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Mixed convection jet impingement cooling of a moving plate

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ABSTRACT

The mixed convection heat transfer details of a jet impinging targeted a moving heated plate are studied numerically under steady laminar flow conditions. The governing parameters for the present problem are: Reynolds number (Re), Grashof number (Gr), the velocity of the heated plate (*U*), jet width and distance from the heated plate. The results are presented to show the fluid and heat flow structure at different conditions with fixed width of the jet and fixed distance from the heated plate and different other parameters. It is observed that the bouyancy effects are more obvious at low values of Reynolds number and plate velocity and the forced convection is dominating at high values of Re and *U*. At low values of the plate velocity ($U/V < 0.5$ with Re = 200 and Gr = 1×10^4) it is observed that the heat transfer is reduced due to the opposing effects of the jet flow and the flow induced by the plate on one side of the plate. However the increase of the plate velocity more than ($U/V > 0.5$ with Re = 200 and Gr = 1×10^4) the heat transfer is increased due to the combined effect of the jet flow and the stronger flow induced by the movement of the heated plate.

***Keywords**:* Mixed convection; jet impingement cooling; moving plate; numerical study.

Nomenclature

C_p	specific heat ($\mathrm{Jkg^{-1}\,K^{-1}}$)
d	jet width (m)
E	total specific energy (J/kg)
$\vec{g}$	gravity acceleration ($\mathrm{ms^{-2}}$)
Gr	Grashof number, $Gr = g\beta(T_h - T_c)d^3/(\mu/\rho)^2$
k	thermal conductivity ($\mathrm{Wm^{-1}K^{-1}}$)
L	length of the heated surface (m)
Nu	local Nusselt number, $Nu = q_w d/[(T_h - T_c)k]$
$\overline{Nu}$	average Nusselt number
p	pressure ($\mathrm{Nm^{-2}}$)
q_w	wall heat flux ($\mathrm{Wm^{-2}}$)
Re	Reynolds number, $Re = \rho V d/\mu$
T	temperature (K)
U	heated plate velocity ($\mathrm{ms^{-1}}$)
$\vec{V}$	velocity vector

V jet velocity (ms^{-1})
w jet-to-heated surface spacing (m)
x, y Cartesian coordinates (m)

Greek Symbols
β coefficient of thermal expansion (K^{-1})
μ dynamic viscosity (kg.m^{-1}s^{-1})
ρ density (kgm^{-3})
$\bar{\bar{\tau}}$ viscous stress tensor (Nm^{-2})
ϕ general dependent variable

INTRODUCTION

The convective heat and mass transfer characteristics of the Impinging jets have attract the attention of many researchers over the last several decades. The interest was estimated due to many applications in, for example the annealing of metal sheets, tempering of glass plate, drying of textile and paper products and other applications [1-3]. Representative studies in the jet impingement heat and fluid flow research area may be found in the studies by Garimella [4] Zuckerman and Lior [5], Lee et al. [6] Saeid [7] among others.

Jet impingement cooling frequently used in manufacturing processes to cool/heat or dry a material of different shapes. Materials are often in motion since many manufacturing processes are designed for mass production. In such cases, there will be two driving forces in moving the fluid around the material. One comes from the jet flow and the other is from the moving surface. The combination effect of the velocities is important to predict the effectiveness of the heat transfer process. The jet impingement cooling of a moving surface is studied by Raju and Schlünder [8]. They have studied experimentally the convection heat transfer using one jet on a moving belt. The jet velocity is selected relatively high to include the turbulent jet flow ($4 < V_{jet} < 40$ m/s) and the belt speed ($0.\ 15 < V_{Belt} < 5.\ 5$ m/s). The results show that the rate of heat transfer are increased to about 1.5 to 2.0 times than the case of stationary surface. Aldabbagh and Mohamad [9] conducted a numerical study to investigate the effects of increasing the number of jet slots on a moving heated plate on the flow field structure and heat transfer characteristics. Three rows and eight columns square jets on a heated moving plate. There results show that Nusselt number variation show periodic oscillation shape. Zumbrunnen [10] used the similarity method to investigate the effects of the moving surface on the laminar planar jet perpendicular to the jet plane. Chattopadhyay and Saha [11] studied numerically the heat and fluid flow characteristics on a moving surface under a bank of slot jets. Axial and knife-jet were selected in their investigations. Their results show that, the heat transfer from the axial jet is much higher than rate of heat transfer using jets with exit angle of 60 deg at all surface velocities considered under fixed other parameters. Chattopadhyay and Saha [12] extended their work [11] to consider the effect of the moving heated surface on the jet impingement cooling by rectangular slot nozzle. They used the large eddy simulation technique to calculate the details of turbulence flow, such as Reynolds stresses. Many other studies considered the effect of turbulence in the jet impingement cooling, examples can be found in the studies of Yang and Hao [13], Sharif and Banerjee [14], Benmouhoub and Mataoui [15].

The unconfined and confined jets were investigated by Rahimi and Mortazaei [16] using a new fully implicit finite-difference technique. The details of the flow and thermal fields were obtained with reasonable agreement with the results of other methods. Recently, Lawal et al. [17] studied numerically the steady and unsteady confined slot jet impingement cooling of an isoflux surface. Their study show that the unsteadiness started at critical Reynolds number of the flow was found to have important impact on the heat and fluid flow for the heated surface. Sarhan [18] conducted a numerical study of slot jet impinging onto a sinusoidal target surface in laminar flow regime. The Nusselt number at the surface are evaluated at different cases of the governing parameters. Ersayın and Selimefendigil [19] investigated numerically the jet impingement flow, on a heated moving plate. They observed that the peak value of Nusselt number can be increased by adding nanoparticles. They found that heat transfer is enhanced at low frequencies of pulsating jet flow. Başaran and Selimefendigil [20] studied numerically the effect of the motion of an isothermal plate on the heat transfer under double slot jets located symmetrically around the vertical axis and impinging vertically on the target plate. A mixture of water and Aluminum oxide nanoparticles was used as working fluid in their investigation. The results prove that the heat transfer can be enhanced by increasing the plate velocity, increasing Reynolds number and increasing the volumetric fraction of the nanofluid. Selimefendigil and Öztop [21] considered the numerical study of the rectangular jet with pulsating nanofluids impingement on an isothermal flat surface. In the steady state cases, they found adding nanoparticles increases the peak value of the stagnation Nusselt number. The results for the pulsating jet cases show that the effect of pulsation and nanoparticles is not increase the Nusselt number at the stagnation point compared with the steady case. Selimefendigil and Öztop [22] extended their work [21] to consider jet impingement heat transfer from a corrugated surface with nanofluid of water–SiO_2. The optimal value for the corrugation amplitude and frequency was found to increase the average Nusselt number for the flow with high Reynolds number. Also, their results show that the jet impingement cooling is enhanced for a corrugated surface compared to a flat case.

It is observed from the above literature review that relatively less studies have been reported on the combination effects of the buoyancy and the surface motion on the jet impingement flow field and heat transfer. Motivated by the lack on such important study, the present study focused on the mixed (forced and free) convection heat and fluid flow of jet impingement heat transfer of a moving heated plate.

MATHEMATICAL MODEL

The schematic diagram of an impinging jet on a moving plate is shown in Figure 1. The jet implement through a slot with width d and distance w from the target-heated surface. Following the benchmark studies, the width of the slot jet is selected to be 10 mm, the jet-to-heated surface spacing is fixed as w = 40 mm, and the heated plate is L = 200 mm (w/d = 4 and L/d = 20) for all the simulation cases.

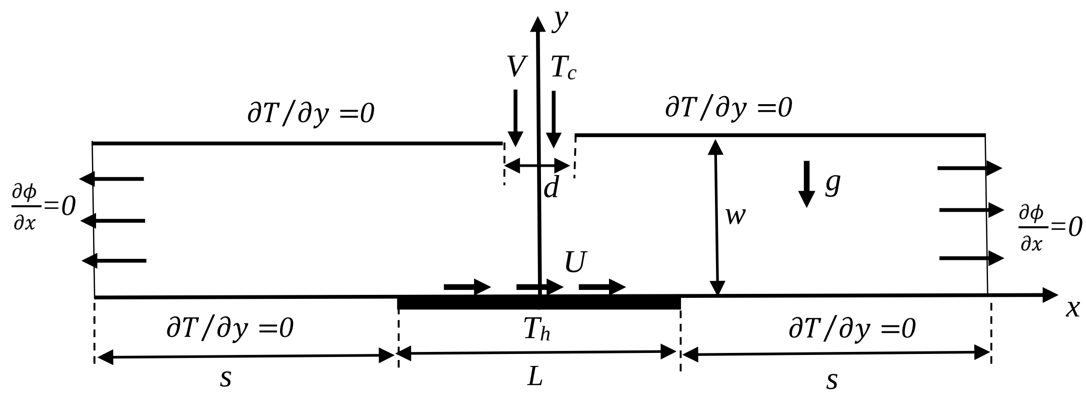

Figure 1. Schematic diagram of the physical model and coordinate system

The mathematical model of the present case is based on the following assumptions:

1. the flow is laminar and incompressible approximated as two-dimensional;
2. the temperature and velocity profiles at the inlet are assumed to be uniform;
3. the physical properties of the air are constants except the density, and
4. the Boussinesq approximation is used for the density variation with temperature;

Therefore, the governing equations for the steady heat and fluid flow can be written as:

$$\nabla.\left(\rho\,\vec{V}\right) = 0 \tag{1}$$

$$\nabla.\left(\rho\,\vec{V}\vec{V}\right) = \rho\vec{g} - \nabla p + \nabla.\bar{\bar{\tau}} \tag{2}$$

$$\nabla.\left\{\vec{V}(\rho E + p)\right\} = \nabla.\left\{k\nabla T + \bar{\bar{\tau}}.\vec{V}\right\} \tag{3}$$

The stress tensor $\bar{\bar{\tau}}$, is given by:

$$\bar{\bar{\tau}} = \mu\left\{\left(\nabla\vec{V} + \nabla\vec{V}^{T}\right) - \frac{2}{3}\nabla.\vec{V}I\right\} \tag{4}$$

where $\vec{V}$ is the physical velocity vector, p is the static pressure, E is the total specific fluid energy ($E = C_pT - p/\rho + V^2/2$), T is the fluid temperature and I is the unit tensor. The fluid properties are ρ is the fluid density, μ is the molecular viscosity, C_p is the specific heat, and k is the thermal conductivity. The Boussinesq approximation $\rho = \rho_o[1 - \beta(T - T_o)]$ is used, where the subscript o for reference point (jet inlet).

No-slip (zero velocities) boundary conditions are imposed on the solid surfaces and outflow boundary condition is imposed at the exit plan with zero gradient in the x-direction of all the dependent variables as shown in Figure 1. The upper and the lower walls are thermally insulated except the target plate where the temperature is fixed constant (T_h) which is need to be cooled by air jet at temperature (T_c).

Most of the benchmark results are presented for air with constant Prandtl number, Pr = 0.71. The average temperature between the jet and the hot plate is assumed to be 70^{o}C so that the Prandtl number is around 0.71. The heated surface temperature is assumed constant at 110^{o}C and the jet temperature is maintained at 30^{o}C. The air properties were found at an average temperature of 70^{o}C as: density ρ_o = 1.028 kg/m^3, specific heat C_p = 1007 J/kgK, thermal conductivity k = 0.02881 W/mK, dynamic viscosity of μ= 2.052$\times$10^{-5} kg/ms and the coefficient of thermal expansion of β = 0.0029 K^{-1}.

NUMERICAL SOLUTION PROCEDURE

The governing equations (Equations (1)-(4)) subjected to the boundary conditions (specified in the mathematical model) are solved numerically using FLUENT-ANSYS 18 [23] software based on the finite volume method [24]. This FVM is based on the meshing of the solution domain into finite number of control volumes (or cells) and integrate the governing equations over each cell. Therefore the domain shown in Figure 1 is discretised into uniform quadrilateral cells.

The central differencing method is used to approximate the diffusion terms of the energy equation and momentum equations. The power law scheme [24] is employed for the convection-diffusion terms discretization of the momentum and energy equations. The resulting algebraic equations were solved using the pressure correction techniques with SIMPLEC algorithm [25]. This method need iteration. To avoid divergence in the iteration the under relaxation factors of 0.3, 0.7, and 1.0 are used for solving the pressure, momentum and energy equations respectively. The criterion to stop the iteration is based on the values of the maximum residual in the equations. In the present study, the residual in the energy equation should be less than 10^{-7} and the residual of other variables were lower than 10^{-3} in the converged solution. As verification, the global heat and mass balance is checked and found within $\pm 10^{-3}$ % in the converged solutions.

The length of the lower thermally insulated surface has an important effect on the accuracy of the results, where the outflow boundary condition at the exit plan can be accurate. In the present investigations the length of the lower wall is fixed to be 5 times the heated surface (L/2) from each side about y-axis. The simulations with longer lower adiabatic wall generate results with negligible difference in the average Nusselt number values.

The numerical solution of the mathematical model leads to velocity and thermal fields. The thermal field is used to calculate the wall heat flux dissipated from the heated surface. The performance of the cooling is evaluated based on the Nusselt number, which is defined by the following equation:

$$Nu = \frac{q_w d}{(T_h - T_c)k} = \frac{-k_{eff}(\partial T/\partial y)_{y=0} d}{(T_h - T_c)k} \tag{5}$$

where q_w is the wall heat flux. The average Nusselt number on the heated plate is calculated by integrating the local values over the length of the heated surface as follows:

$$\overline{Nu} = \frac{1}{L}\int_{-L/2}^{+L/2} Nu\, dx \tag{6}$$

The variation of $\overline{Nu}$ with Re for the forced convection (with Gr = 0) is generated using slot jet width of 10 mm (w/d = 4 and L/d = 20) with fixed target plate temperature of 110 ^{o}C and jet air temperature of 30 ^{o}C. The results are generated for a stationary plate (U = 0) and the assumption of zero gravity acceleration (Gr = 0) and constant air density.

The jet flow is split symmetrical around y-axis into two parts with a stagnation point in the middle. The region near the stagnation point is called the stagnation region. In this region the flow experiences high acceleration turning from vertically downward to horizontally along the heated surface. This leads to high rate of heat transfer and therefore high values of local Nusselt number in this region as shown in Figure 2. Far from the stagnation point region is called the wall region. In the wall region the shear forces generated between the fluid and the heated surface leads to slow down the flow and reduce the rate of heat transfer.

The present results are in good agreement with the references results as presented in Figure 2. The maximum differences between the present simulations results and the references are near the stagnation point and near the end of the heated surface. The differences between the present results using 1mm mesh and the results of Rady is 0.9% % and Al-Sanea et al is 1.02% at the stagnation point. However near the end of the heated surface the error between the present results using 1mm mesh and the results of Rady is less than 15% and Al-Sanea et al is around 40% at x = 0.09 m. The difference between the present study and the references results may be due to solution methods and mesh size and geometry. The effect of mesh size on the accuracy of the results is studied for the above benchmark case. The results presented in Figure 2 and Table 1 show that the meshes with different sizes have minor effects on the values of the local and average Nusselt number respectively. Therefore the results obtained using uniform mesh cells of size 1 mm × 1 mm can be considered acceptable results and will be used for all the simulation cases considered in the present study.

Table 1. Values of $\overline{Nu}$ with different mesh size for the benchmark case of Gr = 0 and *U*/*V* = 0.

Re	Al-Senea [26]	Lin et al [27]	Rady [28]	Present results mesh cell size (0.5 mm×0.5 mm)	Present results mesh cell size (1 mm×1 mm)
100	1.49	1.88	1.46	1.399	1.483
200	2.50	2.65	2.38	2.454	2.439

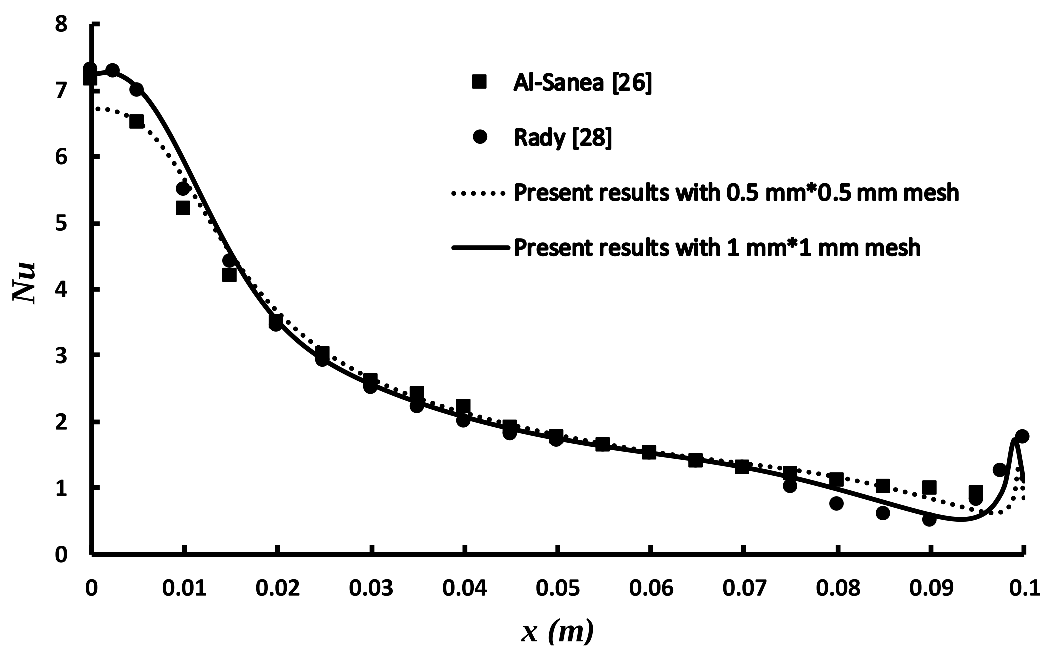

Figure 2. Variation of Nusselt number along the heated plate with Re = 200, Gr = 0 and *U*/*V* = 0

RESULTS AND DISCUSSION

The bouyancy effects on the heat and fluid flow is studied for three values of Grashof number (1.44×10^3, 5.74×10^3 and 1.00×10^4). The values of Grashof number are considered with fixed length scale (jet width) and air properties at average temperature of 70°C. The results with different values of Gr are generated by simply changing the temperature of the heated plate and the temperture at the jet inlet while keeping the average temperature of 70°C.

Figure 3 shows the effect of the bouyancy (Gr) on the jet impingement cooling of a stationary plate ($U = 0$) at different values of jet inlet velocity (Re). The results show that the influence of the bouyancy is more effective at low values of jet inlet velocity (or Re). Figure 3 indicates that at high values of Re (Re>300) the values of the average Nuessult number will form a signle curve indicating the forced convection domination with negligable effects of the bouyancy.

The effects of the moving plate are studied with fixed Reynolds number of 200 and two values of Gr (Gr = 0 and Gr = 1×10^4). For the case of the stationary plate the jet flow will be split symmetrical around y-axis into two parts with a stagnation point in the middle as shown by the solid lines in Figures 4a and 4b. On the other hand, the movement of the heated plate will induce a flow in the same direction of the plate due to the no slip condition. This leads to have opposing flow in one direction and adding flow in the other direction and the stagnation point will be shifted in the direction of the plate movement.

The results presented in Figure 4a showing the variation of the local Nusselt number along the heated plate with different plate velocity and constant jet inlet velocity with negligible buoyancy effects (Gr = 0). The values of the local Nussult number show maximum values at the end of the left hand side of the heated plate and gradually deceased to reach the minimum near the region x = - 0.03m to x = - 0.01m depends on the plate velocity. The local Nuesslt number then increased at the stagnation point, which is shifted in the present case of the plate movement towards the right hand side of the midpoint.

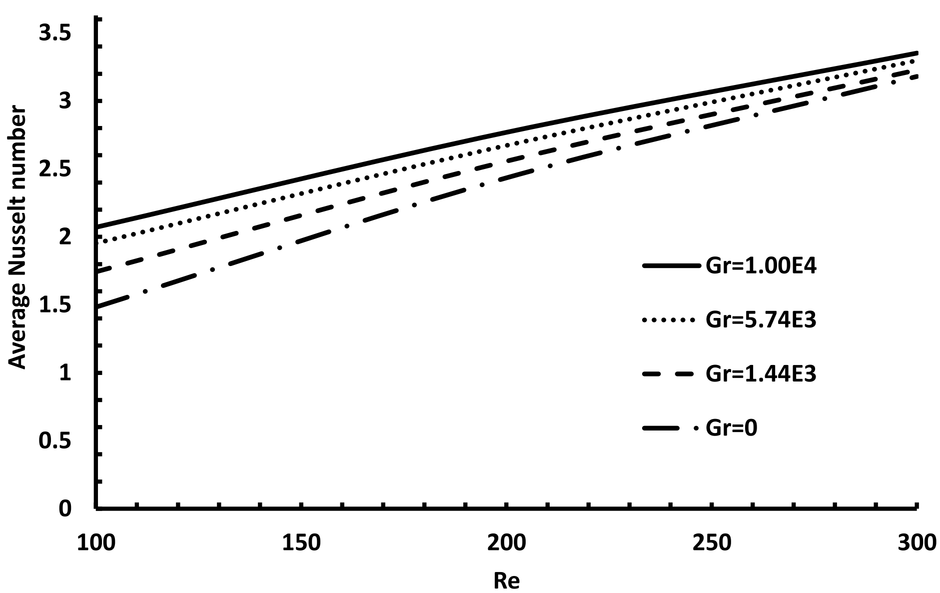

Figure 3. Variation of average Nusselt number with Reynolds number with $U/V = 0$

It is observed from the above results that the values of the local Nusselt number in the stagnation point are decreased with the increase of the plate velocity *U* for all the cases considered in the present study. However the values of the Nusselt number increased in the right hand side compared with the stationary plate case due to the adding effects of the jet flow and the flow induced by the plate movement. Similar observations can be made for the variation of the local Nusselt number for the case of Gr = 1 $\times 10^4$ with higher values of Nusselt number as shown in Figure 4b.

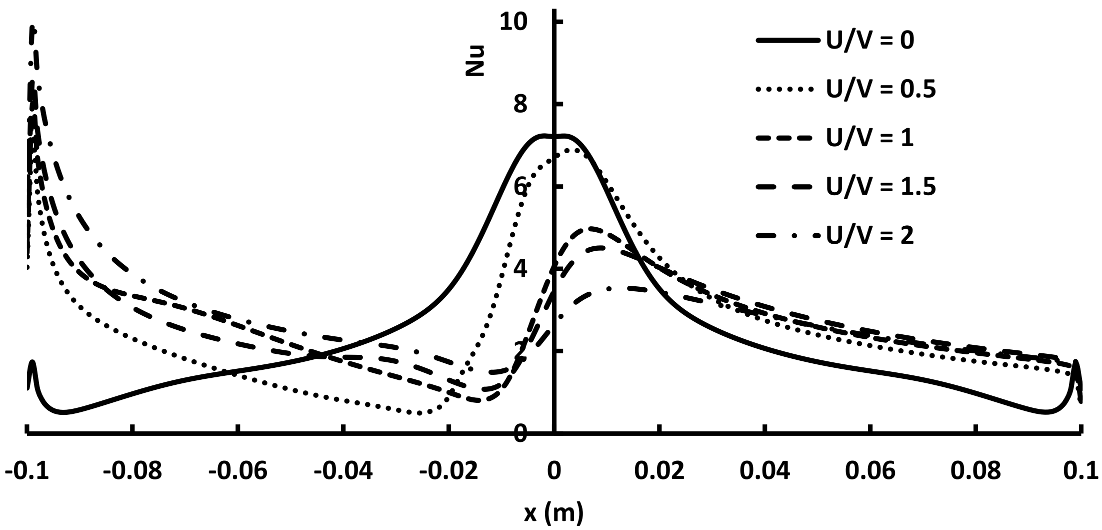

Figure 4a. Variation of Nusselt number along the heated plate with Re = 200 and Gr = 0

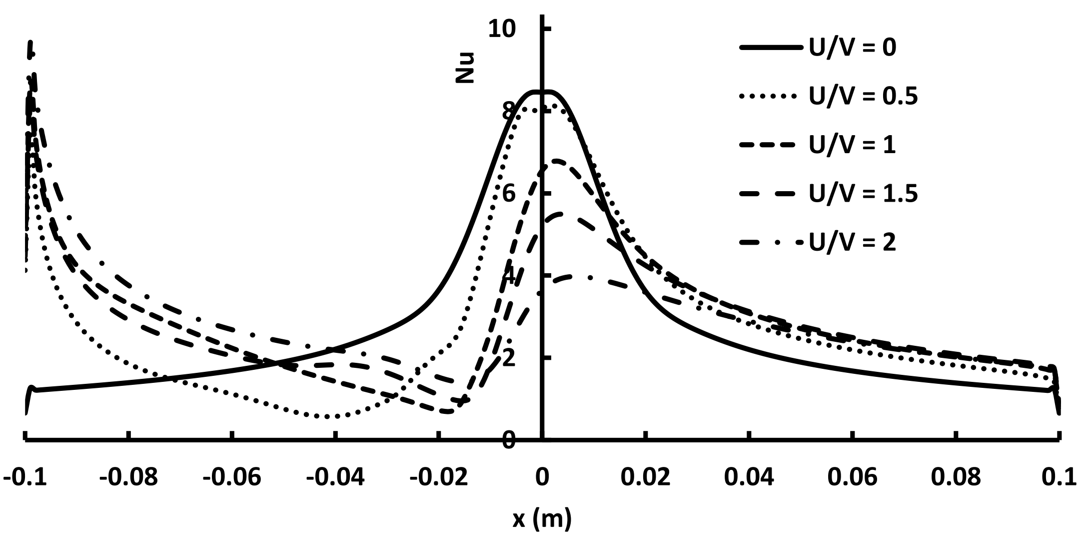

Figure 4b. Variation of Nusselt number along the heated plate with Re = 200 and Gr = 1×10^4

The overall effects of the heated plate movement and the buoyancy effects can be summarized by calculating the average Nusselt number for different plate velocities (*U*/*V*) and different temperatures of the plate and the jet inlet (Grashof number). The results presented in Figure 5 show that the average Nusselt number increased by increasing the temperature differences between the plate and the jet inlet (Gr) with fixed other parameters.

Again it is observed from Figure 5 that the variation curves approaching each other at high values of the plate velocity. Therefore the effects of the buoyancy are more effective for the cases with low velocities of the flow induced by the plate movement. For the cases with Gr = 0, the values of the average Nusselt number are increased by increasing the plate velocity. On the other hand for the cases with Gr > 0, the values of the average Nusselt number is decreased with the plate movement at low velocities. For the cases of plate movement with $U/V < 0.5$ and Re = 200, the heat transfer from the plate is decreased compared with the stagnant plate and fixed other parameters as shown in Figure 5. This is due to the combination effects of the opposing flow of the jet flow and the flow induced by the plate movement and buoyancy.

However the increase of the plate velocity more than ($U/V > 0.5$ with Re = 200) the heat transfer is enhanced and in some cases decreased again due to the combined effects of the jet flow, buoyancy induced flow and the stronger flow induced by the movement of the heated plate.

To investigate the reasons of such variation of the average Nusselt number, the flow structure details near the heated plate are shown in Figures 6 to 9 for the cases of the flow cases when Re = 200 and Gr = 1×10^4 with U/V = 0.5, 1.0, 1.5 and 2.0 respectively.

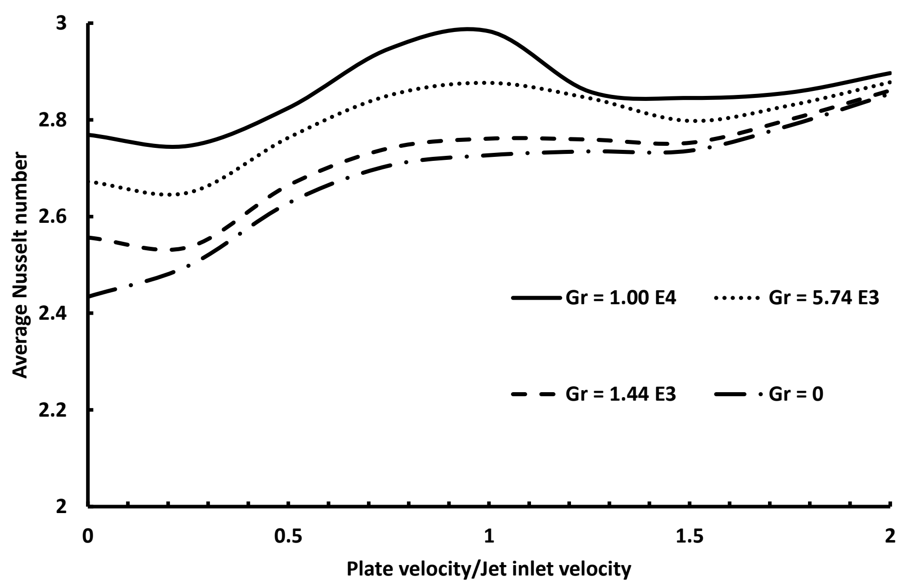

Figure 5. Variation of average Nusselt number against plate velocity with fixed Re = 200

The opposing flow interaction can be seen on the left hand side of the jet inlet in Figure 6. The fluid lyer on the top of the plate is moving in the positive x-direction (to the right) while in the regine far from the plate the flow is in the negative x-direction (to the left). The movement of the layer near the moving plate is induced mainly due to the viscous forces creating a rotating flow cell on the top on the left hand side of the jet inlet as shown in Figure 6. On the other hand the jet flow and the flow induced by the plate are in the same direction on the right hand side of the jet inlet. The pattern of the velocity vectors presented in Figure

6 reflected on the isotherms as shown in the isotherm contours in the bottom of Figure 6. There is a hot spot near the plate in the region on the left hand side of the jet inlet shown in Figure 6 due to the fluid circulation with relatively low velocities. This leads to reduce the rate of heat transfer and therefore the local Nusselt number in this region as shown in Figure 4b and the average Nusselt number as shown in Figure 5.

Figure 7 depicted the velocity vectors and the isotherms for the same case with increasing the plate velocity to U/V = 1.0 Similar observations can be seen in Figure 7 for both vectocity vectors and isotherms. However the flow circulation on the left hand side of the jet inlet is smaller than the previous case as shown by comparing the resultes presented in Figures 6 and 7 for the cases with U/V = 0.5 and U/V = 1.0 respectively. This leads to increase the values of the local Nusselt number on the left hand side of the jet inlet for the case with U/V = 1.0 than that with U/V = 0.5 as shown in Figure 4b. On the right hand side the stagnation Nusselt number is reduced by increasing the plate velocity as shown in Figure 4a and 4b. This is due to the combination effect of the shear driven by moving of the surface and the jet impingement flow. Far from the stagnation point on the right hand side, the local Nusselt number are almost not infuenced by the plate velocity as shown in Figure 4a and 4b.

Increasing the plate velocity further to U/V = 1.5 with Re = 200 and Gr = 1×10^4 leads to reduce the average Nusselt number as depected in Figure 5. The velocity vectors presented in Figure 8 show that the flow circulation on the top of the left hand side of the jet is pushed down towards the heated plate compare to the case with lower plate velocity. It is observed also that the vectocitis magnitude in the circuation cell become smaller compare to the the case with lower plate velocity. This leads to reduce the local (and therefore the average) Nusselt number on the left hand side and the stagnation point as shown in Figure 4b and Figure 5.

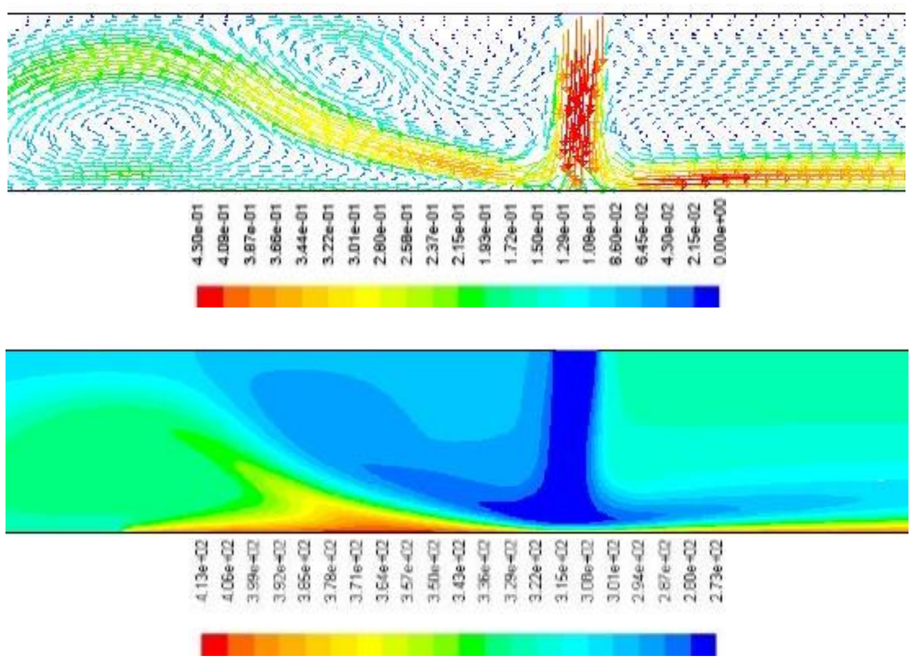

Figure 6. Velocity vectors (m/s) and isotherms (K) with Re = 200, Gr = 1×10^4 and U/V = 0.5. Velocity vectors on the top and isotherms at the bottom.

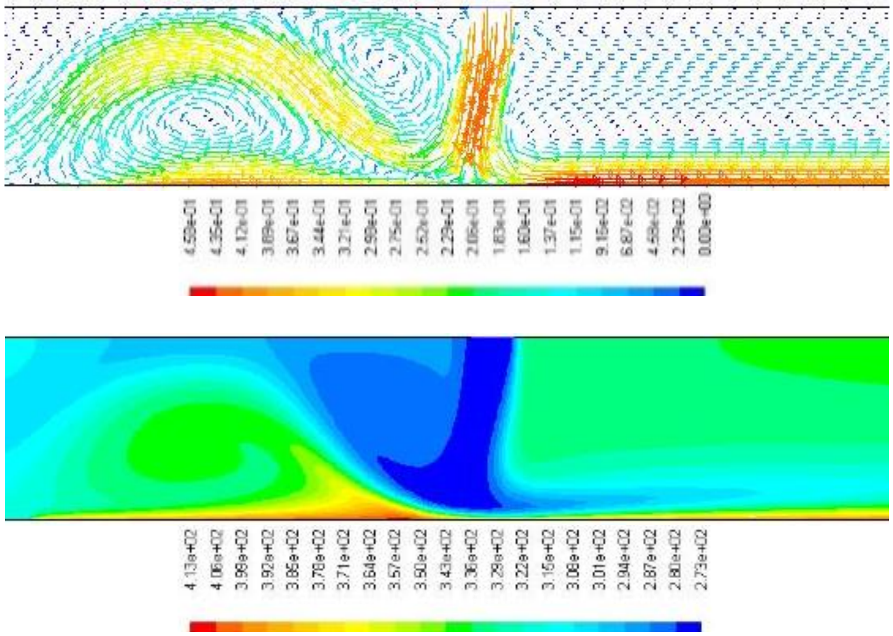

Figure 7. Velocity vectors (m/s) and isotherms (K) with Re = 200, Gr = 1×10^4 and U/V = 1.0. Velocity vectors on the top and isotherms at the bottom.

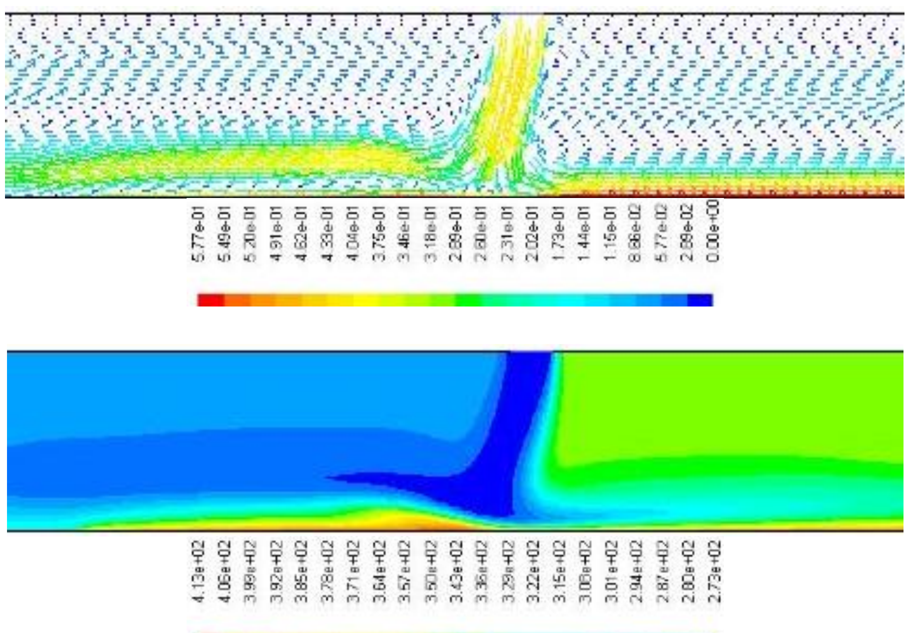

Figure 8. Velocity vectors (m/s) and isotherms (K) with Re = 200, Gr = 1×10^4 and U/V = 1.5. Velocity vectors on the top and isotherms at the bottom.

Finally, Figure 9 depicted the velocity vectors and the isotherms for the same case with increasing the plate velocity to U/V = 2.0 Similar observations can be seen in Figure 8 and 9 for both vectocity vectors and isotherms. However the high velocity of the heated palte leads to reduce the efect of the air circulation on the top of the plate on left hand side of the jet. The heat is transfer by convection currents generated by the plate movement and the average Nuessult number increased as shown in Figure 5.

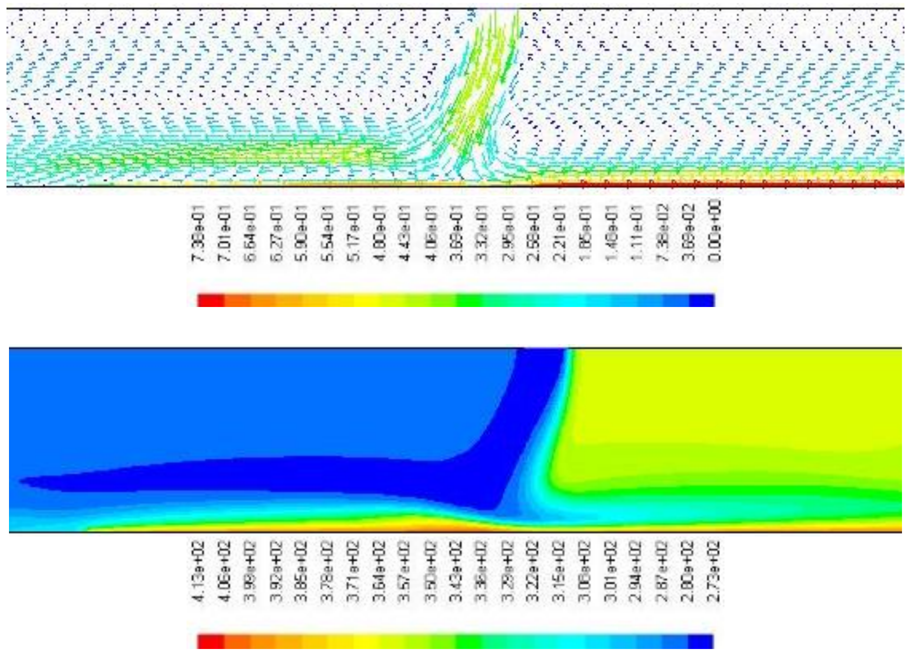

Figure 9. Velocity vectors (m/s) and isotherms (K) with Re = 200, Gr = 1×10^4 and U/V = 2.0. Velocity vectors on the top and isotherms at the bottom.

CONCLUSIONS

In the present investigations the laminar mixed convection jet impingement cooling of a horizontal moving plate is presented. The governing parameters for the present problem are: Reynolds number, Grashof number and the velocity of the heated plate with fixed jet width and distance from the heated plate.

The numerical results show that the influence of the bouyancy is more effective at low values of jet inlet velocity (or Re). At high values of Re the results show the domination of the forced convection with negligable effects of the bouyancy (or Gr). The results show that the buoyancy forces are more effective for the cases with low velocities of the flow induced by the plate movement.

The heated plate movement in the positive x-direction leads to increase the local Nusselt number near the end of the left hand side of the heated plate and gradually decreased to reach the minimum value near the midpoint. The stagnation point is shifted in the case of the plate movement in the direction of the plate movement. It is found that the values of the stagnation Nusselt number are decreased with the increase of the plate velocity for all the cases considered in the present study.

For the cases with Gr = 0, it is found that the values of the average Nusselt number are increased by increasing the plate velocity. However, for the cases with Gr > 0, it is observed that the average Nusselt number is decreased with low velocities of the heated plate (U/V < 0.5 with Re = 200). However the increase of the plate velocity more than (U/V > 0.5 and Re = 200) the average Nusselt number is increased due to the combined effect of the jet flow and the stronger flow induced by the movement of the heated plate.

The present study is limited for laminar flow. For higher values of plate velocity, Reynolds number and Grashof number, turbulence effects will play a role. In these cases, a suitable turbulence model should be selected which need further investigation.

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