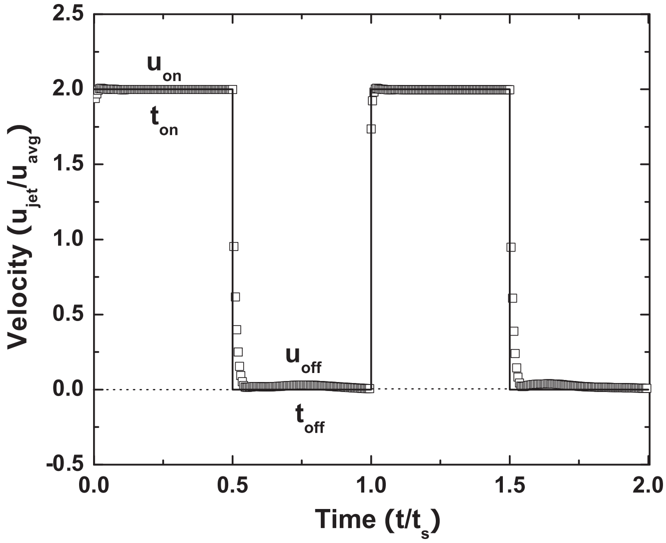

Fig. 2. Square waveform of the impinging jet velocity.

the flow field within the half domain was solved. The jet flow and impingement surface were specified as isothermal and the confinement wall was considered to be adiabatic, no-slip condition was imposed at the impingement wall. The uniform velocity, temperature, turbulent kinetic energy and energy dissipation rate profiles were assumed at the nozzle exit, the symmetry and fully developed outflow boundary conditions were taken at symmetry and outlet planes. The initial conditions ($t = 0$) throughout the computational domain can be described as: $u = v = 0$, $p = p_\infty$, $T = T_\infty$. The flow and thermal fields were computed with the finite volume computational fluid dynamics (CFD) code FLUENT 6.3. The Reynolds Stress Model (RSM) was chosen to model the turbulent flow in the square-wave pulsed impinging jet as it has been shown to be superior to k-ε and k-ω turbulence models in steady impinging jets [17,18]. Grid-independence of the final results was checked with different grid densities. Typically, a grid density of 150×50 provides satisfactory solution for the example shown here. A second upwind discretization scheme was used considering the stability of solution convergence and the SIMPLEC algorithm was employed for the pressure-velocity coupling. The time-step independence was also checked for each case and the time-step has been used as 1×10^{-4} s in the unsteady simulation. This numerical method has been validated by comparison with experimental results [5] and has shown good predictions of flow and thermal fields for pulsed turbulent impinging jets [12].

The local Nusselt number for an isothermal impingement surface was defined as:

$$Nu = \frac{q}{\Delta T}\frac{w}{k(T)} \tag{5}$$

The Nusselt number varies with time and position, therefore, the time-averaged local Nusselt number and time-averaged Nusselt Number can be, respectively, calculated by:

$$Nu_{avg}(x) = \int_0^t \frac{1}{\Delta t} Nu(x,t)dt \tag{6a}$$

$$Nu_{avg} = \int_0^x \frac{1}{\Delta x}\int_0^t \frac{1}{\Delta t} Nu(x,t)dtdx \tag{6b}$$

So far, most previous studies have involved small temperature differences between the jet fluid and the target surface. Under small temperature differences, all fluid properties can be regarded as temperature-independent and the differences between the local Nusselt numbers calculated according to different reference temperatures can be neglected. However, as noted before, in some industrial applications such as paper drying and turbine blade cooling, very high temperatures and large temperature differences do appear. Shi et al's study on steady turbulent slot impingement jet involving temperature differences ranging from 12 °C to 272 °C indicates that large temperature differences lead to significant differences in the impingement heat transfer coefficients [19]. In this study, large temperature differences are included in the modeling of pulsed turbulent impinging jets since some industrial applications of impinging jets involve large temperature differences between the jet fluid and the target surface e.g. in drying of tissue paper, cooling of glass or nonferrous sheets etc. Time-averaged Nusselt number, Nu_f, was calculated at the film temperature following the suggestion of Shi et al. [19].

3. Results and discussion

In this section, the effects of jet flow Reynolds number ranging from 1820 to 8200, Strouhal numbers from to 0.008 to 0.025, pulsation frequency from 33.33 to 66.67 Hz, time ratio (t_{on}/t_{off}) from 0.5 to 2, and temperature differences between the jet flow and that of impinging surface from 50 to 200 K as well the nozzle-to-plate spacing ranging from 3 to 8 on heat and mass transfer are analyzed in detail. In order to compare the heat transfer rates on the impingement wall, the injected mass flow rate was kept as the same for steady and pulsed jets. The cooling case will be described first where the jet flow of a lower temperature is impinged on a hotter target surface for cooling.

Fig. 3 shows the predicted time-averaged local Nusselt number distributions on the target surface at various mean jet Reynolds numbers for both of steady and intermittent pulsed impinging jets. It is clear that the time-averaged local Nusselt number increases as the mean jet Reynolds number increases for both of steady and pulsed cases. It can be seen from Fig. 3 that the time-averaged Nusselt number near the stagnation line ($x = 0$) in steady impinging jet is larger than that for the corresponding pulsed case. However,

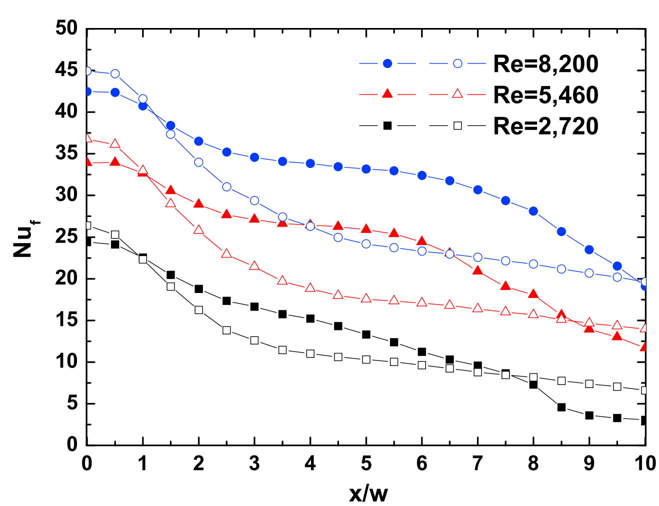

Fig. 3. Effect of mean jet Reynolds number on time-averaged local Nusselt number at $H/w = 5$, $f = 50$ Hz ($t_{on}/t_{off} = 1$), $\Delta T = 100$ K (solid symbols represent pulsed impinging jet and hollow symbols denote steady case, *sic passim*).

clear enhancement of heat transfer by intermittent pulsation can be found in the wall jet region. For example, the intermittent pulsation can increase the local Nusselt number up to 39% near $x/w = 3.5$ at $Re = 2720$, 47% near $x/w = 5$ at $Re = 5460$, and 39% near $x/w = 5.5$ at $Re = 8200$. Compared with the steady impinging jet of the same mean Reynolds number, the decrease of time-averaged local Nusselt number along distance from the stagnation line slows down in the wall jet region for the pulsed case. To explore the reason for this behavior of the intermittent pulsed impingement, the flow and temperature fields under a typical condition ($H/w = 5$, $Re = 2720$, $\Delta T = 100$ K) for steady and pulsed impinging jets are compared and portrayed in Figs. 4–6, respectively. Fig. 4(a) and (b) show the numerically predicted velocity profile and temperature distribution in a steady impinging jet respectively. Fig. 5 indicates that the variations of the velocity magnitude and velocity vector of a pulsed impinging jet at various times during a single cycle. Fig. 6 illustrates the changes of the temperature distribution of a pulsed impinging jet in one representative cycle. It can be seen from Fig. 4(a) that the vortex in the middle of the steady impinging jet is very weak while in the pulsed impinging jet, a very strong vortex can be found from the beginning of the cycle (Fig. 5(a)); it increases with fluid flow injection (Fig. 5(b)) and is present even when the pulsation is in the off mode (closed state Fig. 5(c) and (d)). The relatively strong vortex is believed to increase flow entrainment and mixing, and contribute to the predicted heat transfer enhancement in the pulsed impinging jet.

As Fig. 4 is examined in detail, it can be seen that the velocity in the wall jet region is low and the injected fluid flow with lower temperature can only reach the stagnation zone for the steady jet case. Examination of Fig. 5 shows that the velocity of the pulsed impinging jet is relatively high in the wall jet region, especially near $x/w = 3$–4. Fig. 6 shows that the injected cooler fluid is closer to the target surface in the wall jet region of the pulsed impinging jet than that in the steady impinging jet case, i.e., the hydrodynamic and thermal boundary layers are reduced in thickness by intermittent pulsation to values below those for a steady jet. Thus, the heat transfer performance of the pulsed impinging jet is better than that of the steady case in the wall region. Furthermore, the decrease of the time-averaged local Nusselt number slows down in this region.

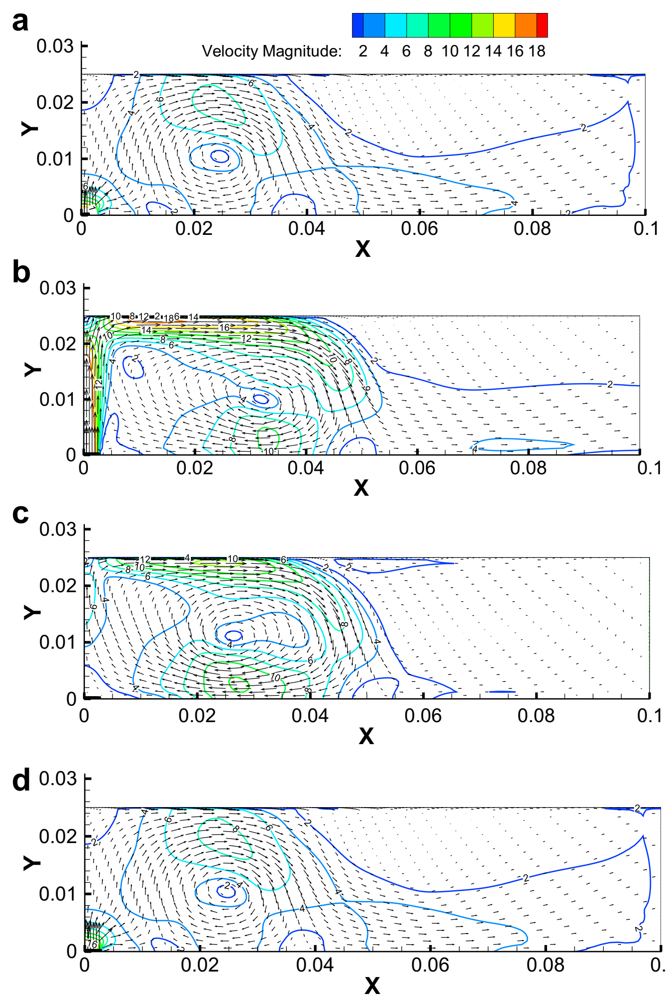

Fig. 5. Velocity magnitude and velocity vector plots in the pulsed impinging jet for $H/w = 5$, $Re = 2720$, $f = 50$ Hz ($t_{on}/t_{off} = 1$) and $\Delta T = 100$ K in one representative cycle: (a) $t = 0$, (b) $t = t_{on}$, (c) $t/t_s = 0.6$ and (d) $t = t_s$.

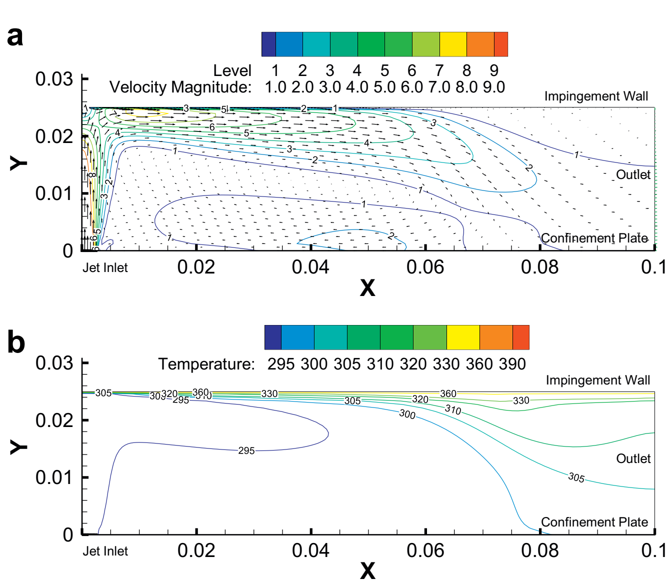

Fig. 4. Predicted velocity (a) and temperature (b) contours in the steady impinging jet at $H/w = 5$, $Re = 2720$, $\Delta T = 100$ K.

In many previous investigations, the time periods in the open (t_{on}) and closed (t_{off}) state were generally specified to be equal [13–16]. However, the current study indicates that the time ratio can provide additional flexibility as it can be adjusted to obtain better heat transfer performance of the intermittently pulsed impinging jet. Fig. 7 shows the effect of pulsation frequency on the heat transfer rate along the target surface. It is clear that the time-averaged local Nusselt number increases with frequency and Strouhal number. For high frequency (100 Hz), the time-averaged Nusselt number in the stagnation region of the pulsed impinging jet is even larger than that in the steady case. These findings are consistent with Zumbrunnen et al.'s experimental results which demonstrate that the heat transfer enhancements increase monotonically with frequency of intermittency and enhancements occur both beneath the jet and at locations several jet widths from the jet axis under high intermittent frequency [13].

The results for the effect of various temperature differences between the jet flow and impingement surface and nozzle-to-plate spacing on heat transfer rate in impinging jet are plotted in Figs. 8 and 9, respectively. The numerical calculations show that reduced temperature difference increases the time-averaged local Nusselt number in pulsed impinging jet, which is similar to that observed in the steady case (Fig. 8). It is clear that the pulsed impinging jet can provide better heat transfer performance in the wall jet region even

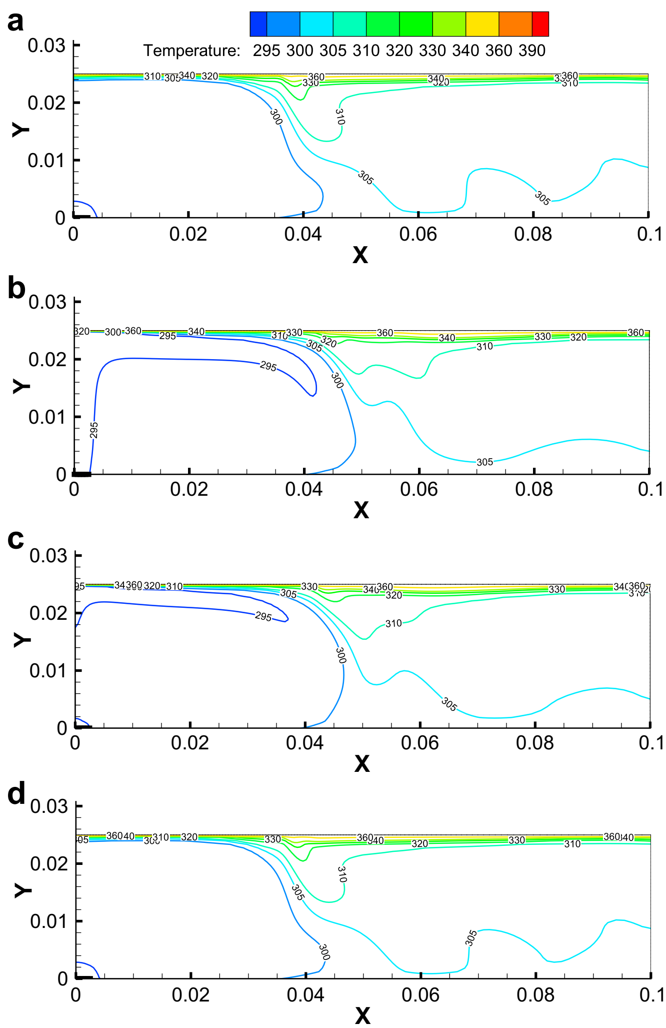

Fig. 6. Temperature distribution in pulsed impinging jet for $H/w = 5$, $Re = 2720$, $f = 50$ Hz ($t_{on}/t_{off} = 1$) and $\Delta T = 100$ K in one representative cycle: (a) $t = 0$, (b) $t = t_{on}$, (c) $t/t_s = 0.6$ and (d) $t = t_s$.

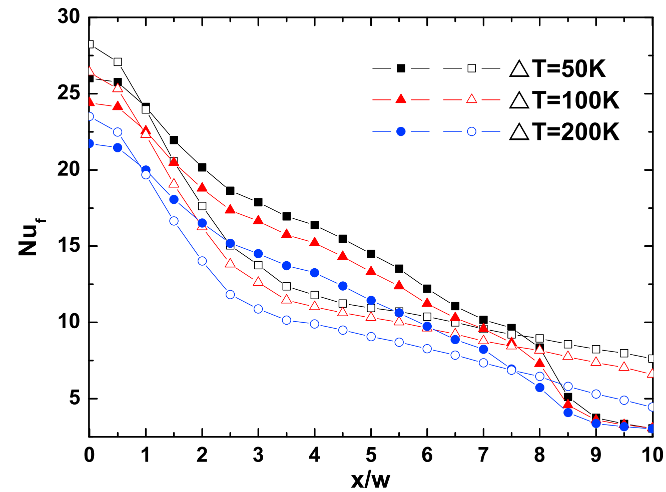

Fig. 8. Effect of temperature difference on time-averaged local Nusselt number distribution at $H/w = 5$, $Re = 2720$, $f = 50$ Hz ($t_{on}/t_{off} = 1$), $St = 0.025$.

for high temperature differences. It can be seen from Fig. 9 that the configuration geometry has an important effect on the heat and mass transfer performances in pulsed and steady impinging jets. The pulsed impinging jet performs better in the wall jet region at $H/w = 3$ and 5, while, heat transfer enhancement can be found in both of stagnation and wall jet regions at $H/w = 8$. At $H/w = 3$, an obvious drop of the time-averaged local Nusselt number distribution can be found in the wall jet region, which can be ascribed to the reflected flow by the confinement plate. That is the fluid can reach the confinement wall after impinging with the impingement wall and possibly induce higher turbulence levels and chaotic mixing, which can contribute to enhanced heat transfer rate.

In the applications for heating, hot jet flow is used to heat the impingement surface which is taken as T_∞ here. Numerical calculations performed for heating cases show that intermittent pulsation can significantly enhance the heat transfer rate at the target surface. For example, the averaged enhancement from the stagnation line ($x = 0$) to the distance ($x/w = 6$) is about 18.6% for heating and 19.1% for cooling. Under current conditions ($H/w = 5$, $Re = 2720$, $f = 50$ Hz ($t_{on}/t_{off} = 1$), $St = 0.025$, $\Delta T = 100$ K), the pulsed impinging jet only shows enhancement of heat transfer in the wall

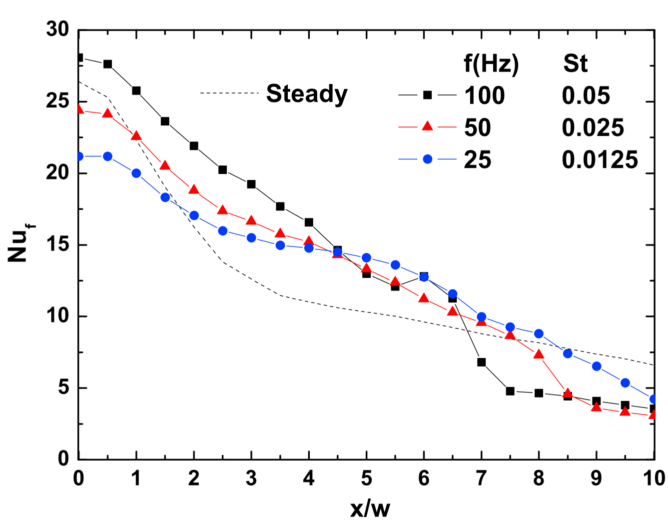

Fig. 7. Influence of frequency of square-wave pulsation on time-averaged local Nusselt number distribution ($H/w = 5$, $Re = 2720$, $t_{on}/t_{off} = 1$, $\Delta T = 100$ K).

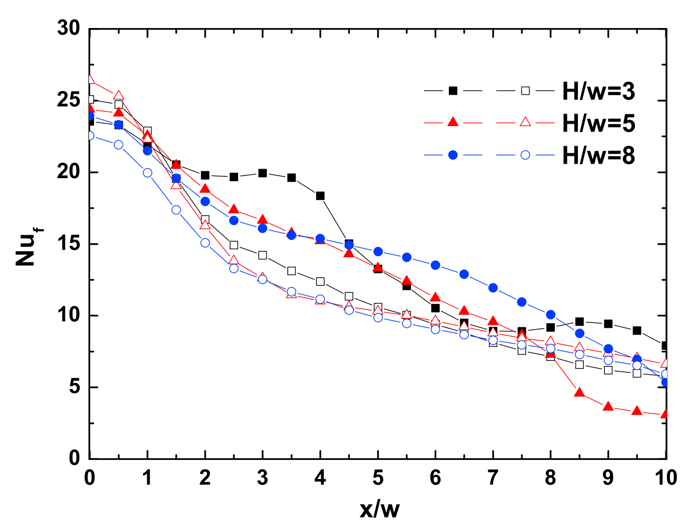

Fig. 9. Effect of nozzle-to-plate distance on time-averaged local Nusselt number at $\Delta T = 100$ K, $Re = 2720$, $f = 50$ Hz ($t_{on}/t_{off} = 1$), $St = 0.025$ and $H/w = 3,5,8$.

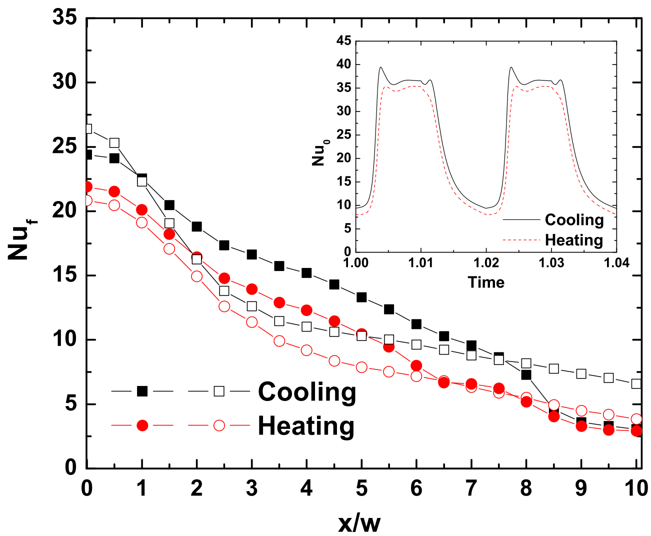

Fig. 10. Local Nusselt number at $H/w = 5$, $Re = 2720$, $f = 50$ Hz ($t_{on}/t_{off} = 1$), $St = 0.025$ and $\Delta T = 100$ K for cooling and heating cases (inserted graph shows local Nusselt number at the stagnation point with time).

jet region for cooling but exhibits enhanced heat transfer rate in both stagnation and wall jet regions for heating. The inserted graph in Fig. 10 shows the variation of local Nusselt number at stagnation line with time.

4. Conclusions

The numerical results presented show significant enhancement of heat transfer at the target surface by the intermittent pulsation in a turbulent impinging jet over a wide range of parameters for both of cooling and heating cases. Parametric studies show that increase of the mean jet Reynolds number and frequency of pulsation as well as reduced temperature difference enhance the time-averaged local Nusselt number. The on-to-off jet time ratio and nozzle-to-plate distance show significant effects on the heat transfer rates, which can be properly adjusted to achieve optimized performance. The observed effects of intermittent pulsation may be attributed to higher turbulence, larger vortices, increased flow entrainment and mixing promoted by flow instabilities as well as reduced instantaneous hydrodynamic and thermal boundary layers in the flow domain. One point should be noted that the mechanism of the intermittent pulsed impinging jets is very complicated; further investigations are needed to explore the key issues of the convective heat transfer enhancement due to the intermittent pulsation.

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