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Single- and two-phase water jet impingement heat transfer on a hot moving surface

M. R. Mohaghegh[1] · Asghar B. Rahimi[1]

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Abstract
In this study, the laminar flow and heat transfer of water jet impingement on a hot moving plate is investigated. A similarity solution is applied to momentum and energy equations formulating the single-phase forced convection in order to determine the flow velocity and heat transfer. The heat flux in flow boiling regime is predicted by a superposition approach which is based on the combination of the single-phase and nucleate pool boiling components. The effects of surface motion and arbitrary surface temperature distribution on important forced convection and nucleate boiling heat transfer parameters for both stationary and moving plates are examined in the stagnation line and its nearby region. The results show that surface motion does not affect the rate of heat transfer in stagnation region when surface temperature is constant, while this motion is found to decrease heat transfer for a non-uniform surface temperature distribution state. However, it is observed that in fully developed nucleate boiling regime, the parameters including the surface velocity, the surface temperature gradient and the local distance from the stagnation line have negligible effect on the rate of heat transfer from the surface.

Keywords Jet impingement · Stagnation region · Moving surface · Surface temperature gradient · Similarity solution · Nucleate boiling

List of symbols

c_p	Specific heat (J kg^{-1} k^{-1})
C	Free-stream velocity gradient in stagnation region
$\overline{C}$	Dimensionless velocity gradient
f	Dimensionless function related to flow velocity
g	Gravity acceleration (m s^{-2})
h	Heat transfer coefficient (W m^{-2} K^{-1})
h_{fg}	Latent heat of vaporization (J kg^{-1})
I	Dimensionless function related to flow velocity due to plate motion
k	Thermal conductivity (W m^{-1} K^{-1})
Nu_w	Local Nusselt number $= hw_j/k$
Nu^*	Ratio of local Nusselt number for moving plate to local Nusselt number for stationary plate
p	Pressure (N m^{-2})
Pr	Prandtl number
q''	Heat flux (W m^{-2})
Re_w	Jet Reynolds number $= V_j w_j/\nu$
S	Suppression factor defined in Eq. (23)
T	Temperature (°C or K)
T_{s_0}	Temperature of impingement surface at stagnation line
ΔT	Temperature difference (°C or K)
u	Velocity component in x direction (m s^{-1})
U_∞	Free-stream velocity (m s^{-1})
$\overline{U}_\infty$	Dimensionless free-stream velocity $= U_\infty/V_j$
v	Velocity component in y direction (m s^{-1})
V_j	Jet velocity (m s^{-1})
V_p	Surface velocity (m s^{-1})
$\overline{V}_p$	Dimensionless surface velocity $= V_p/V_j$
w_j	Jet width (m)
x	Horizontal distance from stagnation line
$\overline{X}$	Dimensionless horizontal distance $= x/w_j$
y	Vertical distance above impingement surface

Greek symbols

α	Molecular thermal diffusivity (m^2 s^{-1})
β	Surface temperature gradient
ν	Molecular kinematic diffusivity (m^2 s^{-1})
μ	Molecular dynamic diffusivity (kg m^{-1} s^{-1})
ρ	Density
λ	Parameter defined in Eq. (22)

✉ Asghar B. Rahimi
rahimiab@yahoo.com; rahimiab@um.ac.ir

[1] Faculty of Engineering, Ferdowsi University of Mashhad, P.O. Box 91775-1111, Mashhad, Iran

σ Surface tension (N m^{-1})
η Dimensionless distance from surface
θ Dimensionless temperature
θ_s Dimensionless surface temperature

Subscripts
f Film temperature
FDB Fully developed boiling
j Jet related value
nb Nucleate boiling
onb Onset of nucleate boiling
s Surface (wall, plate)
sp Single phase
sub Subcooled
sup Superheat
w Related to the jet width
∞ Free stream-related value

Superscripts
$'$ First derivative
$''$ Second derivative

Introduction

Cooling of hot surfaces by impinging jets due to extracting high heat flux is widely used in many industrial and engineering applications such as cooling of electronic chips and microelectronic circuits, hot rolling steel strip and nuclear power plants. Many researchers studied single-phase liquid impinging jets on a stationary plate experimentally [1–3] and numerically [4–9] and the two-phase state experimentally [10–16]. A comprehensive review of the jet impingement boiling has been published by Wolf et al. [17]. They compiled various available correlations of boiling curve from steady-state measurements.

Most studies have investigated jet impingement flow on stationary surfaces, while relatively fewer researches have studied the effect of surface motion over the heat transfer field. In some engineering applications of impinging jets, the surface being cooled by jet moves and its velocity is comparable to the jet velocity or exceeds it. For instance, a plate cooled by a planar jet in a manufacturing process may be in motion such as continuous casting and hot rolling process. Zumbrunnen [18] studied the stagnation region of a laminar, planar jet on a moving surface in single-phase force convection heat transfer. His result showed that convective heat transfer rate was unaffected by surface velocity when the surface temperature is spatially constant. However, with decreasing surface temperature in the direction of surface motion, Nusselt numbers became smaller. Chen et al. [19] developed a numerical model to determine convective heat transfer of an array of submerged planar jets impinging on a uniform heat flux or constant temperature moving surface. They found that neglecting surface motion effects could lead to significant overestimates of heat transfer rate.

Chattopadhyay et al. [20] investigated the effect of plate motion on the turbulent flow and heat transfer in axial slot jets. Then, Chattopadhyay et al. [21] studied their prior problem for a knife-jet with an exit angle of 60 degrees. They found that with increasing velocity of the impinging surface, the total heat transfer reduces. Ja'fari and Rahimi [22] studied effect of moving plate with time-dependent axial velocity and uniform transpiration on an axisymmetric stagnation-point flow and heat transfer. This study showed that the heat transfer coefficient increases with increasing transpiration rate and Prandtl number. Zumbrunnen et al. [23] developed an experimental method and apparatus to measure local heat transfer distributions on moving and stationary plates in different heat transfer regimes, such as single-phase forced convection, nucleate boiling and film boiling. Their experimental results showed that in single-phase regime, heat transfer rate near the stagnation point was not considerably affected by surface motion. In nucleate boiling, heat transfer coefficients depend strongly on the surface temperature and location of maximum heat transfer occurs directly beneath the jet and is not affected significantly by the plate motion. Experimental investigations were carried out by Gradeck et al. [24] for quenching of a hot rotating cylinder by a planar water jet. The initial temperature was about 500–600 °C. They found that for a moving surface, the maximum of heat transfer (for a given regime) is moving during the cooling time from downstream (film boiling regime) to upstream (forced convection).

A thorough review of the literature reveals lots of experimental and numerical works on single-phase liquid jet impinged on a stationary plate. Relatively fewer numerical studies exist on jet impingement problem along with effect of surface motion. Moreover, to our knowledge, no numerical studies in the literature exist on the effect of surface motion and surface temperature gradient on steady laminar planar jets, neither on single phase nor on flow boiling. However, a numerical mechanistic modeling of jet impingement boiling on a stationary plate with constant temperature was reported by the present authors [25]. The investigation of surface motion with an arbitrary surface temperature distribution on forced convection and nucleate boiling heat transfer parameters is undertaken in the present work. For this purpose, effect of surface motion and surface temperature gradient is considered using Navier–Stokes and energy equations and similarity solution is obtained for single-phase state. For nucleate boiling regime, a numerical mechanistic model based on superposition of the forced convection and pool boiling is applied. The mentioned effects are investigated in heat transfer characteristics of both single- and two-phase states such as Prandtl number, dimensionless number of heat flux, Nusselt number, the temperature of onset of nucleate boiling and boiling curve, and also the distribution of these characteristics is derived in different locations of stagnation region and the obtained result is discussed.

Model of the problem

The impinging jets are categorized in five different configurations, as: free-surface jet, plunging jet, submerged jet, confined jet and wall jet [17]. This study focuses on the free-surface jets. The detail of an impinging jet configuration and two-dimensional flow profile in the impingement region of a planar jet with a velocity v_j and temperature T_∞ on a stationary flat plate is illustrated in Fig. 1.

As the water jet impinges on the heated surface, the flow is assumed to divert symmetrically around the stagnation line and extends to the surface in a parallel manner. The velocity profiles in viscous boundary layer and free-stream (inviscid) region are depicted in stagnation and acceleration regions. The pressure is maximum at the stagnation line ($x = 0$) and decreases away from it. The pressure on the surface at the stagnation line can be obtained by the Bernoulli's equation:

$$P = P_0 + \frac{1}{2} V_j^2 \quad (1)$$

The majority of jet impingement studies investigate the area of the stagnation region since the heat transfer enhancement is significant in this area and continuously decreases in the direction away from the stagnation region [17, 26–32]. The motion of the plate with a typical velocity of V_p has significant effect on the velocity field and surface tension. This study considers effect of the surface velocity in motion parallel to the flow direction ($x \geq 0$). The distribution of the dimensionless flow velocity components $\bar{U}$ in boundary layer width (η) for three different locations in stagnation region is presented in Fig. 2 for a stationary plate ($\overline{V}_p = 0$) and a moving plate ($\overline{V}_p = 1$). As it can be seen, the velocity profiles for the moving plate differ remarkably from those for the stationary plate.

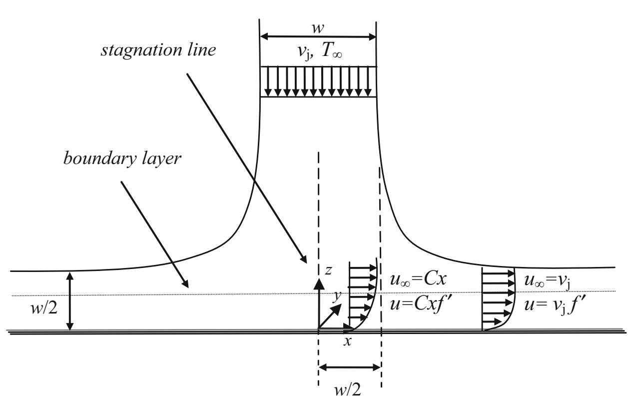

Fig. 1 Flow velocity profile of an impinging planer jet on a stationary plate in stagnation region and away from it

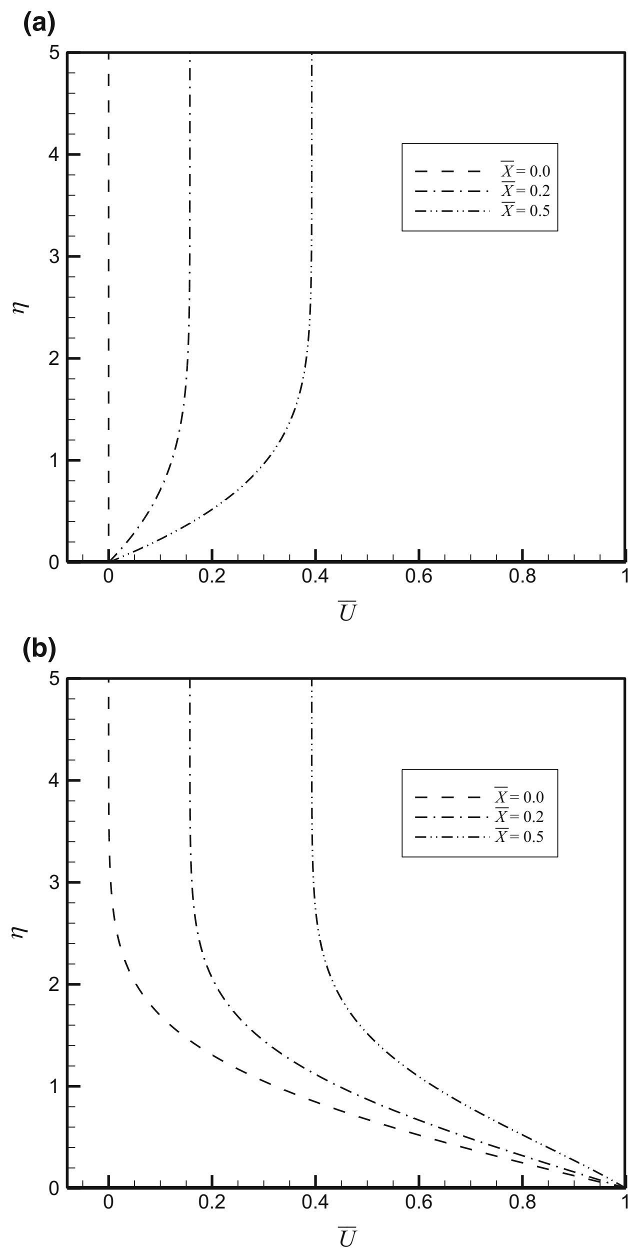

Fig. 2 Dimensionless profile of flow velocity component u in the vicinity of a **a** stationary and **b** moving plate

Mathematical formulation

Single-phase forced convection heat transfer

The forced convective heat flux can be calculated by Newton's law of cooling as:

$$q''_{\rm fc} = h(T_{\rm s} - T_{\rm f}) = h(\Delta T_{\rm f}) \tag{2}$$

To calculate heat flux, it is needed to know the correlations of the heat transfer coefficient h. This coefficient can be obtained by solving the boundary layer equations. Consider a steady, two-dimensional, incompressible, laminar boundary layer flow and heat transfer of a viscous fluid in the neighborhood of a stagnation point on a flat plate located in the plane $y = 0$. The problem is formulated in Cartesian coordinates (x, y) with corresponding velocity components (u, v) under the following assumptions:

- Liquid properties in boundary layer are estimated in film temperature:

$$T_{\rm f} = \frac{1}{2}(T_{\rm s} + T_\infty) \tag{3}$$

- The Bernoulli's equation is employed in the potential region:

$$-\frac{1}{\rho}\frac{\partial p}{\partial x} = U_\infty \frac{{\rm d}U_\infty}{{\rm d}x} \tag{4}$$

- Boundary layer flow is assumed laminar. For stagnation flow, critical Reynolds number ($Re_{\rm w,cr} = \frac{v_{\rm j} w}{\nu}$) was reported about 4000 by Lienhard [33]. The range of Reynolds number in the present study (with $v_{\rm j} \le 1.25$ and $w_{\rm j} = 1$ mm) is less than 4000.

The simplified form of the two-dimensional laminar boundary layer equations is presented as following:

Continuity equation:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{5}$$

Momentum equation:

$$u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} = U\frac{{\rm d}U}{{\rm d}x} + \nu\frac{\partial^2 u}{\partial y^2} \tag{6}$$

Energy equation:

$$u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} = \alpha\frac{\partial^2 T}{\partial y^2} \tag{7}$$

The free-stream velocity component of the classical potential flow solution is $U = Cx$. The C parameter introduces the velocity gradient which is expressed in terms of the jet velocity and the jet width as $C = \bar{C}\frac{v_{\rm j}}{w}$, [34], where the value of $\bar{C} = \frac{\pi}{4}$ is valid for $0 \le x \le 1$, [1].

Similarity solution

To convert partial differential Eqs. (6, 7) into a set of ordinary differential equations, the following dimensionless similarity variables are introduced:

$$\eta = \sqrt{\frac{C}{\nu}}y,\ u = Cxf'(\eta) + V_{\rm p}I(\eta),\ v = -Cf(\eta),\ \theta(\eta) = \frac{T - T_\infty}{T_{\rm s} - T_\infty} \tag{8}$$

where $V_{\rm p}$ is the velocity of plate motion and the function $I(\eta)$ reflects effects of this motion on the flow velocity in the boundary layer. The no-slip condition because of viscous effects causes that the horizontal flow velocity u near the surface at each point (x) to have a velocity of the surface speed $V_{\rm p}$. So, the function $I(\eta)$ should be the largest (unit) at the plate surface ($y = 0$) and become smaller away from the surface, as expected from Fig. 2. For a stationary plate $V_{\rm p} = 0$, the velocity component u reduces to the form $Cxf'(\eta)$ which is used for the stationary plate [25].

To generalize the equation, the dimensionless variables of the problem are defined as the following:

$$\overline{X} = \frac{x}{w_{\rm j}},\ \overline{U} = \frac{u}{V_{\rm j}},\ \overline{V}_{\rm p} = \frac{V_{\rm p}}{V_{\rm j}}, \tag{9}$$

and

$$\theta_{\rm s} = \frac{T_{\rm s} - T_\infty}{T_{\rm s_0} - T_\infty} \tag{10}$$

where $T_{\rm s_0}$ is surface temperature at the stagnation line. A linear local variation of the surface temperature near the stagnation line is considered as follows:

$$\theta_{\rm s} = 1 + \beta\overline{X} \tag{11}$$

where β is the dimensionless surface temperature gradient in the vicinity of the stagnation line. Note, that for a surface with constant temperature ($T_{\rm s} = T_{\rm s_0}$ at each local distance), according to Eq. (10), $\theta_{\rm s} = 1$ and therefore $\beta = 0$. Substituting the transformations (6) into the momentum and energy Eqs. (6) and (7) and applying Eqs. (9–11) yields the following nonlinear ordinary differential equations:

$$f''' + ff'' - f'^2 + 1 = 0 \tag{12}$$

$$I'' + fI' - f'I = 0 \tag{13}$$

$$\theta'' + pr\left[f\theta' - \left(\overline{X}f' + \frac{\overline{V}_{\rm p}I}{\overline{C}}\right)\frac{\beta}{w_{\rm j}}\frac{\theta}{\theta_{\rm s}}\right] \tag{14}$$

The boundary conditions in Eqs. (12)–(14) are as:

$$\eta = 0 : \begin{cases} u = 0,\ v = 0,\ T = T_s \\ f' = f = I = 0,\ \theta = 1 \end{cases} \tag{15}$$

$$\eta \to \infty : \begin{cases} u = U_\infty,\ T = T_\infty \\ f' = 1,\ I = 0,\ \theta = 0 \end{cases} \tag{16}$$

The heat transfer coefficient h can be calculated by the following relation:

$$h = -\frac{K\frac{\partial T}{\partial y}\Big|_{y=0}}{\Delta T_f} = -\rho c_p Pr^{-1}\sqrt{Cv}\,\theta'(0) \tag{17}$$

Equations (12–14) with the boundary conditions (15) and (16) are a set of coupled highly nonlinear ordinary differential equations with boundary values. One of the most convenient and efficient method to solve boundary value problems of a set of nonlinear ODEs is the fourth-order Runge–Kutta numerical method. The boundary value problems such as mentioned equations have an unknown parameter in upper boundary condition (η_∞). So, the upper boundary conditions $f'(\infty)$, $I(\infty)$ and $\theta(\infty)$ may be substituted by the initial boundary conditions $f''(0)$, $I'(0)$ and $\theta'(0)$, to convert boundary value problem to an initial value problem. For this purpose, the shooting technique is applied along with the fourth-order Runge–Kutta method with initial guesses for values of $f''(0)$, $I'(0)$ and $\theta'(0)$ and an iterative solution procedure till satisfaction of the upper (initial) boundary conditions $f'(\infty)$, $I(\infty)$ and $\theta(\infty)$.

Nucleate pool boiling heat transfer

In different surface temperatures, various heat transfer regimes are observed. Within the surface temperature below the temperature required for vapor bubble nucleation, i.e., $T_s \prec T_{sonb}$, single-phase cooling regime occurs. When the surface temperature is well above the saturation temperature of the liquid, isolated vapor bubbles begin to nucleate and grow on the surface. This temperature is shown as T_{onb}(onset of nucleate boiling) in Fig. 3. The results of investigations into rates of heat transfer in boiling are usually plotted on a graph of surface heat flux (q'') against wall superheat (ΔT_{sat}) which is called the boiling curve. The trend of the pool and flow boiling curves in forced convection (single phase) and nucleate (partial and fully developed) boiling (two-phase) regions which are under study in this paper is shown schematically in Fig. 3, [25].

As it is seen, in a fixed wall superheat (ΔT_{sat}), the heat transfer rate of flow boiling is more than pool boiling because of the forced convection effects. According to the figure, total heat flux can be expressed as a combination of forced convection and pure pool boiling heat flux. This idea of additive contributions was first introduced by Rohsenow [35]. Then, Bergles and Rohsenow [36] developed this idea by considering this point that just prior to incipient boiling, heat flux still can be expressed by forced convection heat flux. The present authors in their last study [25] presented the following equation for flow boiling as:

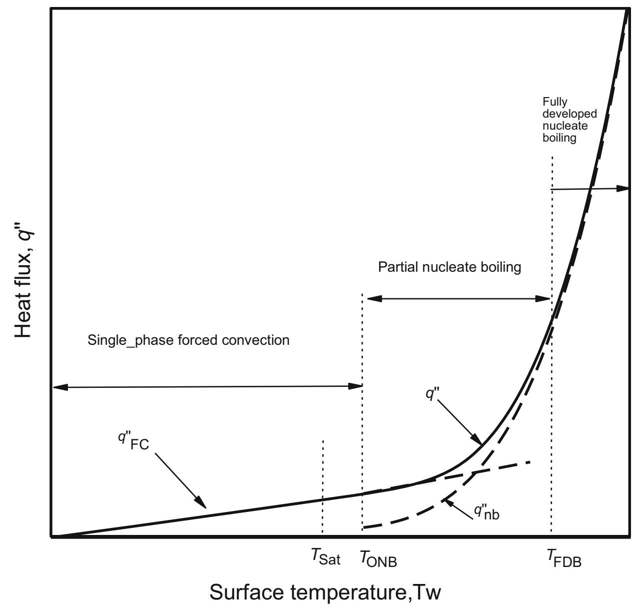

Fig. 3 Schematic of the boiling curves: Solid line denotes the flow boiling; dashed line denotes the pool boiling curve

$$q'' = \left(q_{fc}''^2 + \left(Sq''_{nb}\right)^2\right)^{\frac{1}{2}} \tag{18}$$

The nucleate boiling heat flux q''_{nb} is calculated from the following equation:

$$q''_{nb} = h_{nb}(T_s - T_{sat}) = h_{nb}\Delta T_{sat} \tag{19}$$

where h_{nb} is computed by Gorenflo's correlation [25, 37].

The suppression factor, S, in Eq. (18) shows the effects of forced convection on pool boiling in flow boiling regime and is defined as the following [25]:

$$S = 1 - \left(\frac{\Delta T_{sat_{onb}}}{\Delta T_{sat}}\right)^3, \tag{20}$$

where

$$(\Delta T_{sat})_{onb} = \frac{1 + (1 + 4\lambda\Delta T_{sub})^{\frac{1}{2}}}{2\lambda} \tag{21}$$

And,

$$\lambda = \frac{k_f h_{fg}}{8\sigma v_g T_{sat} h} \tag{22}$$

Using Eqs. (21) and (22) and substituting into Eq. (20) yields the following expression for suppression factor as an explicit function of thermo-physical properties of liquid jet and surface temperature as:

$$S = 1 - \left(\frac{1 + (1 + 4\lambda\Delta T_{\mathrm{sub}})^{\frac{1}{2}}}{2\lambda\Delta T_{\mathrm{sat}}}\right)^{3} \tag{23}$$

As it can be seen, S considers effects of jet velocity and liquid temperature (subcooling) with considering forced convection heat transfer coefficient h and also thermo-physical property of the liquid jet, properly under the physical conception of the problem.

Results and discussion

Single phase

Results of the numerical integration of Eqs. (12–14) for a plate with arbitrary surface temperature gradient are plotted in Fig. 4. As it can be seen, the numerical solution shows that the results have not significant variations for $\eta \succ 3$ which means dimensionless boundary layer thickness η_∞ to be order of 3.

Along the stagnation line ($\overline{X} = 0$), the x component of flow velocity,u, is governed by dimensionless function $I(\eta)$ according to Eq. (8), which rapidly approaches zero as η increases. Hence, for $\eta \succ 3$, surface motion has a negligible effect on the flow. In vicinity of the stagnation line, flow velocity u depends on both function $f'(\eta)$ and $I(\eta)$ (Eq. (8)). The effect of the free stream is governed by $f'(\eta)$ which is coincided to unity for $\eta \succ 3$. As can be seen from Fig. 4, the functions $f'(\eta)$ and $I(\eta)$ are in contradiction to each other, and the importance of surface motion depends on the distances from the stagnation line $\overline{X}$ as well as from the surface η.

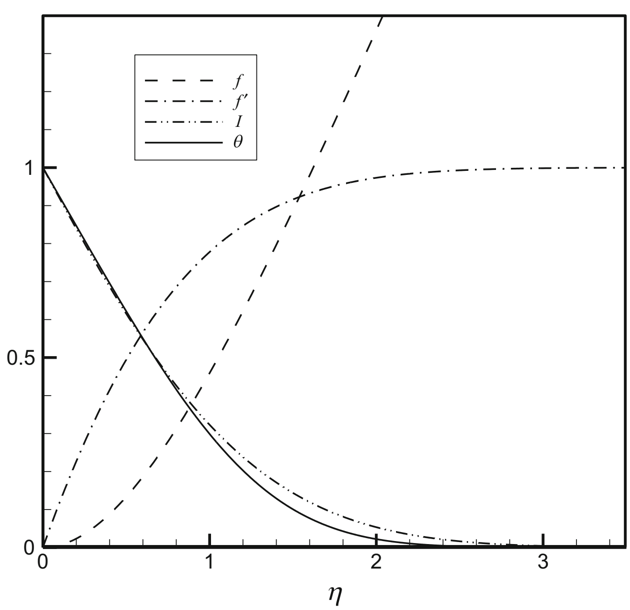

Fig. 4 Dimensionless profiles regarding flow velocity (f, f', I) and temperature (θ) in the stagnation region

Energy Eq. (14) is dependent on the four parameters Prandtl number Pr, dimensionless surface velocity $\overline{V}_{\mathrm{p}}$, dimensionless surface temperature gradient β and dimensionless local distance from stagnation line $\overline{X}$. First, the effect of Prandtl number on the heat transfer rate is investigated and then the effect of the other parameters will be investigated on the heat transfer for a given water jet temperature ($T_\infty = 75$ °C) which means a constant Pr (Pr = 2.4) and a given surface temperature at stagnation line in a range where single-phase cooling happens ($T_{\mathrm{s}_0} = 90$ °C). The local Nusselt number Nu_{w} can then be calculated from the following relation:

$$Nu_{\mathrm{w}} = \frac{hw_{\mathrm{j}}}{k} \tag{24}$$

The heat flux distributions in stagnation region can be presented with $\overline{X}$ as the independent variable and Nu_{w} as the dependent variable. However, as it is shown further in Fig. 5, the heat transfer distributions can be presented more generally with the dimensionless number $Nu_{\mathrm{w}}Re_{\mathrm{w}}^{-\frac{1}{2}}$ as the dependent variable. Using Eqs. (17) and (24) and with definition of free-stream velocity gradient near stagnation line as $C = \bar{C}\frac{v_{\mathrm{j}}}{w}$, dimensionless number $Nu_{\mathrm{w}}Re_{\mathrm{w}}^{-\frac{1}{2}}$ is calculated as:

$$Nu_{\mathrm{w}}Re_{\mathrm{w}}^{-\frac{1}{2}} = \frac{-\theta'(0)}{\theta_{\mathrm{s}}(x)}\sqrt{\overline{C}} \tag{25}$$

The variation of heat flux q'' and dimensionless number $Nu_{\mathrm{w}}Re_{\mathrm{w}}^{-\frac{1}{2}}$ with respect to the Pr number is depicted in Figs. 5 and 6. As can be seen, with increasing Pr as a result

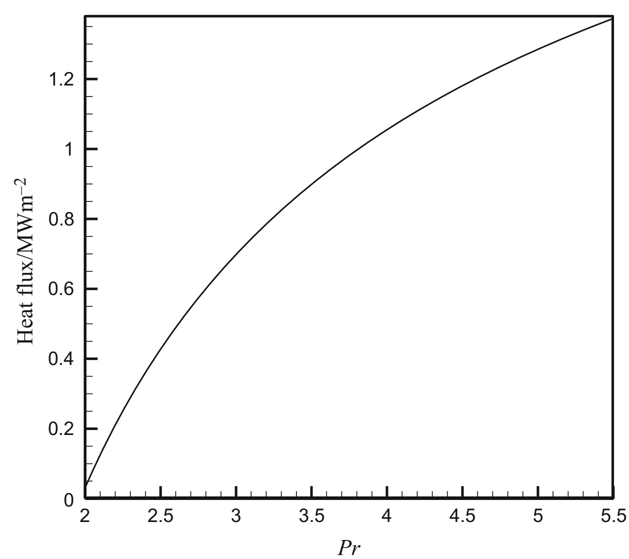

Fig. 5 Variation of q'', with respect to Pr

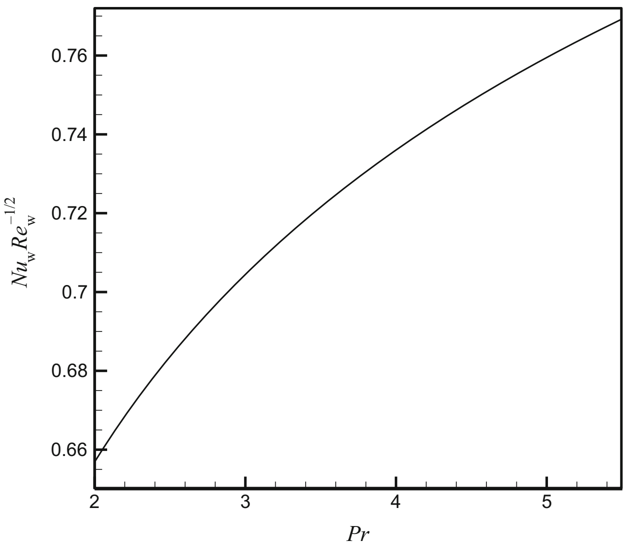

Fig. 6 Variation of heat transfer dimensionless number $Nu_w Re_w^{-\frac{1}{2}}$, respect to Pr

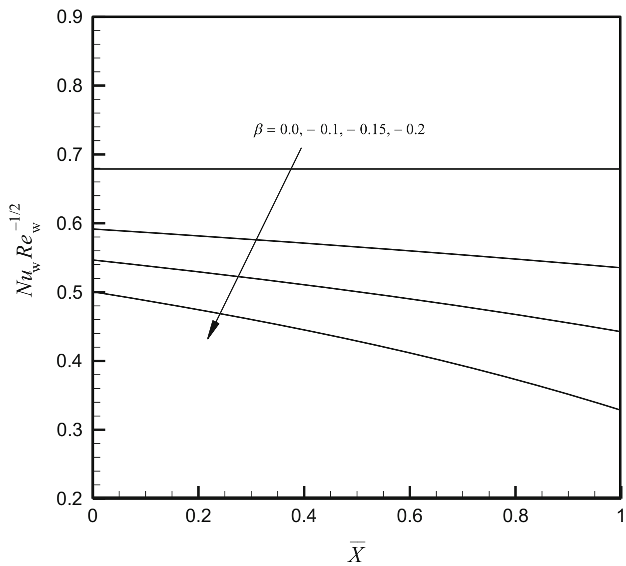

Fig. 7 Effect of dimensionless surface temperature gradient β on heat transfer in the stagnation region with $\overline{V}_p = 1$

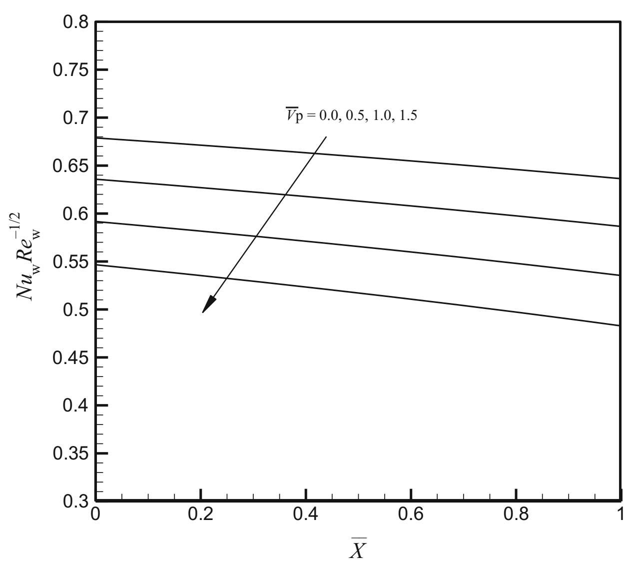

Fig. 8 Effect of surface velocity $\overline{V}_p$ on heat transfer in the stagnation region with $\beta = -0.1$

of decreasing free-stream temperature (jet temperature), the rate of the heat transfer increases. This is because with increasing the difference between flow and wall temperatures according to Eq. (2), the heat flux increases. It also can be observed that the trend of heat flux variation and dimensionless number $Nu_w Re_w^{-\frac{1}{2}}$ is totally the same. So, to investigate the effect of the other parameters on heat flux, the distribution of $Nu_w Re_w^{-\frac{1}{2}}$ is investigated and illustrated in the rest of this part.

When surface temperature is constant ($\beta = 0$), energy Eq. (14) is not a function of local distance and hence heat transfer does not change in stagnation region. So, values of $Nu_w Re_w^{-\frac{1}{2}}$ are equal in the vicinity of the stagnation line and have a constant amount of 0.68 in all points of stagnation region. With increasing temperature gradient at the surface ($|\beta| \uparrow$), temperature along the surface would decrease. As a result, the rate of heat transfer decreases. As shown in Fig. 7, with increasing β, this reduction along the surface would be more significant compared to the value of that in the stagnation line. Furthermore, when $\beta = 0$, the effect of the surface velocity is disappearing in energy equation and variations in $\overline{V}_p$ do not affect the rate of heat transfer in the stagnation region (Fig. 8).

To see the effect of surface velocity, distribution of $Nu_w Re_w^{-\frac{1}{2}}$ in stagnation region is illustrated for $\beta = -0.1$ and different surface velocities. As it can be seen, when the motion of surface increases, the rate of heat transfer decreases in all points of stagnation zone, and for the larger value of surface velocity, at a fixed location in stagnation region, $Nu_w Re_w^{-\frac{1}{2}}$ is smaller. The value of $Nu_w Re_w^{-\frac{1}{2}}$ decreases since warmer fluid near the wall is transported by the surface motion to regions located downstream with respect to the direction of surface motion which is the result of decreasing fluid temperature gradients in the direction perpendicular to the plate.

As previously shown, heat transfer at the stagnation line ($\overline{X} = 0$) on a moving plate depends on the surface speed $\overline{V}_p$ only when the surface temperature is spatially varying in accordance with Eqs. (14) and (22). A comparison between values of Nusselt number at the stagnation line of a moving plate with different velocities with those of the stationary plate for different values of β is shown in Fig. 9. The

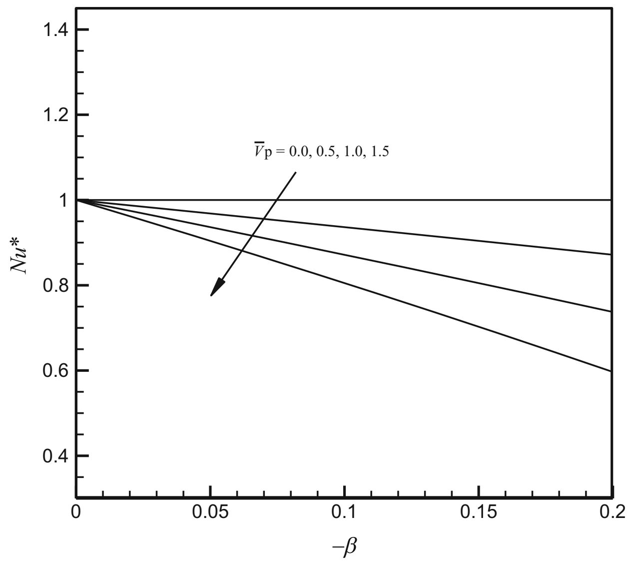

Fig. 9 Effect of surface velocity $\overline{V}_{\mathrm{p}}$ on heat transfer at the stagnation line ($\overline{X} = 0$) for different values of β

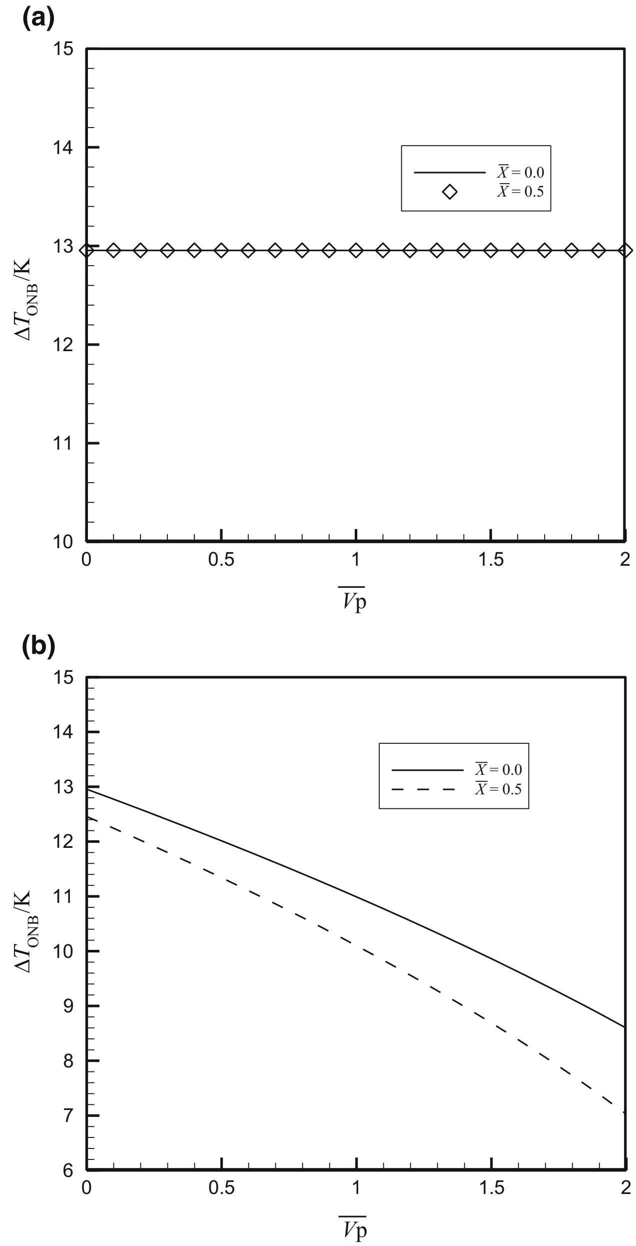

Fig. 10 Variations of temperature difference in the onset of nucleate boiling respect to the surface velocity in two local location of stagnation region **a** $\beta = 0$, **b** $\beta = -0.2$

values of Nu^* are smaller for larger magnitudes of the dimensionless surface temperature gradient. In terms of Nu^*, reductions are greater at higher values of $\overline{V}_{\mathrm{p}}$.

Two phases

It is known that the onset of nucleate boiling is an important point in boiling problems and especially in the present model which depends on it through suppression factor. The effect of velocity and temperature gradient of the surface on the temperature difference in the onset of nucleate boiling,ΔT_{onb}, at the stagnation line and at a typical local distance from it, $\overline{X} = 0.5$, is depicted in Fig. 10a, b. When $\beta = 0$, plate motion has no influence on ΔT_{onb} in all locations of stagnation region (Fig. 10a). When $\beta \neq 0$, ΔT_{onb} decreases with increasing $\overline{V}_{\mathrm{p}}$. Also, this reduction becomes larger with increasing local distance of stagnation line. As it was discussed before, in these conditions, the rate of heat transfer decreases and thus h decreases, and hence, according to Eq. (21) with decreasing h, the quantity ΔT_{onb} decreases (Fig. 10b).

Boiling incipience depends strongly on heat transfer coefficient of forced convection, Eq. (21). By decreasing h(as a result of decreasing surface temperature or in other word increasing β), the onset of nucleate boiling is shifted to the lower wall superheat temperatures as shown in Fig. 10b and also is observed in Fig. 11.

Figure 11 shows boiling curve that is plotted with heat flux through heated surface as the ordinate against wall superheat temperature when $\bar{V}_{\mathrm{p}} = 2$ and $\Delta T_{\mathrm{sub}_0} = 15\ ^{\circ}\mathrm{C}$. By considering jet velocity and liquid temperature (subcooling), as seen, onset of nucleate boiling temperature is larger than saturation temperature (Also seen in Fig. 3). For a surface temperature, less than T_{onb} flow is in single phase and heat transfer regime is forced convection. So, slope of the boiling curve is almost constant (Newton's law of cooling: Eq. (2)). Upon reaching temperature T_{onb}, the first nucleation sites are activated and the first bubbles are formed and grow. So, the evaporation is initiated and discrete bubbles begin to attach on the surface, leading to enhancing heat transfer. In this level, boiling regime is partial nucleate boiling where vapor generation is limited to a small population of bubbles and the bulk flow

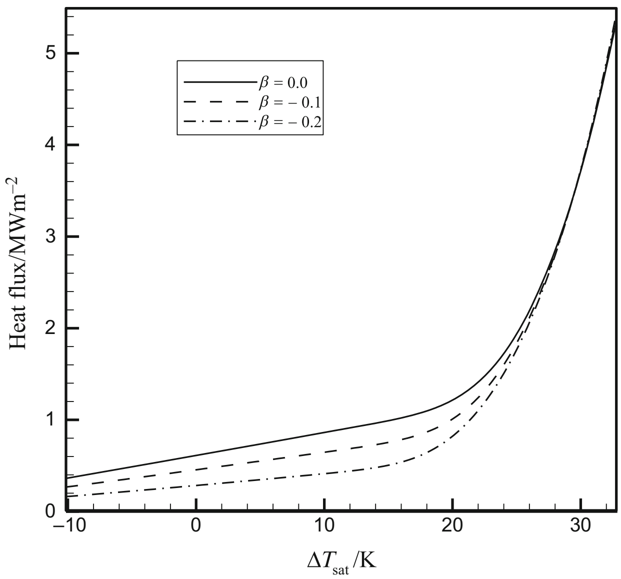

Fig. 11 Effect of dimensionless surface temperature gradient on the heat flux, when $\bar{V}_p = 2$ and $\Delta T_{sub_0} = 15\ °C$

continuing to strongly influence convection heat transfer from the surface. As a result of latent heat and bubble formation effects, the rate of heat transfer exceeds the single-phase case. With further increase in surface temperature, more bubbles form (increase in bubble density) and bubble departure from the surface increases. So, the heat flux increases abruptly in such a way that the slope of the boiling curve has a significant growth in fully developed boiling regime (Figs. 3, 11). The effect of temperature gradient of surface in single and nucleate boiling regime (partial and fully developed) through illustration of boiling curve at the stagnation line is presented in Fig. 11. As it can be observed, surface temperature gradient has a noticeable effect in single phase and less effect in partial nucleate boiling.

The effect of plate motion and local distance from stagnation line on boiling curve has the similar trends to the surface temperature gradient effect (Figs. 12, 13). With increasing surface velocity or local distance from stagnation line, the heat flux in single and partial nucleate boiling regime decreases. In the fully developed nucleate boiling regime, the surface conditions including the velocity and temperature gradient of the surface and also local distance from stagnation line have no significant influence on the rate of the heat transfer and the results almost collapse to a single curve. This is because in fully developed regime, the heat transfer mechanism is dominated by evaporation and bubbles motion and agitation [25].

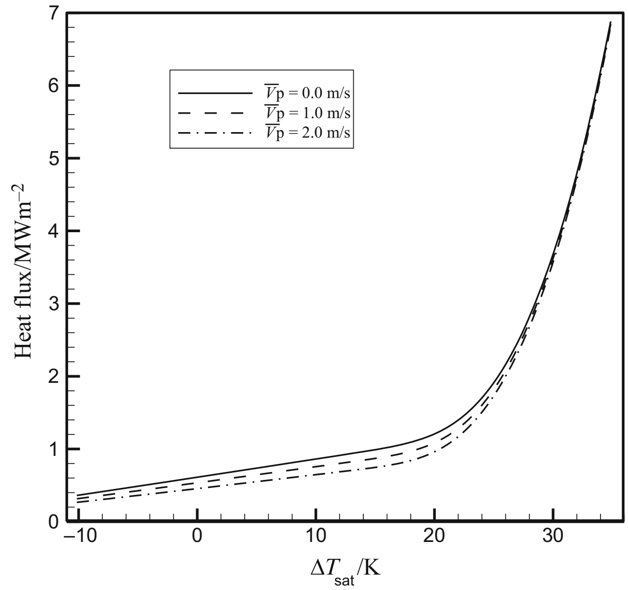

Fig. 12 Effect of surface velocity on the heat flux, when $\beta = -0.2$ and $\Delta T_{sub_0} = 15\ °C$

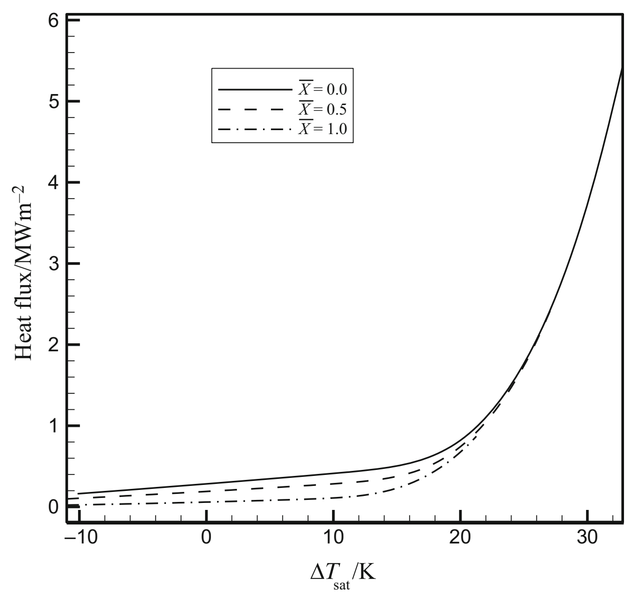

Fig. 13 Boiling curve in different distances of stagnation line, when $\bar{V}_p = 2$, $\beta = -0.2$ and $\Delta T_{sub_0} = 15\ °C$

Conclusions

The combination of a similarity solution and a numerical mechanistic model has been presented to investigate the effects of surface motion, arbitrary surface temperature distribution and local distance of stagnation line on single and nucleate boiling heat transfer in stagnation region of a laminar planar water jet impingement. In addition, the roles of main heat transfer parameters have been clearly disclosed. Results indicate that surface motion does not affect heat transfer when the surface temperature is constant. For a non-uniform surface temperature, the effect of surface motion has found to decrease convective heat transfer and this reduction becomes larger with distance from the stagnation line. Furthermore, when the surface temperature gradient is nonzero, the onset of nucleate boiling temperature decreases with increasing surface velocity in stagnation line and its neighborhood. However, these effects were most pronounced in the single-phase and partial boiling regimes, where heat transfer mechanism is dominated by the temperature and hydrodynamics of the bulk flow. Finally, it has been shown that within the fully developed boiling regime where heat transfer mechanism is dominated by evaporation and bubbles motion and mixing, the surface conditions including the velocity and temperature gradient of the surface and also local distance from stagnation line have no significant influence on the rate of the heat transfer.

References

1. Inada S, Miyasaka Y, Izumi R. A study on the laminar-flow heat transfer between a two-dimensional water jet and a flat surface with constant heat flux. Bull JSME. 1981;24(196):1803–10. https://doi.org/10.1299/jsme1958.24.1803.
2. Wang B, Guo X, Xie Q, Wang Z, Wang G. Heat transfer characteristic research during jet impinging on top/bottom hot steel plate. Int J Heat Mass Transf. 2016;101:844–51.
3. Dagtekin I, Oztop HF. Heat transfer due to double laminar slot jets impingement onto an isothermal wall within one side closed long duct. Int Commun Heat Mass Transf. 2008;35(1):65–75.
4. Ellahi R, Alamri SZ, Basit A, Majeed A. Effects of MHD and slip on heat transfer boundary layer flow over a moving plate based on specific entropy generation. J Taibah Univ Sci. 2018;12(4):476–82. https://doi.org/10.1080/16583655.2018.1483795.
5. Majeed A, Zeeshan A, Alamri SZ, Ellahi R. Heat transfer analysis in ferromagnetic viscoelastic fluid flow over a stretching sheet with suction. Neural Comput Appl. 2018;30(6):1947–55. https://doi.org/10.1007/s00521-016-2830-6.
6. Mahian O, Pop I, Sahin AZ, Oztop HF, Wongwises S. Irreversibility analysis of a vertical annulus using TiO_2/water nanofluid with MHD flow effects. Int J Heat Mass Transf. 2013;64:671–9. https://doi.org/10.1016/j.ijheatmasstransfer.2013.05.001.
7. Mahian O, Oztop H, Pop I, Mahmud S, Wongwises S. Entropy generation between two vertical cylinders in the presence of MHD flow subjected to constant wall temperature. Int Commun Heat Mass Transf. 2013;44:87–92. https://doi.org/10.1016/j.icheatmasstransfer.2013.03.005.
8. Selimefendigil F, Öztop HF. Analysis of MHD mixed convection in a flexible walled and nanofluids filled lid-driven cavity with volumetric heat generation. Int J Mech Sci. 2016;118:113–24. https://doi.org/10.1016/j.ijmecsci.2016.09.011.
9. Hassan M, Marin M, Alsharif A, Ellahi R. Convective heat transfer flow of nanofluid in a porous medium over wavy surface. Phys Lett A. 2018;382(38):2749–53. https://doi.org/10.1016/j.physleta.2018.06.026.
10. Miyasaka Y, Inada S. The effect of pure forced convection on the boiling heat transfer between a two-dimensional subcooled water jet and a heated surface. J Chem Eng Jpn. 1980;13(1):22–8. https://doi.org/10.1252/jcej.13.22.
11. Robidou H, Auracher H, Gardin P, Lebouché M. Controlled cooling of a hot plate with a water jet. Exp Thermal Fluid Sci. 2002;26(2):123–9.
12. Karwa N, Gambaryan-Roisman T, Stephan P, Tropea C. Experimental investigation of circular free-surface jet impingement quenching: transient hydrodynamics and heat transfer. Exp Thermal Fluid Sci. 2011;35(7):1435–43.
13. Modak M, Nirgude V, Sharma AK, Sahu SK. Experimental study on heat transfer characteristics of circular jet impingement boiling on the variety of structured copper surfaces in stagnation zone. 2016(50053):V005T15A48. https://doi.org/10.1115/icone24-60682.
14. Nobari AH, Prodanovic V, Militzer M. Heat transfer of a stationary steel plate during water jet impingement cooling. Int J Heat Mass Transf. 2016;101:1138–50.
15. de Brún C, Jenkins R, Lupton TL, Lupoi R, Kempers R, Robinson AJ. Confined jet array impingement boiling. Exp Therm Fluid Sci. 2017;86(Supplement C):224–34. https://doi.org/10.1016/j.expthermflusci.2017.04.002.
16. Agrawal C, Kumar R, Gupta A, Chatterjee B. Determination of rewetting velocity during jet impingement cooling of hot vertical rod. J Therm Anal Calorim. 2016;123(1):861–71. https://doi.org/10.1007/s10973-015-4905-5.
17. Wolf D, Incropera F, Viskanta R. Jet impingement boiling. Adv Heat Transf. 1993;23:1–132.
18. Zumbrunnen DA. Convective heat and mass transfer in the stagnation region of a laminar planar jet impinging on a moving surface. J Heat Transf. 1991;113(3):563–70. https://doi.org/10.1115/1.2910603.
19. Chen J, Wang T, Zumbrunnen DA. Numerical analysis of convective heat transfer from a moving plate cooled by an array of submerged planar jets. Numer Heat Transf Part A Appl. 1994;26(2):141–60. https://doi.org/10.1080/10407789408955985.
20. Chattopadhyay H, Biswas G, Mitra NK. Heat transfer from a moving surface due to impinging slot jets. J Heat Transf. 2002;124(3):433–40. https://doi.org/10.1115/1.1470489.
21. Chattopadhyay H, Saha SK. Simulation of laminar slot jets impinging on a moving surface. J Heat Transf. 2002;124(6):1049–55. https://doi.org/10.1115/1.1501089.
22. Ja'fari M, Rahimi AB. Axisymmetric stagnation-point flow and heat transfer of a viscous fluid on a moving plate with time-dependent axial velocity and uniform transpiration. Sci Iran. 2013;20(1):152–61. https://doi.org/10.1016/j.scient.2012.12.010.
23. Zumbrunnen DA, Incropera FP, Viskanta R. Method and apparatus for measuring heat transfer distributions on moving and stationary plates cooled by a planar liquid jet. Exp Thermal Fluid Sci. 1990;3(2):202–13. https://doi.org/10.1016/0894-1777(90)90088-O.
24. Gradeck M, Kouachi A, Lebouché M, Volle F, Maillet D, Borean JL. Boiling curves in relation to quenching of a high temperature moving surface with liquid jet impingement. Int J Heat Mass Transf. 2009;52(5):1094–104. https://doi.org/10.1016/j.ijheatmasstransfer.2008.09.015.
25. Mohaghegh MR, Rahimi AB. Modeling of nucleate boiling heat transfer of a stagnation-point flow impinging on a hot surface. Therm Sci. 2019. https://doi.org/10.2298/TSCI171220163M
26. Hall DE, Incropera FP, Viskanta R. Jet impingement boiling from a circular free-surface jet during quenching: part 2—two-phase jet. J Heat Transf. 2001;123(5):911–7.

27. Robidou H, Auracher H, Gardin P, Lebouche M, Bogdanić L. Local heat transfer from a hot plate to a water jet. Heat Mass Transf. 2003;39(10):861–7.
28. Robidou H, Auracher H, Gardin P, Lebouché M, Bogdanić L. Local heat transfer from a hot plate to a water jet. Heat Mass Transf. 2003;39(10):861–7. https://doi.org/10.1007/s00231-002-0335-6.
29. Vader D, Incropera F, Viskanta R. Convective nucleate boiling on a heated surface cooled by an impinging, planar jet of water. J Heat Transf. 1992;114(1):152–60.
30. Wolf D, Incropera F, Viskanta R. Local jet impingement boiling heat transfer. Int J Heat Mass Transf. 1996;39(7):1395–406.
31. Omar A, Hamed M, Shoukri M. Modeling of nucleate boiling heat transfer under an impinging free jet. Int J Heat Mass Transf. 2009;52(23):5557–66.
32. Timm W, Weinzierl K, Leipertz A. Heat transfer in subcooled jet impingement boiling at high wall temperatures. Int J Heat Mass Transf. 2003;46(8):1385–93.
33. Lienhard V JH. Liquid jet impingement. Ann Rev Heat Transf. 1995;6:199–270.
34. Vader DT, Incropera FP, Viskanta R. Local convective heat transfer from a heated surface to an impinging, planar jet of water. Int J Heat Mass Transf. 1991;34(3):611–23. https://doi.org/10.1016/0017-9310(91)90110-Z.
35. Rohsenow W. Heat transfer with evaporation, heat transfer—a symposium held at the University of Michigan during the summer of 1952. Ann Arbor: Univ. of Michigan Press; 1952.
36. Bergles A, Rohsenow W. The determination of forced-convection surface-boiling heat transfer. J Heat Transf. 1964;86(3):365–72.
37. Gorenflo D, Kenning D. H2 pool boiling. Vdi heat atlas. New-York: Springer; 2009. p. 757–92.