

Effect of Oscillating Jet Velocity on the Jet Impingement Cooling of an Isothermal Surface

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Abstract

Numerical investigation of the unsteady two-dimensional slot jet impingement cooling of a horizontal heat source is carried out in the present article. The jet velocity is assumed to be in the laminar flow regime and it has a periodic variation with the flow time. The solution is started with zero initial velocity components and constant initial temperature, which is same as the jet temperature. After few periods of oscillation the flow and heat transfer process become periodic. The performance of the jet impingement cooling is evaluated by calculation of friction coefficient and Nusselt number. Parametric study is carried out and the results are presented to show the effects of the periodic jet velocity on the heat and fluid flow. The results indicate that the average Nusselt number and the average friction coefficient are oscillating following the jet velocity oscillation with a small phase shift at small periods. The simulation results show that the combination of $Re=200$ with the period of the jet velocity between 1.5 sec and 2.0 sec and high amplitude (0.25 m/s to 0.3 m/s) gives average friction coefficient and Nusselt number higher than the respective steady-state values.

Keywords: Heat Transfer, Unsteady Convection, Jet Impingement, Periodic Oscillation, Numerical Study

1. Introduction

Jet Impinging is widely used for cooling, heating and drying in several industrial applications due to their high heat removal rates with relatively low pressure drop. In many industrial applications, such as in cooling of electronics surfaces, the jet outflow is confined between the heated surface and an opposing surface in which the jet orifice is located. Recently many researchers [1–7] have carried out numerical and experimental investigations of laminar impinging jet cooling with different fluids and under various boundary conditions.

The literature review reveals that the behavior of the two-dimensional laminar impinging jet is not well understood. Numerical results of Li *et al.* [8] indicate that there exist two different solutions in some range of geometric and flow parameters of the laminar jet impingement flow. The two steady flow patterns are obtained under identical boundary conditions but only with different initial flow fields. This indicates that the unsteady state analysis is important to have better understanding of the flow and heat transfer in jet impingement. Fi-

nite-difference approach was used by Chiriac and Ortega [9] in computing the steady and unsteady flow and heat transfer due to a confined two-dimensional slot jet impinging on an isothermal plate. The jet Reynolds number was varied from $Re=250$ to 750 for a Prandtl number of 0.7 and a fixed jet-to-plate spacing of $H=W=5$. They found that the flow becomes unsteady at a Reynolds number between 585 and 610. Chung *et al.* [10] have solved the unsteady compressible Navier–Stokes equations for impinging jet flow using a high-order finite difference method with non-reflecting boundary conditions. Their results show that the impingement heat transfer is very unsteady and the unsteadiness is caused by the primary vortices emanating from the jet nozzle.

Camci and Herr [11] have showed that it is possible to convert a stationary impinging cooling jet into a self-oscillating-impinging jet by adding two communication ports at the throat section. Their experimental results show that a self-oscillating turbulent impinging-jet configuration is extremely beneficial in enhancing the heat removal performance of a conventional (stationary) impinging jet. It is of great importance to investigate the

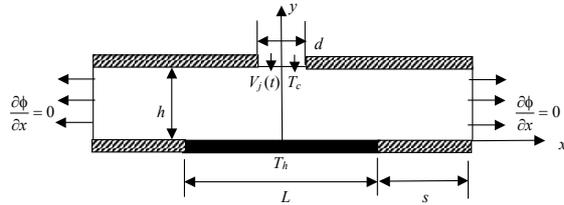


Figure 1. Schematic diagram of the physical model and coordinate system.

effect of periodic flow on the performance of the laminar jet impingement cooling process. Such investigation has been carried out numerically by Poh *et al* [12] to study the effect of flow pulsations on time-averaged Nusselt number under a laminar impinging jet. The target wall in this study is considered from the stagnation point until the exit. The whole target wall is subjected to a constant heat flux. The working fluid is water and the flow is assumed to be axi-symmetric semi-confined. They found that the combination of $Re = 300$, $f = 5$ Hz and $H/d = 9$ give the best heat transfer performance.

In applications such as electronics the components are usually considered as discrete heat sources and the cooling fluid is air. Therefore the objective of the present study is to investigate the periodic laminar jet impingement of air to cool a discrete and isothermal heat source.

2. Mathematical Model

A schematic diagram of impinging jet is shown in Figure 1. The jet exits through a slot of width d with distance h from the target-heated surface. All walls are adiabatic except the target plate where temperature is constant (T_h) and higher than the jet exit temperature (T_c).

The mathematical formulation of the present problem is based on the following assumptions:

- 1) the flow is two-dimensional, laminar and incompressible;
- 2) initial temperature and velocity profiles are assumed to be uniform across the jet width;
- 3) the thermo-physical properties of the fluid are constants and obtained at average temperature of the jet inlet and heater temperatures; and
- 4) the viscous heating is neglected in the energy conservation.

Based on the above assumptions, the governing equations for the unsteady heat and fluid flow are as follows:

Mass conservation equation

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \quad (1)$$

Momentum conservation equations

$$\frac{\partial u}{\partial t} + u \frac{\partial u}{\partial x} + v \frac{\partial u}{\partial y} = \frac{\partial}{\partial x} \left(\nu \frac{\partial u}{\partial x} \right) + \frac{\partial}{\partial y} \left(\nu \frac{\partial u}{\partial y} \right) - \frac{1}{\rho} \frac{\partial p}{\partial x} \quad (2)$$

$$\frac{\partial v}{\partial t} + u \frac{\partial v}{\partial x} + v \frac{\partial v}{\partial y} = \frac{\partial}{\partial x} \left(\nu \frac{\partial v}{\partial x} \right) + \frac{\partial}{\partial y} \left(\nu \frac{\partial v}{\partial y} \right) - \frac{1}{\rho} \frac{\partial p}{\partial y} \quad (3)$$

Energy conservation equation

$$\frac{\partial T}{\partial t} + u \frac{\partial T}{\partial x} + v \frac{\partial T}{\partial y} = \frac{\partial}{\partial x} \left(\alpha \frac{\partial T}{\partial x} \right) + \frac{\partial}{\partial y} \left(\alpha \frac{\partial T}{\partial y} \right) \quad (4)$$

where u and v are velocity components in x and y -directions respectively, T is temperature, p is pressure and t is time. ρ , ν and α are kinematic viscosity and thermal diffusivity of the fluid respectively.

Due to the symmetry around y -axis, only one-half of the flow field is considered for computational purpose. Therefore the initial and boundary conditions are:

Initial condition

$$u(x, y, 0) = v(x, y, 0) = 0 \text{ and } T(x, y, 0) = T_c \quad (5)$$

At $x = 0$ symmetry

$$\frac{\partial u}{\partial x} = \frac{\partial v}{\partial x} = \frac{\partial T}{\partial x} = 0 \quad (6a)$$

At $x = (L/2+s)$ exit

$$\frac{\partial u}{\partial x} = \frac{\partial v}{\partial x} = \frac{\partial T}{\partial x} = 0 \quad (6b)$$

At $y = 0$ lower wall

$$u = v = 0 \text{ and } T = T_h \text{ for } x \leq L/2 \text{ otherwise } \frac{\partial T}{\partial y} = 0 \quad (6c)$$

At $y = h$ upper boundary

$$u = 0, v = -V_j(t) \text{ and } T = T_c \text{ for } x \leq d/2 \text{ otherwise } u = v = \frac{\partial T}{\partial y} = 0 \quad (6d)$$

The present study investigates the effect of the jet velocity - $V_j(t)$ when it has a periodic variation with the flow time as:

$$V_j(t) = \bar{V} + \varepsilon \times \cos\left(\frac{2\pi}{\tau} t\right) \quad (7)$$

where \bar{V} is the average jet velocity, and ε and τ are the amplitude and period of the oscillation respectively.

The length of the lower adiabatic wall has an important influence on the accuracy of the results, where the exit boundary condition can be realistic. In the present study the length of the lower adiabatic wall is selected to be 3 times the heated surface ($L/2$) similar to that adopted by Rady [4].

3. Numerical Solution Procedure

The solution domain was meshed by divided it into

spacing quadrilateral cells. The cells were clusters near the symmetry axis where steep variations in velocity and temperature are expected.

FLUENT 6.3 is used as a tool for numerical solution of the governing equations based on finite-volume method. QUICK discretization scheme [13] is selected for convection-diffusion formulation for momentum and energy equations. The central differencing scheme is used for the diffusion terms. The discretized equations were solved following the SIMPLEC algorithm [14]. Relaxation factors are used to avoid divergence in the iteration. The typical relaxation factors were used as 0.7 for momentum equations, 0.3 for the pressure and 1.0 for the energy equation. For time integral the first order implicit scheme is used, which is unconditional stable.

The convergence criterion is based on the residual in the governing equations. The maximum residual in the energy was 10^{-7} and the residual of other variables were lower than 10^{-5} in the converged solution. In all the computational cases the global heat and mass balance are satisfied in the converged solution within $\pm 10^{-3}\%$.

Air is used as working fluid with constant physical properties. Most of the benchmark results are presented with constant Prandtl number, $Pr = 0.71$, for air. The average temperature between the cold incoming jet and the hot plate is selected to be 300K so that the Prandtl number is approximately 0.71. The plate temperature is fixed at 310K and the incoming jet temperature is maintained at 290K. The properties were found from the properties tables of air at an average temperature of 300K as: density $\rho = 1.1614 \text{ kg/m}^3$, specific heat $c_p = 1007 \text{ J/kgK}$, thermal conductivity $k = 0.0263 \text{ W/mK}$ and viscosity of $\mu = 1.846 \times 10^{-5} \text{ kg/ms}$.

4. Results and Discussions

The performance of the jet impingement cooling is evaluated based on the friction coefficient and Nusselt number, which are defined respectively as:

$$c_f = \frac{\tau_w}{\frac{1}{2}\rho\bar{V}^2} = \mu(\partial u/\partial y)_{y=0} / \frac{1}{2}\rho\bar{V}^2 \tag{8}$$

$$Nu = \frac{q_w d}{(T_h - T_c)k} = -d(\partial T/\partial y)_{y=0} / (T_h - T_c) \tag{9}$$

where τ_w is the wall shear stress and q_w is the wall heat flux. The average friction coefficient and the average Nusselt number at the heated plate are also calculated by integrating the local values over the length of the plate as follows:

$$\overline{c_f} = \frac{2}{L} \int_0^{L/2} c_f dx \tag{10}$$

$$\overline{Nu} = \frac{2}{L} \int_0^{L/2} Nu dx \tag{11}$$

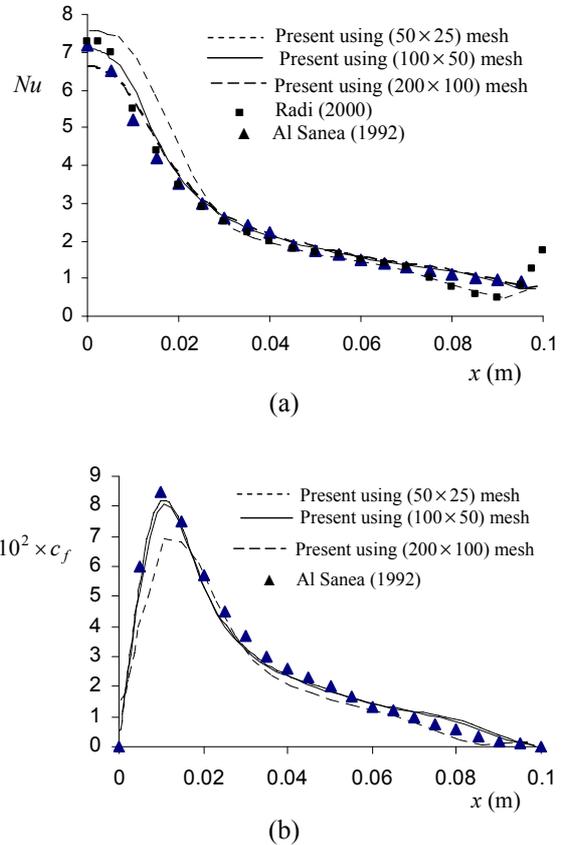


Figure 2. (a) Variation of Nusselt number along the heated plate. $Re = 200$, $h/d = 4$, $L/d = 20$ and $Pr = 0.71$; (b) Variation of friction coefficient along the heated plate. $Re = 200$, $h/d = 4$, $L/d = 20$ and $Pr = 0.71$.

Table 1. Values of \overline{Nu} with grid refinement compared with reference values.

Re	\overline{Nu}				
	Al-Sanea (1992)	Rady (2000)	Present results using different mesh sizes		
			(50 × 25)	(100 × 50)	(200 × 100)
100	1.596	1.46	1.4880	1.4743	1.4599
200	2.505	2.38	2.4332	2.4539	2.4473

Table 2. Values of average friction coefficient with grid refinement.

Re	V (m/s)	$\overline{c_f}$		
		Present results using different mesh sizes		
		(50 × 25)	(100 × 50)	(200 × 100)
100	0.159	0.030436	0.031934	0.031900
200	0.318	0.025352	0.028638	0.029346

The effect of mesh size on the accuracy of calculating friction coefficient and Nusselt number is studied for steady flow with constant jet velocity. The present results obtained using different mesh sizes are compared with the results of Al-Senea [2] and Rady [4]. The results presented in Figure 2(a) and Table 1 shows the comparison of local and average Nusselt number respectively. Figure 2(b) and Table 2 show the simulation results for local and average friction coefficient respectively. The present results show that the mesh with 100×50 quadrilateral cells in the x and y directions respectively gives results with acceptable accuracy. The mesh is designed so that the jet width, which is $d/2 = 0.005$ m is divided into 10 cells (control volumes). The heated surface $L/2 = 0.1$ m (which gives $L/d = 20$) is discretized into 50 divisions and the remaining adiabatic lower wall is divided into 50 divisions. The height $h = 0.04$ m (where $h/d = 4$) in the vertical direction is divided into 50 divisions. The results shown in Figures 2 and Tables 1 and 2 also show that halving or duplicating the mesh size have minor effects on the values of the Nusselt number and friction coefficient. Therefore the results obtained using mesh with 100×50 quadrilateral cells can be considered as grid independent results. Good agreements of the present results with those references cited in [2] and [4] are observed for two different values of the Reynolds number in the laminar regime. Where Re is the Reynolds number defined based on average jet velocity and jet width as: $Re = \rho \bar{V} d / \mu$. It is worth mentioning that the values of \bar{c}_f are not listed in the references [2] and [4].

In unsteady flows in general and especially periodic flows, the time step size has a great influence of the accuracy of the results. The time step size can be made to be a function of the frequency/period of the flow oscillation as implemented by Saeid [15,16]. In the present periodic flow problem, the time step size is selected a function of the period of the jet flow oscillation as $\Delta t = \tau/100$ sec.

To study the effect of the amplitude ε of the oscillation on the flow, the jet velocity is made to oscillate with time according to Equation (7) with fixed values of period $\tau = 10$ sec and $Re = 200$. It is important to note that the definition of Reynolds number in the present study is based on average jet velocity. To get $Re = 200$, the average jet velocity should be 0.318 m/s since the geometry of the problem and the air properties are assumed constants. Therefore the maximum amplitude of the oscillation is selected to be 0.3 m/s so that there will be always positive impinging velocity on the target surface.

The initial conditions in the unsteady simulation are defined in (5) which assume that the solution domain is filled with stagnant air at jet temperature. Then the jet starts to inflow and the target surface temperature in-

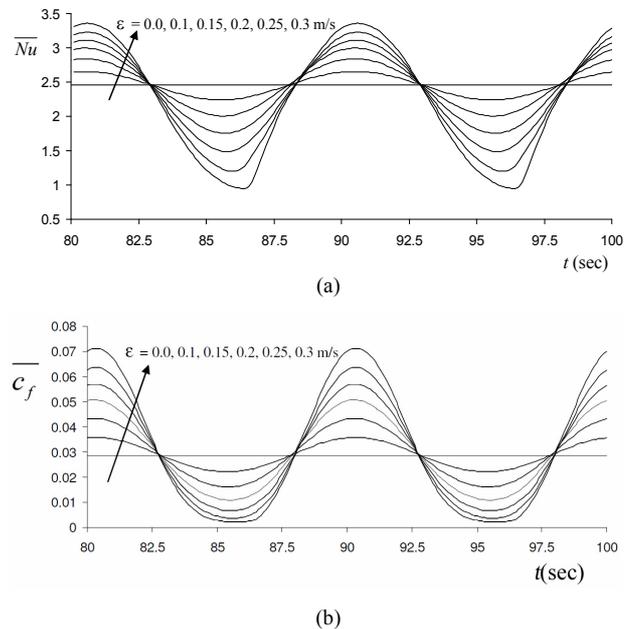


Figure 3. (a) Oscillation of \bar{Nu} with $\tau = 10$ sec and $Re = 200$; (b) Oscillation of \bar{c}_f with $\tau = 10$ sec and $Re = 200$.

creases suddenly from T_c to T_h . At this time the value of Nusselt number goes to very high value. Then, when the jet velocity oscillates the calculated values of average Nusselt number is found to oscillate accordingly. This oscillation becomes steady periodic oscillation after some periods of oscillation. The steady periodic oscillation is achieved when the amplitude and the average values of the average Nusselt number become constant for different periods.

The numerical results of oscillation of the average Nusselt number in the ninth and tenth periods with $\tau = 10$ sec and $Re = 200$ is shown in Figure 3(a). The corresponding oscillation of the average friction coefficient in the ninth and tenth periods is shown in Figure 3(b).

Both the Nusselt number and the friction coefficient are observed to oscillate in all the cases for different values of ε with a small phase change with the jet oscillation (which is cosine wave). For small values of the amplitude of the jet inflow oscillation ($\varepsilon = 0.1$ m/s to 0.2 m/s), the calculated average Nusselt number is oscillating in smooth sinusoidal oscillation as shown in Figure 3.

The effect of the period of the jet inflow velocity is studied and the results are shown in Figures 4a and 4b as \bar{Nu} against ωt and \bar{c}_f against ωt respectively, where ω is the frequency of the oscillation ($\omega = 2\pi/\tau$). Figure 4 shows clearly how the period of the jet velocity influences the periodic variation of \bar{Nu} and \bar{c}_f for

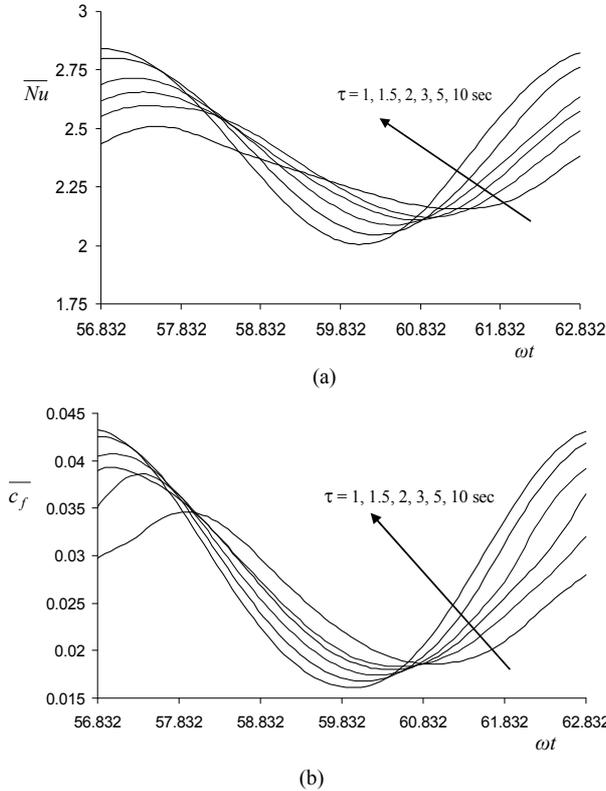


Figure 4. (a) Periodic oscillation of \overline{Nu} with $\epsilon = 0.1$ m/s, and $Re = 200$; (b) Periodic oscillation of $\overline{c_f}$ with $\epsilon = 0.1$ m/s, and $Re = 200$.

$Re = 200$ with forcing amplitude $\epsilon = 0.1$. At high values of τ there will be enough time for the momentum and heat transfer to follow the effect of the periodic variation of the jet velocity. Therefore the average Nusselt number and average friction coefficient are found to follow the jet velocity function (cosine-function) for high values of τ (5 and 10) as shown in Figure 4.

The amplitude of both the average Nusselt number and average friction coefficient oscillation is higher for larger periods of jet oscillation. Figure 4 shows also, as τ decreases, the peak values of average Nusselt number and average friction coefficient are delayed.

The results presented in Figures 3 and 4 show the oscillation of both the average Nusselt number and average friction coefficient according to the jet velocity oscillation; therefore it is important to introduce the cyclic average value of the space-averaged friction coefficient and Nusselt number defined respectively as:

$$\overline{c_f} = \frac{1}{\tau} \int_{t_0}^{t_0+\tau} c_f dt \tag{12}$$

$$\overline{Nu} = \frac{1}{\tau} \int_{t_0}^{t_0+\tau} Nu dt \tag{13}$$

where t_0 represents the time required to reach the steady periodic oscillation process (around 9 periods of oscillation). Figures 5(a) and 5(b) show respectively the variation of \overline{Nu} and $\overline{c_f}$ with ϵ for different values of the period of the jet oscillation and constant $Re = 200$.

For small values of ϵ (less than 0.15 m/s), the cyclic average value of the space-averaged Nusselt number (\overline{Nu}) is decreasing with the increase of either ϵ or τ as shown in Figure 5(a). Figure 5(b) shows that $\overline{c_f}$ also decreases with the increase of either ϵ or τ for small values of ϵ .

This means that the cooling process is deteriorated by using oscillating jet under these conditions. The results presented in Figure 5 show also the possibility of cooling enhancement when the period of the jet velocity between 1.5 sec and 2.0 sec and high amplitude (0.25 m/s to 0.3 m/s) with $Re = 200$.

At these conditions the cyclic average value of both the space-averaged friction coefficient and Nusselt number are found to be higher than the steady-state value (when $\epsilon = 0$) as shown in Figure 5.

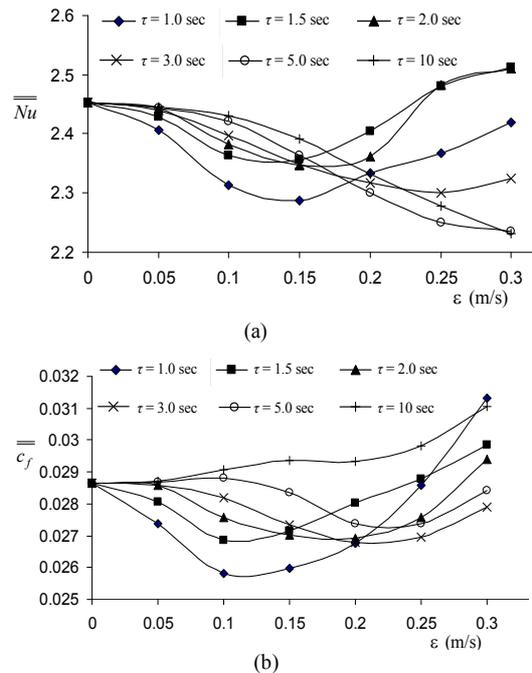


Figure 5. (a) Variation of \overline{Nu} with ϵ at $Re = 200$; (b) Variation of $\overline{c_f}$ with ϵ at $Re = 200$.

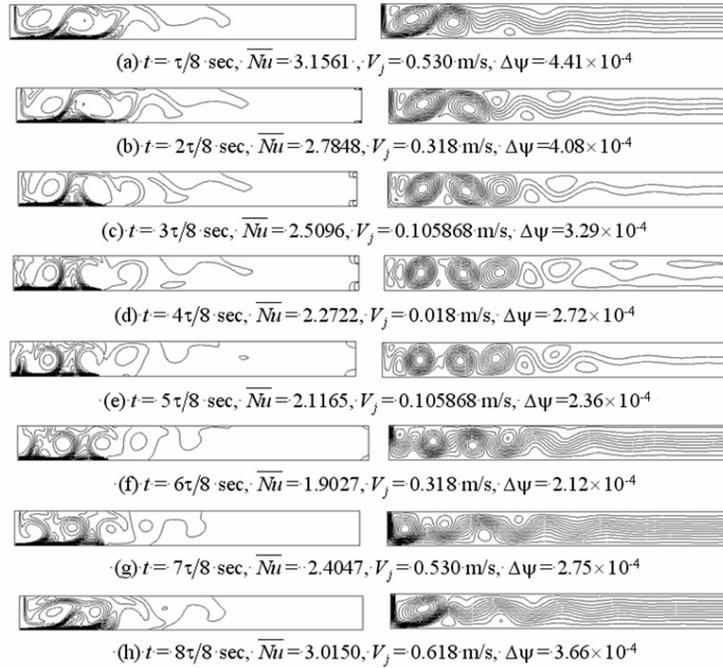


Figure 6. Isotherms, $\Delta T = 1\text{K}$ (left) and streamlines (right) for a cycle of oscillation with $\varepsilon = 0.3$ m/s, $\tau = 2$ sec, and $Re = 200$.

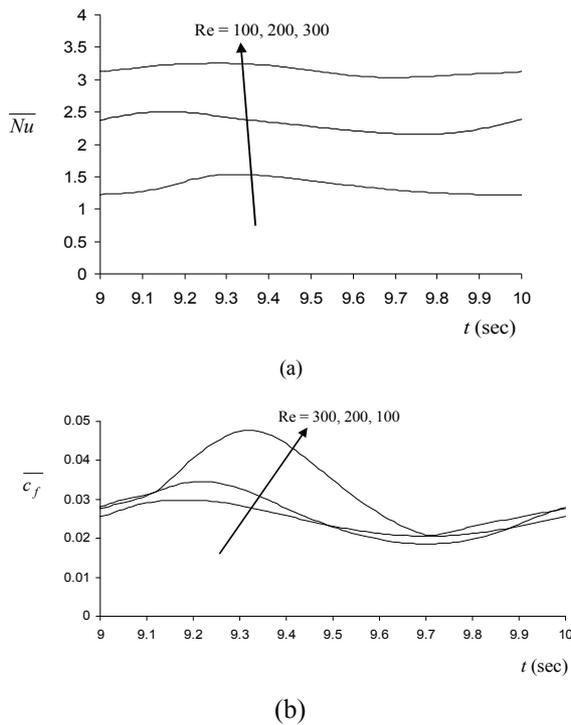


Figure 7. (a) Periodic oscillation of \overline{Nu} with $\varepsilon = 0.1$ m/s and $\tau = 1$ sec; (b) Periodic oscillation of $\overline{c_f}$ with $\varepsilon = 0.1$ m/s and $\tau = 1$ sec.

From the results presented in Figure 5 it can be seen that the increase of \overline{Nu} is about 2.3% while the increase in

$\overline{c_f}$ is 2.6% when the period of the jet velocity is 2.0 sec and amplitude of 0.3 m/s with $Re = 200$.

In order to have better understanding, the period of the last cycle is divided into eight time steps. At each time step the isotherms and streamlines are shown in Figure 6 for the periodic oscillation with $\varepsilon = 0.3$ m/s, $\tau = 2$ sec and $Re = 200$.

The isotherms show some high temperature points on the heated target wall. These hot spots are moving along the heated surface according to the jet velocity oscillation. Obviously when the jet velocity is small near the minimum at $t = 4\tau/8$ sec ($V_j = 0.018$ m/s) the temperature near the target surface is high. Figure 6 show that the oscillation of the jet velocity leads to wash away the heated spots after they appear above the heated surface with some delay. The average Nusselt number value at $t = \tau/8$ sec ($V_j = 0.530$ m/s) is higher than that at maximum velocity at $t = 8\tau/8$ sec, ($V_j = 0.618$ m/s).

Finally the effect of the Reynolds number on the periodic jet impingement cooling process is studied and the results are depicted on Figure 7. The range of the Reynolds number is selected to be in the laminar regime. Obviously increasing the Reynolds number by increasing the jet velocity leads to the increase in the average Nusselt number and reduce the friction coefficient as shown in Figure 7(a) and (b) respectively. It is observed that the oscillation of both the average Nusselt number and the av-

erage friction coefficient at different values of Re have small phase shift in the steady periodic oscillation as shown in Figure 7.

5. Conclusions

In the present study the periodic laminar jet impingement cooling of a horizontal surface is considered for numerical investigation. The periodic jet impingement cooling is generated when there is periodic oscillation of the jet inflow velocity. It has been shown that the Nusselt number oscillates as a result of oscillating jet inflow velocity. The results are presented to show the effects of the amplitude and the period of the jet velocity on the Nusselt number and friction coefficient in the steady periodic state. The results indicate that both the average friction coefficient and Nusselt number are oscillating following the jet velocity oscillation with a small phase change. The periodic average friction coefficient and the Nusselt number are found to follow the jet velocity function for high values of period τ . This is due to the fact that there is enough time for the momentum and heat transfer to follow the effect of the periodic variation of the jet velocity. The simulation results show that it is possible to enhance the cooling process for some combination of the Reynolds number with period and amplitude of the jet velocity. The combination of $Re = 200$ with the period of the jet velocity between 1.5 sec and 2.0 sec and high amplitude (0.25 m/s to 0.3 m/s) gives average friction coefficient and Nusselt number higher than the respective steady-state values.

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