Numerical Investigation of Three Dimensional Laminar Jet Impingement on Heated Plate Surface

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Abstract

A numerical three dimensional investigation is performed to study the effects of the heat flux and Reynolds number on the local heat transfer distribution on smooth flat plate surface impinged with normal air jet. Reynolds number based on nozzle exit condition is varied between 1000 and 4000 and heat flux from 500 to 1000 W/m^2.

The flow domain was studied by solving Navier Stokes equations by finite volume while the conduction in the plate was studied by solving Laplace equation. The results are expressed in terms of flow field and isotherm contours. The temperature of the plate surface increases when heat flux increases. The thickness of the thermal boundary layer and plate surface temperature decreases when Reynolds number increases.

Keywords: heat flux, Reynolds number, Naviar Stokes equation, finite volume.

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Nomenclature

d	diameter (m)
k	conductivity
L	domain height (m)
Nu	Nusselt number
Pr	Prandtl number
P	pressure (N/m^2)
q	heat flux (W/m^2)
Re	Reynolds number
S	source term for discretization equation.
T	temperature (C)
ΔT	wall temperature difference.
u	velocity in x direction (m/s)
v	velocity in y direction (m/s)
w	velocity in z dirction (m/s)
x	coordinate (m)
y	coordinate (m)
z	coordinate (m)

Greek symbols

Φ	dependent variable used in discretization equation.
Γ_Φ	diffusion coefficient used in discretization equation.
μ	dynamic viscosity (kg/m s)
ρ	density of the fluid (kg/m^3)

Subscripts

h	hydraulic
in	inlet
s	surface

1.Introduction

The power consumption and heat dissipation from the electronic components is very vital. The restrictions in both speed and size of the electronic components compelled to upgrade the circuit design and materials to decrease power dissipation considerably through cooling technologies such as free convection and forced convection.

There are number of traditional cooling techniques are in use, major traditional techniques are heat sink, heat sink with fan, heat exchanger, heat pipes etc. These conventional methods have lot of limitations such as low heat removal rate, space constraints in very small electronic equipments. Therefore, an effective and efficient method for removing heat is needed. Jet impingement cooling is an efficient method as it provides high heat removal rate [1]. Jet impingement is one of the very efficient solutions of cooling hot objects in industrial processes as it produces a very high heat transfer rate through forced-convection. There is a large class of industrial processes in which jet impingement cooling is applied such as the cooling of blades/vanes in a gas turbine, the quench of products in the steel and glass industries and the enhancement of cooling efficiency in the electronic industry. Over the past 30 years, experimental and numerical investigations of flow and heat transfer characteristics under single or multiple impinging jets remain a very dynamic research area [2]. Heat transfer rates in case of impinging jets are influenced by various parameters like Reynolds number, jet-to-plate spacing, radial distance from stagnation point, Prandtl number, target plate inclination, confinement of the jet, nozzle geometry, curvature of target plate, roughness of the target plate and turbulence intensity at the nozzle exit. A large number of investigations have been carried out in the area of jet impingement heat transfer over the years. Lytle and Webb [3] studied the effect of very low nozzle-to-plate spacing ($L/d < 1$) on the local heat transfer distribution on a flat plate impinged by a circular air jet issued by long pipe nozzle which allows for fully developed flow at the nozzle exit and found that in the acceleration range of the nozzle plate spacing ($L / d < 0.25$), maximum Nusselt number shifts from the stagnation point to the point of secondary peak and the effect being more pronounced at higher Reynolds number. Lytle and Webb [4] focused their study on low nozzle-to-plate spacing, and measured the heat transfer coefficient for distance of 6 diameters. Lee et al. [5] studied the effect of nozzle diameter on impinging jet heat transfer and fluid flow. Anwarullah et al [6] presented experimental study of the effect of nozzle-to-surface spacing of the electronic components and Reynolds number on the heat transfer in cooling of electronic components by an impinging submerged air jet. The results are expected to help the designers in coming up with more effective designs for cooling of electronic components $Nu_{Cor} = 0.8\ (Red)^{0.5}\ (Pr)^{0.36}\ (H/d)^{-0.06}$. Vadiraj and Prabhu [7] conducted theoretical and experimental investigations for the local heat transfer distribution between smooth flat surface and impinging air jet from a circular nozzle. They found that an increase in Reynolds number increases the heat transfer at all radial locations for a given radial distance. Amy and Sharareh [8] conducted an experimental study on the influence of a protruding pedestal on a single circular impinging air jet on heat transfer rate with different Reynolds numbers, jet exit diameters and jet exit-to-surface distances. They found that, at constant Reynolds numbers, the Nusselt number increases due to increase in jet diameters. Puneet et al [9] performed an experimental investigation to study the effects of the shape of the nozzle, jet-to-plate spacing and Reynolds number on the local heat transfer distribution to normally impinging submerged air jet on smooth and flat surface. Jeevanlal and Kumar [1] presented an experimentally work to investigate the heat transfer on a heated plate impinged with cold air jet. Varied the nozzle exit to target plate distance (Z/D) and varied jet velocity for obtaining six Reynolds numbers. Detailed reviews on jet impingement cooling are presented by Jambunathan et al. [10], Viskanta [11] and Zuckerman and Lior [12]. These reviews are mainly focused on the jet impingement cooling of flat surfaces. Hollworth and Durbin [13] investigated the impingement cooling of electronics, Roy et al. [14] investigated the jet impingement heat transfer on the inside of a vehicle windscreen and Babic et al. [15] used jet impingement for the cooling of a grinding process.

In this paper, the lack of three dimensional numerical solution lead to study the cooling of heated plate by a jet numerically. Finite volume will be used to study the jet domain while finite difference will be used to calculate the temperature of the plate. Different parameters will be studied, Reynolds number based on nozzle exit condition will vary between 1000 and 4000 and heat flux from 500 to 1000 W.

2.Numerical Formulations

This study focuses on a square air jet impinging at heated plate (30 cm x 03 cm x 30 cm), as shown in Figure 1. The plate is subjected to a constant heat flux from below.

For the jet domain (30 cm x 30 cm x 30 cm), the flow is considered to be incompressible, steady and three dimensional. The distance between the nozzle exit and the plate surface is z, the flow within the inducer is assumed to be steady incompressible three dimensional laminar flow. The governing equations for three dimensional, steady, and laminar flow are represented by continuity, momentum and energy equation as follow:

Continuity:

$$\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}+\frac{\partial(\rho w)}{\partial z}=0 \tag{1}$$

x-direction momentum equation

$$\frac{\partial(\rho uu)}{\partial x}+\frac{\partial(\rho vu)}{\partial y}+\frac{\partial(\rho wu)}{\partial z}=\frac{\partial}{\partial x}\left[\mu\left(\frac{\partial u}{\partial x}\right)\right]+\frac{\partial}{\partial y}\left[\mu\left(\frac{\partial u}{\partial y}\right)\right]+\frac{\partial}{\partial z}\left[\mu\left(\frac{\partial u}{\partial z}\right)\right]+S_u \tag{2}$$

y-direction momentum equation

$$\frac{\partial(\rho uv)}{\partial x}+\frac{\partial(\rho vv)}{\partial y}+\frac{\partial(\rho wv)}{\partial z}=\frac{\partial}{\partial x}\left[\mu\left(\frac{\partial v}{\partial x}\right)\right]+\frac{\partial}{\partial y}\left[\mu\left(\frac{\partial v}{\partial y}\right)\right]+\frac{\partial}{\partial z}\left[\mu\left(\frac{\partial v}{\partial z}\right)\right]+S_v \tag{3}$$

z-direction momentum equation

$$\frac{\partial(\rho uw)}{\partial x}+\frac{\partial(\rho vw)}{\partial y}+\frac{\partial(\rho ww)}{\partial z}=\frac{\partial}{\partial x}\left[\mu\left(\frac{\partial w}{\partial x}\right)\right]+\frac{\partial}{\partial y}\left[\mu\left(\frac{\partial w}{\partial y}\right)\right]$$
$$+\frac{\partial}{\partial z}\left[\mu\left(\frac{\partial w}{\partial z}\right)\right]+S_w \qquad (4)$$

Energy:

$$\frac{\partial(\rho uT)}{\partial x}+\frac{\partial(\rho vT)}{\partial y}+\frac{\partial(\rho wT)}{\partial z}=\frac{\partial}{\partial x}\left[\Gamma\left(\frac{\partial T}{\partial x}\right)\right]+\frac{\partial}{\partial y}\left[\Gamma\left(\frac{\partial T}{\partial y}\right)\right]$$
$$+\frac{\partial}{\partial z}\left[\Gamma\left(\frac{\partial T}{\partial z}\right)\right]+S_T \qquad (5)$$

Where,

$$S_u=-\frac{\partial P}{\partial x} \qquad (6)$$

$$S_v=-\frac{\partial P}{\partial y} \qquad (7)$$

$$S_w=-\frac{\partial P}{\partial z} \qquad (8)$$

In thermal energy equation the radiation heat transfer, the viscous dissipation, pressure work and Joule heating are ignored, so the source term S_T becomes:

$$S_T=0 \qquad (9)$$

Boundary conditions:

Only quarter of the jet domain will be studied, so the boundary conditions are as expressed as:

- At the right and left sides, $\frac{\partial \phi}{\partial z}=0$

- At the front and backward sides, $\frac{\partial \phi}{\partial x} = 0$

- At top $y = L;\ v = v_{in},\ T = T_{in}$

- At bottom $y = 0;\ u = 0,\ v = 0,\ w = 0,\ T = T_s$

The SIMPLE algorithm by Patankar and Spalding [16,17] is applied to solve the conservation equation of mass, momentum and energy by Finite volume. The number of mesh was 24 x 24x 24.

For iron plate with conductivity 45 W/m C, Laplace equation was used to calculate the temperature in the plate under the effect of heat flux from below (500, 750, 1000 W/m^2).

$$\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} + \frac{\partial^2 T}{\partial z^2} = 0 \qquad (6)$$

The Finite difference was used to find the temperature of the plate. A subroutine for solving Laplace equation was inserted in main program, so in each iteration of the main program, the temperature of the surface of the plate was calculated and became boundary condition of the temperature at y = 0 for the main program. The main problem in the calculation is the value of the heat transfer coefficient between the plate and the jet domain. Ref [18] empirical formula was used:

$$Nu = 1.16\, Re^{0.447} Pr^{0.333} \qquad (7)$$

Where,

$$Re = \frac{\rho\, v_{in} d_h}{\mu},\ Nu = \frac{h\, d_h}{k}$$

d_h hydraulic diameter based on inlet jet area. A program in Fortran 90 was built to solve the problem and the figures were plotted by software Tecplot7.

3.Results and discussion

The results from fluid flow and heat transfer calculation of an impinging air jet are presented and analyzed in this section. The fluid flow includes calculating three components of the fluid velocity in a jet impingement domain. The heat transfer includes calculating temperatures in the jet domain and in heated plate. As detailed before, there are many parameters that affect the heat transfer to an impinging jet including heat flux and jet exit Reynolds number (Re). Results are presented for three values of the heat flux (500, 750, 1000 W/m^2) and four Reynolds numbers (Re = 1000, 2000, 3000, 4000). Figure 2 shows isotherm contours for jet domain for different Reynolds with q = 1000 W/m^2. High Reynolds number comes from high inlet velocity or high inlet area. High Reynolds number forces the gradient in thermal field to become close to the surface of the plate. Reynolds 1000 plot shows higher temperature gradient field while Reynolds 4000 shows lower temperature gradient field. Figure 3 shows isotherm contours for heated plate for different Re with q = 1000 W/m^2. High Reynolds number shows lower temperature while low Reynolds number shows higher temperature. The temperatures of different points in the heated plate were very close because of the conduction effect. The differences in temperatures were not more than 2 degrees. Figure 4 represents vector field in three planes for selective planes for Re = 1000 and q = 1000 W/m^2. Three selective planes were chosen to shows the existing main vector field. In the first and second plots, the main existing vector field was the normal components, v, while the third plot shows the x and z components, u and w because the v components turn into u and w components as the jet hits the plate. Figure 5 shows the average plate temperature variations with heat flux for different Reynolds number. The average temperature of the plate decreases with Reynolds increases with different values of the heat fluxes. Figure 6 shows the variation of the average plate temperature with heat fluxes for different Reynolds number. The temperature of the heated plate increases with heat flux increases. Same behavior was observed for different Reynolds numbers.

4.Conclusions

- The temperature of heated plate was decided by heat flux and Reynolds number.
- The temperature of the heated plate increases with increases of heat flux and decreases with increases of Reynolds number.
- The whole points in the heated plate had close temperature according to conduction effect.
- The impingement area of the plate had lower temperature relative to other points in the plate.
- The thickness of the thermal boundary layer decreases with Reynolds number increases.
- The effect of plate temperature with Reynolds number decreases as Reynolds number increases.

5.Future Works

- Study the effect jet angle.
- Non-uniform heating of the heated flux.
- Turbulent jet impingement.

References

[1] V. M .Jeevanlal , B.C. Anil Kumar, "Experimental Investigation of Heat Transfer from a Flat and Surface Indented Plate Impinged with Cold Air Jet- Using Circular Nozzle" International Journal of Emerging Engineering Research and Technology Volume 2, Issue 5, August 2014, PP 160-170.

[2] Thakare S. S. and Joshi J. B. " CFD modeling of heat transfer in turbulence pipe flows" AIche Journal, 2000, Vol.46, pp1798-1812

[3] D. Lytle, B.W. Webb “Air jet impingement heat transfer at low nozzle plate spacing’s” Int. J. Heat Mass Transfer 37 (1994) 1687–1697.

[4] Lytle, D., Webb, B. W., Air jet impingement heat transfer at low nozzle plate spacing, Int. J. Heat and Mass Transfer, , (1994), 37., 1687– 1697.

[5] D.H. Lee, J. Song, C.J. Myeong, “The effect of nozzle diameter on impinging jet heat transfer and fluid flow”, J. Heat Transfer 126 (2004) 554–557.

[6] M. Anwarullah, V. Vasudeva Rao , K.V. Sharma "Effect of Nozzle Spacing on Heat Transfer and Fluid Flow Characteristics of an Impinging Circular Jet in Cooling of Electronic Components", Int. J. of Thermal & Environmental Engineering, 2012 Volume 4, No. 1, pp7-12.

[7] Vadiraj Katti, Prabhu S. V., “Experimental study and theoretical analysis of local heat transfer distribution between smooth flat surface and impinging air jet from a circular straight pipe nozzle”, Int. J. Heat Mass Transfer, (2008), 51, 4480–4495.

[8] Amy S. Fleischer., Sharareh R. Nejad, “Jet impingement cooling of a discretely heated portion of a protruding pedestal with a single round air jet”, Experimental Thermal and Fluid Science, (2004), 28, 893– 901.

[9] Puneet Gulati, Vadiraj Katti and S. V. Prabhu," Influence of the shape of the nozzle on local heat transfer distribution between smooth flat surface and impinging air jet" International Journal of Thermal Sciences 48 (2009) 602–617

[10] R. Viskanta, Heat transfer to impinging isothermal gas and flame jets, Experimental Thermal and Fluid Science 6 (1993) 111–134.

[11] N. Zuckerman, N. Lior, Jet impingement heat transfer physics, correlations and numerical modeling, Advances in Heat Transfer 39 (2006) 565–632.

[12] F. Gori, L. Bossi, “On the cooling effect of an air jet along the surface of a cylinder”, International Communications in Heat and Mass Transfer 27 (2000) 667–676.

[13] B. R. Hollworth, M. Durbin, “Impingement cooling of electronics”, ASME J. Heat Transfer 114 (1992) 607–613.
[14] S. Roy, K. Nasr, P. Patel, B. AbdulNour, An experimental and numerical study of heat transfer off an inclined surface subject to an impinging airflow, Int. J. Heat Mass Transfer 45 (2002) 1615–1629.
[15] D. M. Babic, D. B. Murray, A.A. Torrance, “Mist jet cooling of grinding processes”, Int. J. Mach. Tools Manuf. 45 (2005) 1171–1177.
[16] Patankar, S.V. and Spalding, D.B., " A calculation procedure for heat, mass and momentum transfer in three-dimensional parabolic flows." Int. J. Heat Mass Transfer, Vol. 3, 269-289, 1974.
[17] Patankar, S. V., "Numerical Heat Transfer and Fluid Flow", McGraw-Hill, New York (1980).
[18] John N. B. and Peter Hrycak, “Impingement heat transfer from turbulent air jets to flat plates”, NASA TM x-2778., 1973

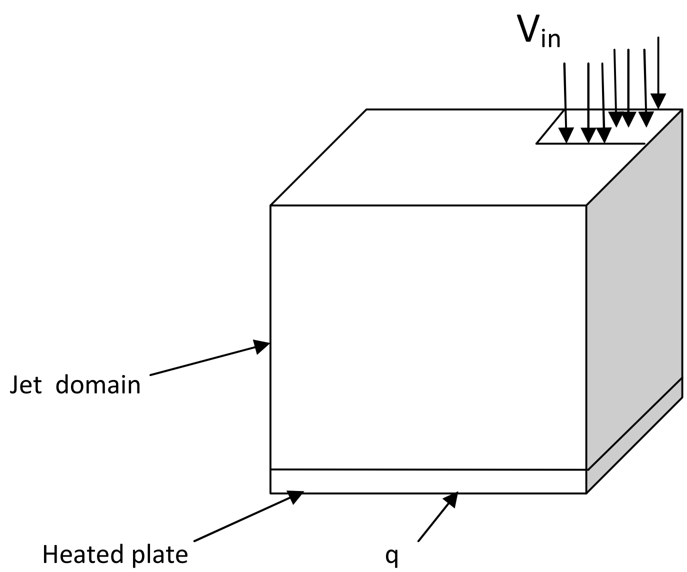
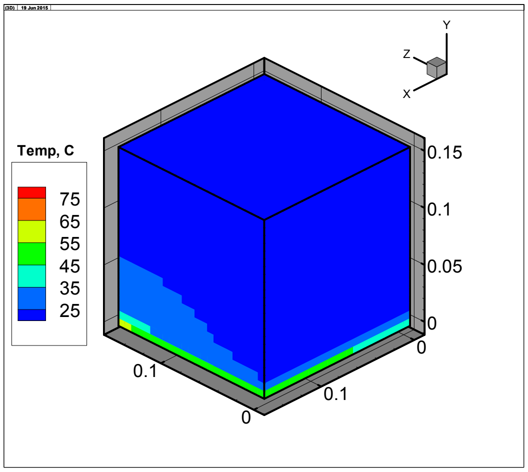
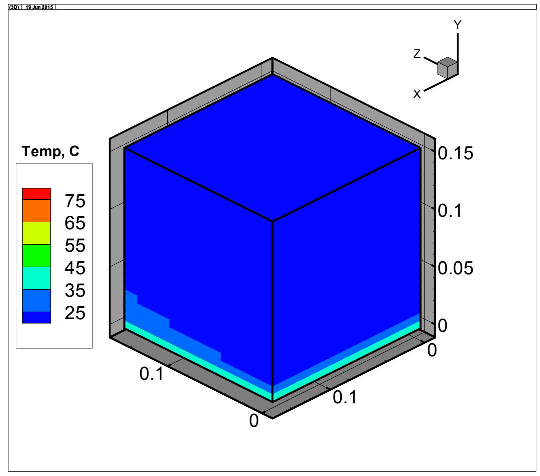


Fig. 1 Schematic diagram of the problem



Re = 1000



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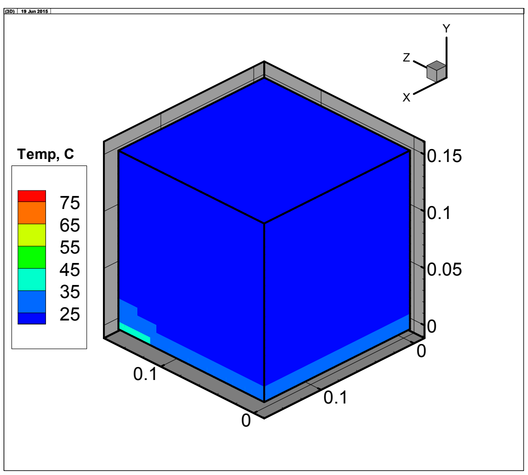
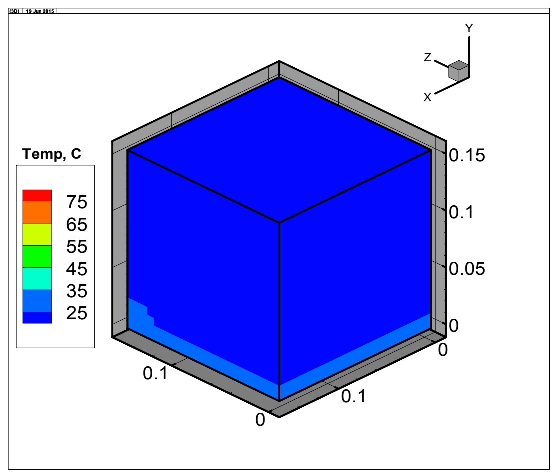
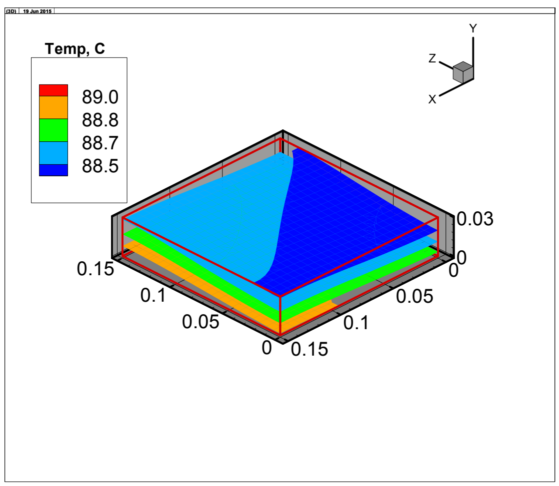


Re=3000

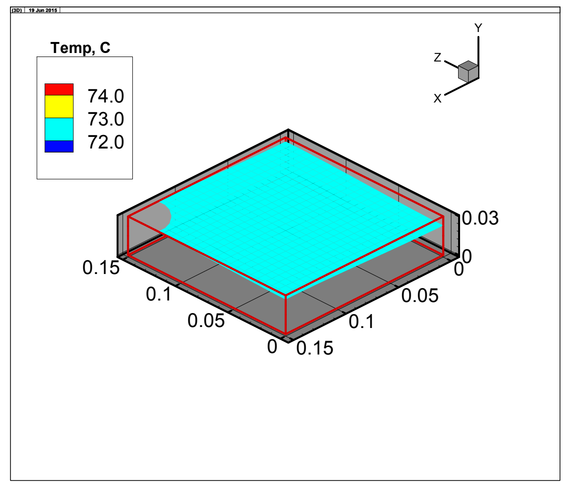


Re=4000

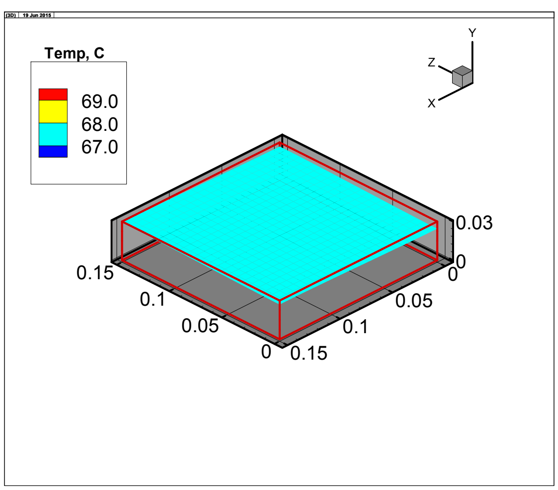
Fig.2 Isotherms for jet domain for different Reynolds number with q = 1000 W/m^2



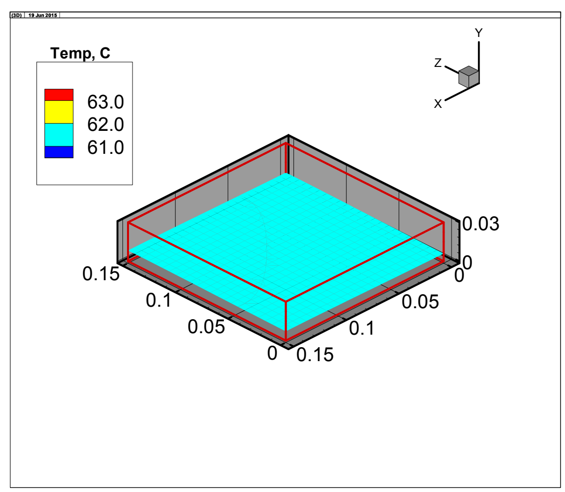
Re = 1000



Re = 2000



Re = 3000



Re = 4000

Fig. 3 Isotherm contours for heated plate for different Reynolds number with q = 1000 W/m^2

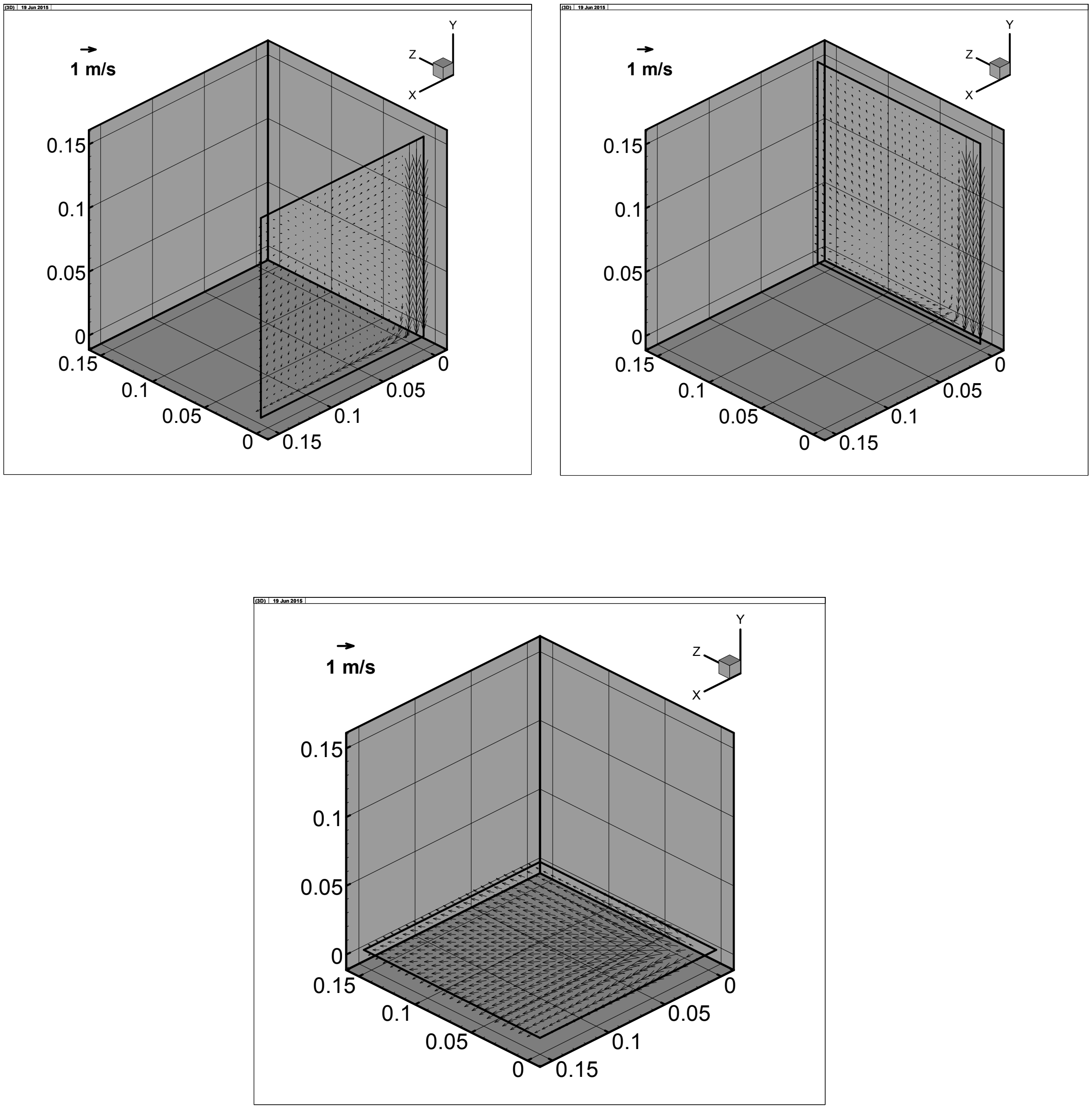


Fig. 4 Vector field in three planes for selective planes for Re = 1000 and $q = 1000\ W/m^2$

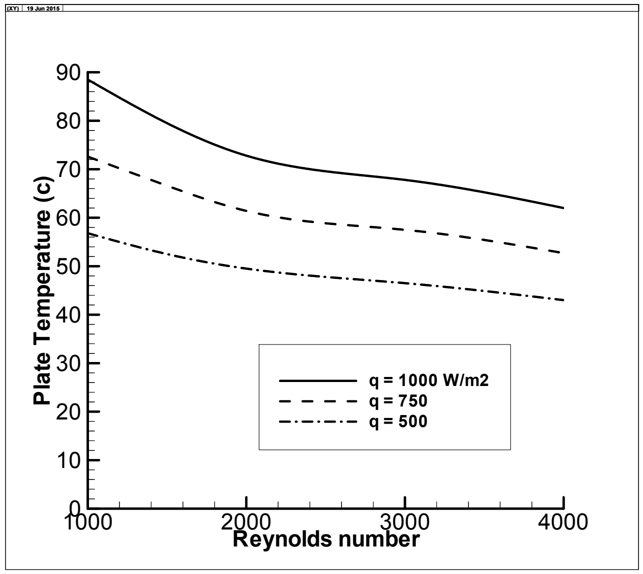


Fig. 5 Plate temperature variations with Reynolds numbers for different heat flux.

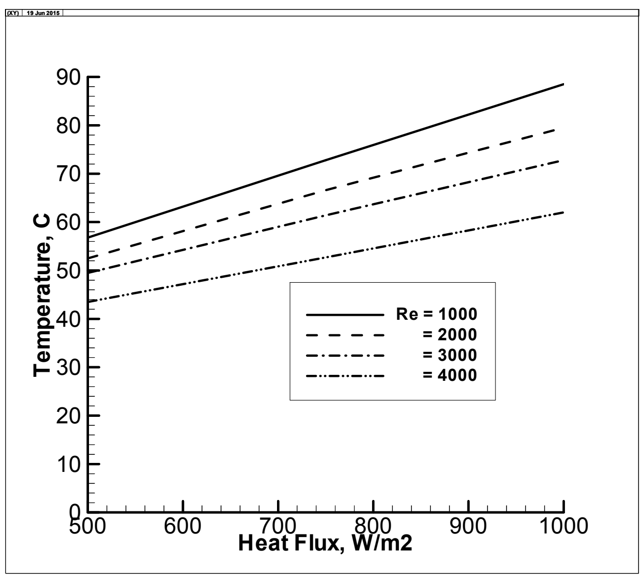


Fig. 6 Plate temperature variations with heat flux for different Reynolds number.

فحص رقمي ثلاثي الابعاد لتاثير اصطدام نفث طباقي على سطح ساخن

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المستخلص

انجز بحث رقمي ثلاثي الابعاد لدراسة تاثير الفيض الحراري ورقم رينولدز على توزيع انتقال حرارة موضعي من اصطدام هواء النفث مع سطح لوح املس ، استناداً الى تغير حالة خروج الهواء من فوهة النفاث اعتمد رقم رينولد مابين 1000 – 4000 رينولد وانبعاث حراري مابين 500 - 1000 w/m^2.

دُرس حيز التدفق بحل معادلات نافيير ستوكس بطريقة الحجم المتناهى بينما درس الانتقال الحراري على السطح عن طريق حل معادلة لابلاس وعُبرعن النتائج على شكل خطوط تدفق ايزوثرمية.

أضهرت النتائج ازدياد في درجة حرارة سطح اللوح مع ازدياد الفيض الحراري . واتضح انخفاض بدرجة حرارة سطح اللوح وسمك الطبقة الحرارية المتاخمة مع ازدياد عدد رينولد.

الكلمات المفتاحية: الفيض الحراري، رقم رينولد, معادلة نافيير ستوكس, الحجم المتناهي

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