

EXPERIMENTAL AND NUMERICAL INVESTIGATION ON THE EFFECTS OF JET REYNOLDS NUMBER AND NOZZLE-PLATE SPACINGS ON SINGLE ROUND AIR JET IMPINGEMENT HEAT TRANSFER

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Abstract

Jet impingement is an effective heat transfer method which has been used in various industrial applications. In this study, the effect of jet Reynolds number and nozzle-plate spacing on the single round air jet impingement heat transfer rates is studied both experimentally as well as numerically. The computational study was performed using OpenFOAM and validated it against the experimental results. Single jet impingement heat transfer produced a non uniform cooling of the heating plate. The heat transfer rates were found to be increased when the Reynolds number of the jet was varied from 8000 to 24000. Increase in H/D ratio from 1 to 3 resulted in increased heat transfer rates expect for the jet Reynolds number equal to 8000. At lower jet Reynolds number, the maximum heat transfer rate was found to occur when the heating plate was closer to the orifice plate. The maximum average Nusselt number was observed on the heating plate for H/D ratio 3 and jet Reynolds number equal to 24000 whereas for H/D ratio 2 and jet Reynolds number equal to 8000 the minimum average surface Nusselt number was obtained.

Keywords: Jet impingement, jet Reynolds number, nozzle-plate spacing, heat transfer, OpenFOAM

Nomenclature

A	Area of the heating plate (m^2)
D	Diameter of the jet (m)
H	Distance between orifice plate and heating plate (m)
h	Local heat transfer coefficient on the heating plate ($\text{W}/\text{m}^2\text{K}$)
\bar{h}	Average heat transfer coefficient on the heating plate ($\text{W}/\text{m}^2\text{K}$)
k	Thermal conductivity of air (W/mK)
Nu	Local Nusselt number on the heating plate
\overline{Nu}	Average Nusselt number on the heating plate

Q_w	Total heat flux generated by the heater (W/m ²)
Re	Jet Reynolds number
T	Temperature (K)
T_w	Local temperature on the heating plate (K)
T_j	Inlet temperature of the air jet (K)
x	Distance along the heating plate (m)
ρ	Density of air (kg/m ³)
μ	Dynamic viscosity of air (kg/ms)

1. Introduction

Technology is developing day by day. This is the era of miniaturization. Computers have changed from room size to laptops and palmtops. Not only computers, manufacturers are trying to reduce the size of each and every electronic device with the help of semi conductors, IC chips, etc. Better performance and effective life of such devices can be achieved only by efficient removal of the generated heat. Hence cooling of electronic devices is an active research topic. Many researchers have come up with different cooling technologies among which jet impingement cooling has received considerable attention due to their inherent characteristics of simple geometry and higher rates of heat transfer. As an effective and flexible means of high heat/mass transfer, jet impingement has been widely used in many industrial applications, which include cooling of gas turbines, annealing of metals, processing of food and so on.

In this study the effect of jet Reynolds number and nozzle-plate spacing on single round jet impingement heat transfer is studied. A steady, incompressible, air jet at 306 K is being issued through a 10 mm diameter orifice plate attached at the bottom of a plenum chamber on to a 120 mm * 120 mm flat heat plate which is placed at a certain distance from the orifice plate. It is required to cool the heat plate which is producing a uniform heat flux of 3472.22 W/m². The jet Reynolds number chosen for the study are 8000, 16000 and 24000 where as the H/D ratio was taken as 1, 2 and 3.

2. Experimental setup and procedures

The photograph of the experimental facility is shown in Fig. 1. Atmospheric air is drawn by a centrifugal blower of 3HP capacity and is preheated to a constant temperature of 306 K. The air flow is regulated by the valve and is measured by an orifice meter. After that, it enters a plenum chamber. The turbulent motion of air is damped by honeycomb meshes set inside the chamber and it also ensures a uniform velocity distribution. Air jet is issued through an orifice plate having a hole diameter and thickness of 10 mm which is attached at the bottom of the chamber.

A 120 * 120 mm test section made of Aluminum is mounted on an X-Y traverser below the orifice plate. Uniform heat flux is provided to the test section by attaching a foil type heater at its bottom side having a power rating of 50 W. All sides of the test section except the top side were properly insulated to minimize the possible heat losses to the surroundings.



Fig. 1 Jet impingement test facility

K-type thermocouples were attached at different locations of the test section as shown in Fig. 2. The measured temperatures were recorded using a personal computer through a Data Acquisition (DAQ) system. The thermocouples, DAQ, U tube manometer and preheater were NABL calibrated.

During the experiment, the exit velocity of the jet was kept as 39 m/s which corresponds to a jet Reynolds number of 24000. The uncertainty in the jet velocity is estimated to be $\pm 10.6\%$. The

test section was kept at a distance of 20 mm from the orifice plate which results in the H/D ratio to be 2.

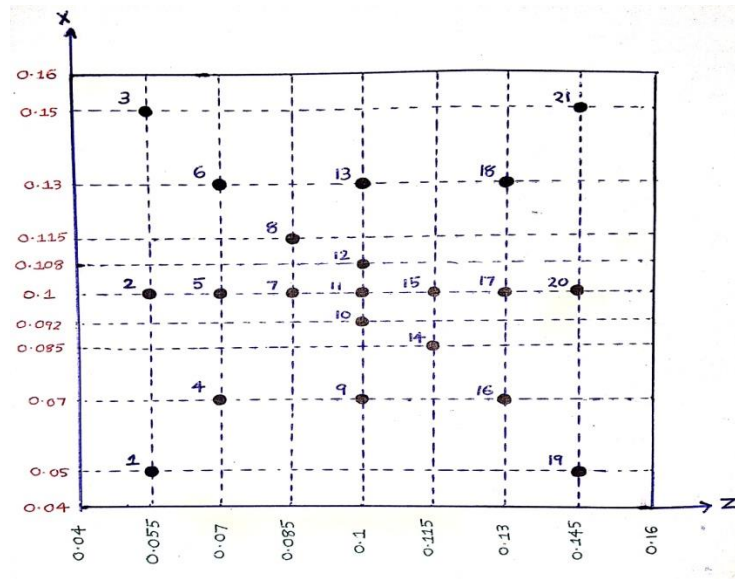


Fig. 2 Schematic representation of the thermocouple locations on the heating plate

The centrifugal blower was turned on and allowed to run for 20 minutes till the flow became steady. Then the valves were adjusted for obtaining the required flow towards the plenum chamber. The heating plate was turned on and the air jet was made to impinge on it for about 20 minutes till steady flow conditions were obtained. Temperatures at different locations of the heating plate were recorded by the DAQ at a rate of one reading per second and were continuously monitored using a personal computer. After about 15 minutes, the temperature readings became steady. It was continued to run for 5 more minutes and the average of the steady state temperature readings were taken for each thermocouple location.

3. Numerical approach

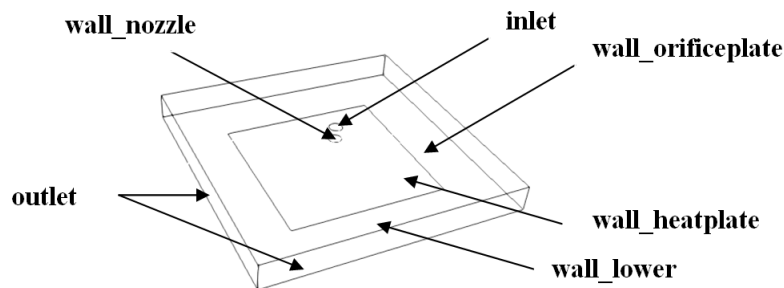


Fig. 3 Wire frame model of the computational domain

The wire frame representation of the computational domain modeled using the open source 3D modeling software FreeCAD is shown in Fig. 3. The different named sections of the geometry were exported in ASCII format as separate ".stl" files keeping the units in meters. Meshing was done using the snappyHexMesh utility of OpenFOAM. Fig. 4 represents the computational domain after meshing.

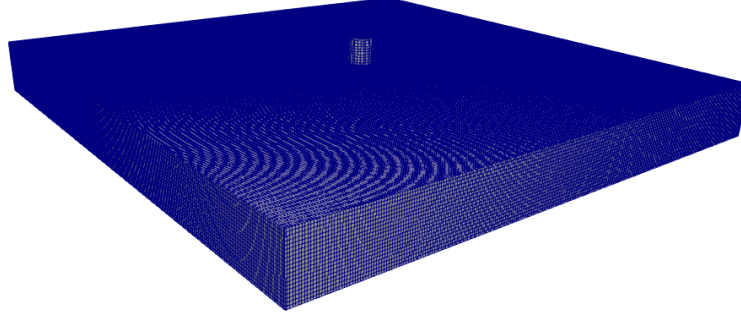


Fig. 4 Meshed model of the computational domain

Air flow was assumed to be steady, turbulent and incompressible. The effect of gravity, viscous dissipation and radiation were neglected. Thermo-physical properties of air were assumed to be constant. Computational studies were performed using the bouyantBoussinesqSimpleFoam solver which is a steady-state heat transfer solver for buoyant, turbulent flow of incompressible fluids in OpenFOAM. Table 1 shows the different boundary conditions imposed on the computational domain.

Named sections	k	omega	Pressure	Velocity	Temperature
<i>inlet</i>	fixedValue value uniform 0.085	fixedValue value uniform 4.7999	zeroGradient	fixedValue value uniform (0 -39 0) or (0 -26 0) or (0 -13 0)	fixedValue value uniform 306
<i>outlet</i>	zeroGradient	zeroGradient	fixedValue value uniform 0	zeroGradient	fixedValue value uniform 306
<i>wall_heatplate</i>	kqRWallFunction	omegaWallFunction	zeroGradient	fixedValue value uniform 0	fixedGradient gradient uniform 128634.1726
<i>wall_lower</i>	kqRWallFunction	omegaWallFunction	zeroGradient	fixedValue value uniform 0	zeroGradient

<i>wall_orificeplate</i>	kqRWallFunction	omegaWallFunction	zeroGradient	fixedValue value uniform 0	zeroGradient
<i>wall_nozzle</i>	kqRWallFunction	omegaWallFunction	zeroGradient	fixedValue value uniform 0	zeroGradient

Table 1. Important boundary conditions

According to convection boundary condition, heat flux, $q'' = k(\nabla T)$.

$$\text{So, } \nabla T = \frac{q''}{k} = \frac{3472.22}{0.026993} = 128634.1726$$

This is how the gradient value for uniform heat flux boundary condition was calculated.

k - omega SST is the best turbulence model for jet impingement heat transfer computational studies [1]. Hence this model was chosen as the turbulence model for the present study. The turbulence intensity was fixed as 5% for all cases. Convergence is assumed to be reached when the residuals for pressure and temperature falls below 10^{-2} as well as the residuals for the different turbulence terms (k, omega) and velocity falls below 10^{-3} and 10^{-4} respectively.

Reynolds number of the jet is defined based on the diameter of the jet:

$$Re = \frac{\rho v D}{\mu}$$

Jet Reynolds number 8000, 16000 and 24000 corresponds to an inlet jet velocity of 13 m/s, 26 m/s and 49 m/s respectively.

The local Nusselt number on the heating plate is defined based on the diameter of the jet:

$$Nu = \frac{hD}{k}$$

where the local heat transfer coefficient is calculated as:

$$h = \frac{Q_w}{T_w - T_j}$$

where Q_w is the total heat flux generated by the heater which is equal to 3472.22 W/m², T_w is the heating plate temperature and T_j is the inlet jet temperature.

The average Nusselt number on the heating plate is defined as:

$$Nu = \frac{\bar{h}D}{k}$$

where the average heat transfer coefficient is calculated as:

$$\bar{h} = \frac{1}{A} \iint h dA$$

where A is the area of the heating plate which is equal to 0.0144 m².

Post processing was done using ParaView software. In order to evaluate the surface integrals, the "wall_heatplate" surface was extracted using *Extract Surface filter* and then the local heat transfer coefficient values over the extracted surface was calculated using the *calculator tool*. Using the *integrate variables filter* available in ParaView, the expression for calculating local heat transfer coefficients was integrated throughout the surface domain of the heating plate.

4. Grid independency study and validation

Grid independency study was carried out by varying the number of elements from 225360 to 1801720 and the average heat transfer coefficient on the heating plate was monitored in each case as shown in Fig. 5. The jet Reynolds number and H/D were set as 24000 and 2 respectively. 1261230 elements was found to be the optimum number of elements for the present study from the view point of accuracy and computational study.

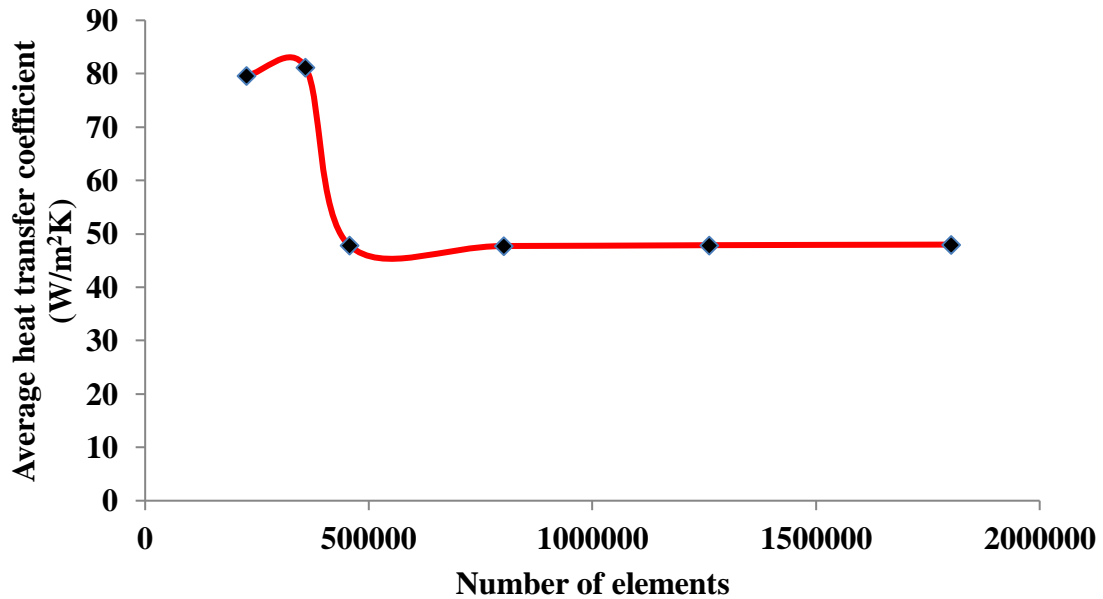


Fig. 5 Grid independency test

Numerically predicted temperature values at different locations on the heating plate were compared with the experimental results as shown in Fig. 6. It is observed that the numerical prediction is in good agreement with the experimental values with a maximum percentage variation of 0.3%.

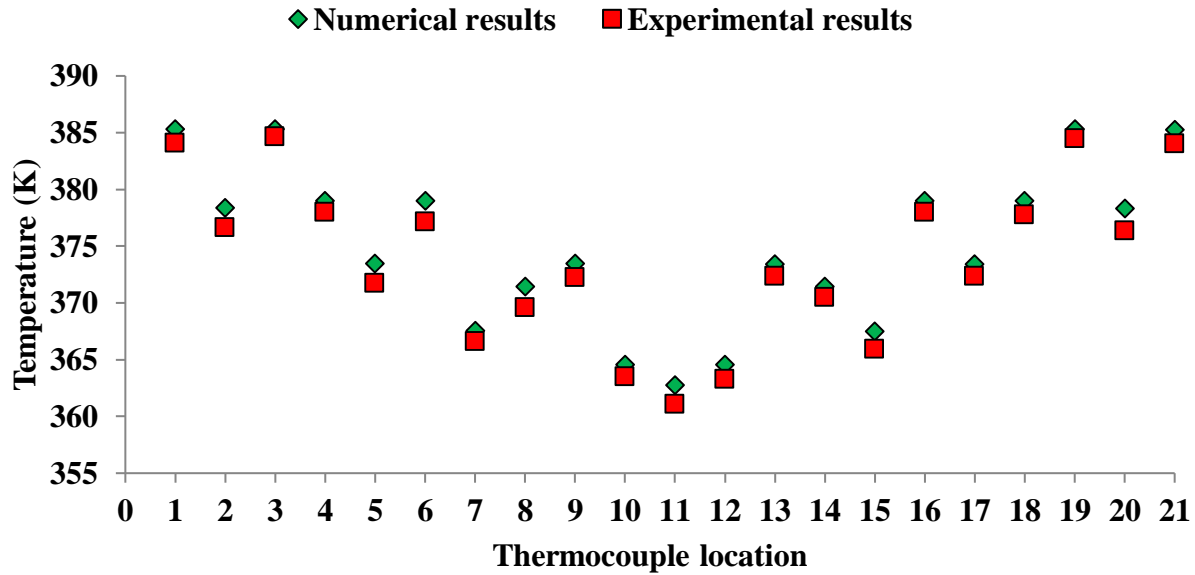


Fig. 6 Comparison of experimental and numerical values of temperatures at different thermocouple locations

5. Results and discussions

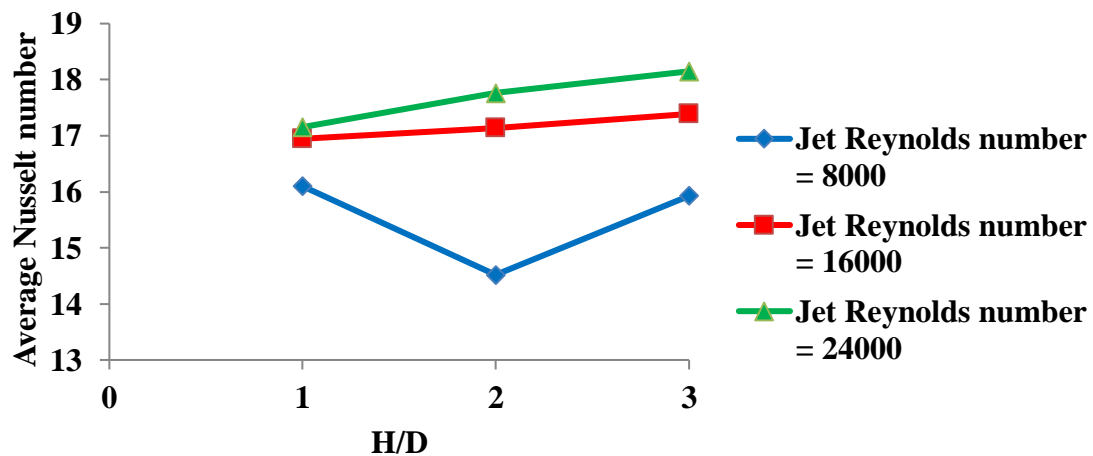


Fig. 7 Effect of variation of H/D ratio on average surface Nusselt number at different jet Reynolds numbers

Fig. 7 shows the variation in average Nusselt number on the heating plate for different H/D ratios at different jet Reynolds numbers. It is observed that with increase in the jet velocity, average surface Nusselt number is also increased. Increase in H/D ratio produces increased heat transfer rates expect for the jet Reynolds number equal to 8000. At lower jet Reynolds number, the maximum heat transfer rate occurs when the heating plate is closer to the orifice plate. Increase in the spacing between the orifice plate and heating plate first decreases and then increases the average surface Nusselt number. This may be due to the decrease in the momentum with which jet strikes on the heating plate at larger spacing and lower jet velocities.

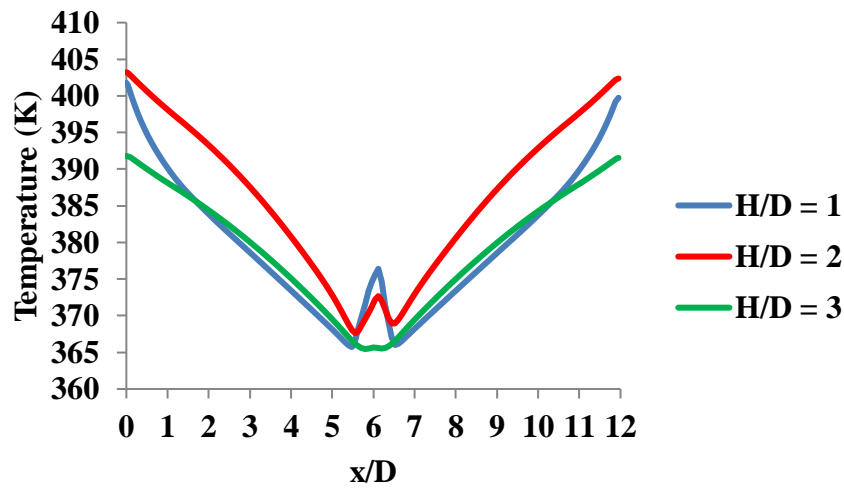


Fig. 8 Temperature distribution along the centre line of the heating plate at jet Reynolds number 8000

Figures 8, 9 and 10 shows the variation in temperature along the centre line of the heating plate for different H/D ratios at different jet Reynolds numbers. The location of minimum temperature coincides with the stagnation zone of the jet. The region where the jet impinges on the heating plate and deflects towards both sides is known as the stagnation zone. Due to increased jet momentum, heat transfer rate will be maximum in this zone. As we proceed towards both sides of the stagnation zone the heat transfer rates are found to be monotonically decreasing due to the diminishing jet momentum which may also result in the separation of flow from the surface of the heating plate. At jet Reynolds number of 8000 a peak in the temperature distribution is observed at the stagnation zone. That means the heat transfer rate at the location where the jet impinges on the heating plate is lower compared to the surrounding region. This is due to the occurrence of toroidal vortices around the jet. As the air jet is issued from the nozzle toroidal

vortices are produced circumferentially around the jet due to the shear interaction of the jet with atmospheric air. Such vortices absorb the kinetic energy of the jet from its central region and convert it into rotational energy. At low jet velocities, this may result in an overall decrease in jet momentum as it strikes the heating plate compared with higher jet velocities. This produces an increase in temperature at the stagnation zone whereas in the immediate vicinity of this zone where the toroidal vortices impinge on the heating plate a dip in the temperature distribution is observed.

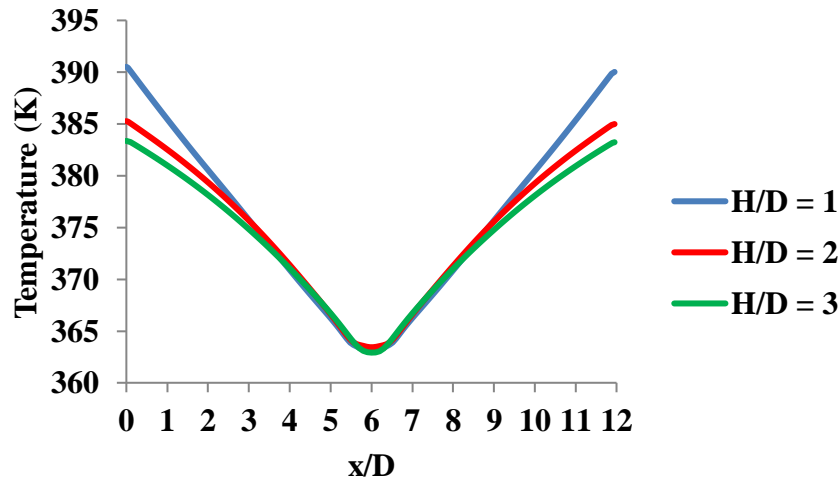


Fig. 9 Temperature distribution along the centre line of the heating plate at jet Reynolds number 16000

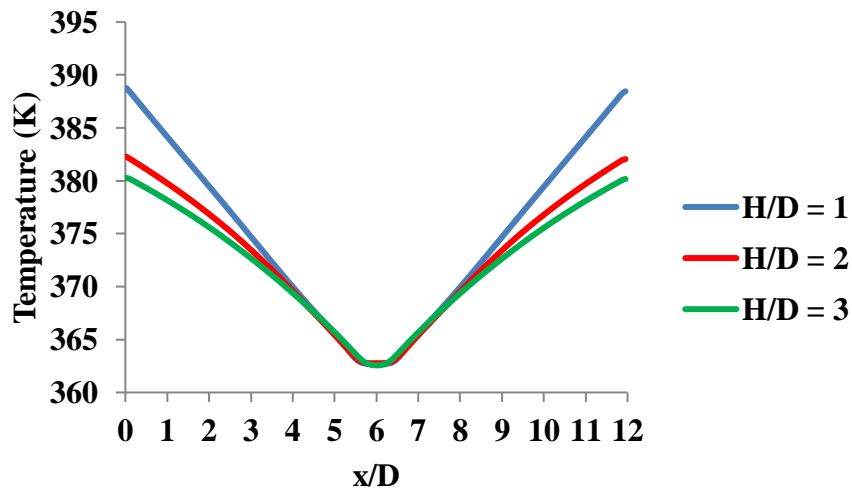


Fig. 10 Temperature distribution along the centre line of the heating plate at jet Reynolds number 24000

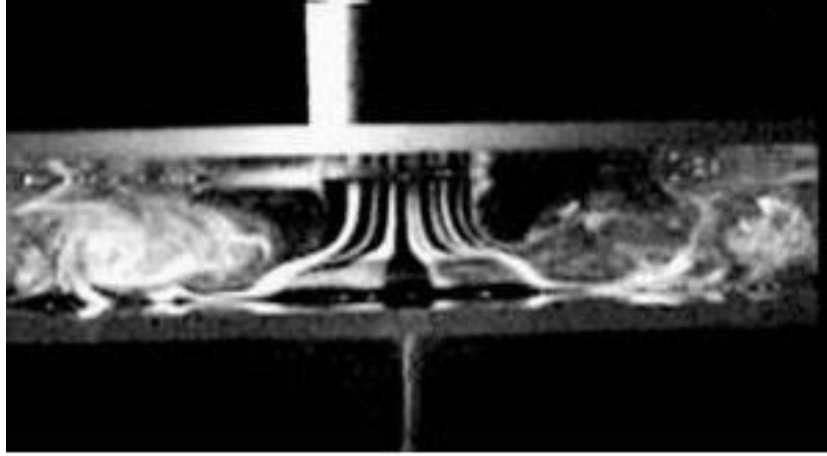


Fig. 11 Flow visualization image of an impinging air jet from an orifice plate attached to a plenum chamber [2]

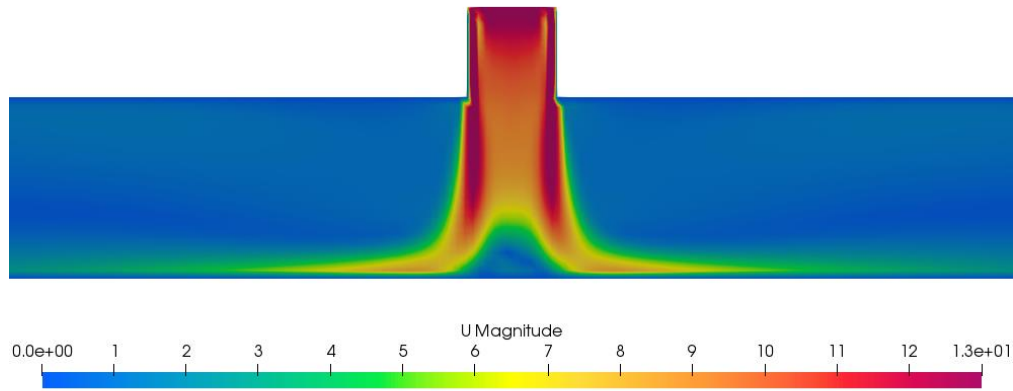


Fig. 12 Velocity contour along the central plane of the computational domain corresponding to H/D ratio 2 and jet Reynolds number 8000

Figures 11 and 12 shows a comparison between the velocity contour over the central plane of the computational domain and the flow visualization images of an air jet at lower jet Reynolds number. The formation of toroidal vortices and the decrease in momentum with which the jet strikes on the heating plate is clear from both figures.

Temperature contours on the heating plate for the cases in which maximum and minimum average surface Nusselt numbers are obtained is compared in Fig. 13. For H/D ratio 3 and jet Reynolds number equal to 24000, the maximum average Nusselt number (value equal to 18.15)

is observed on the heating plate whereas for H/D ratio 2 and jet Reynolds number equal to 8000, the minimum average surface Nusselt number (value equal to 14.52) is observed. From Fig. 13 it is noticed that single round air jet impingement produces a non uniform temperature distribution on the heating plate. This may produce thermal stresses and it may be treated as the most significant disadvantage of single jet impingement cooling.

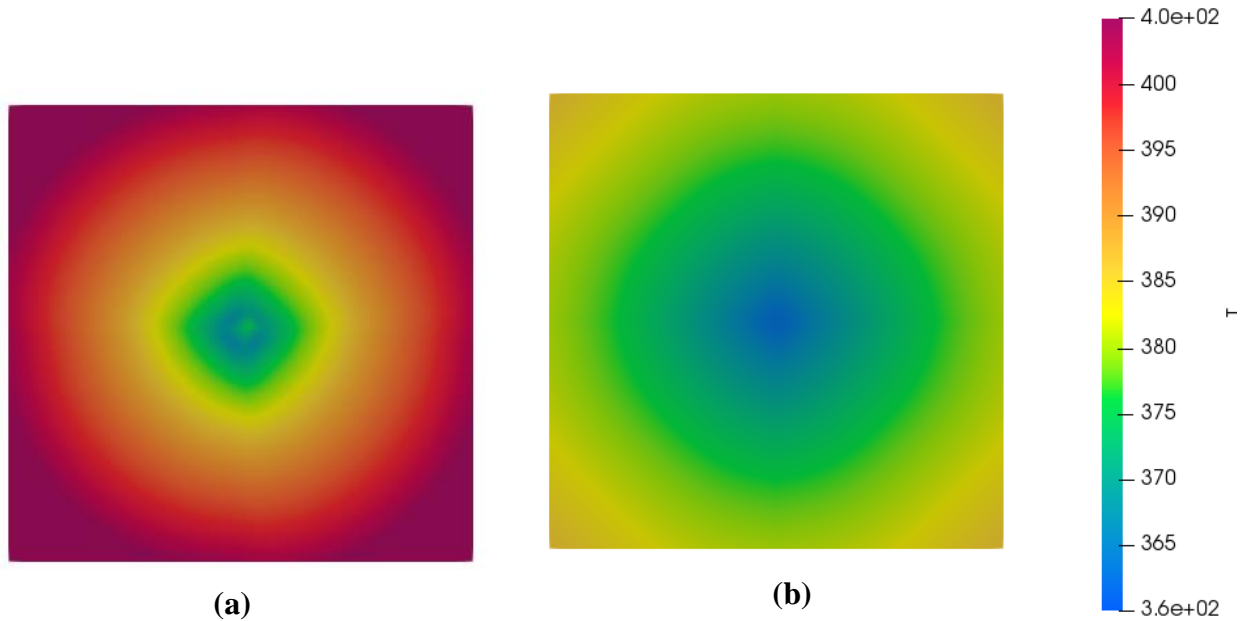


Fig. 13 Temperature contour on the heating plate for *(a)* H/D ratio 2, jet Reynolds number 8000 and *(b)* H/D ratio 3, jet Reynolds number equal 24000

7. Conclusions

- The numerically predicted temperature values using the open source Computational Fluid Dynamics software OpenFOAM is in good agreement with the experimental results with a maximum percentage variation of 0.3%.
- For H/D ratio 3 and jet Reynolds number equal to 24000, the maximum average Nusselt number (value equal to 18.15) is observed on the heating plate whereas for H/D ratio 2 and jet Reynolds number equal to 8000, the minimum average surface Nusselt number (value equal to 14.52) is observed.
- With increase in the jet Reynolds number, average Nusselt number on the heating plate is also increased.

- Increase in H/D ratio produces increased heat transfer rates expect for the jet Reynolds number equal to 8000. At lower jet Reynolds number, the maximum heat transfer rate occurs when the heating plate is closer to the orifice plate.
- Least temperature values were found to be observed at the stagnation zone of the impinging jet where as for a jet Reynolds number equal to 8000 peak in temperature distribution is noticed at the impingement region. This may be attributed to the formation of the toroidal vortices around the jet.
- Single round air jet impingement produces non uniform cooling of the heating plate which may lead to the development of thermal stresses. This disadvantage may be addressed by employing multi jet impingement with or without cross flow.

References

- [1] N. Zuckerman, N. Lior, Jet impingement heat transfer: physics, correlations and numerical modeling, *Adv. Heat Transfer* 39 (6) (2006) 565-631.
- [2] E. Baydar, Y. Ozmen An experimental investigation on flow structures of confined and unconfined impinging air jets, *Heat and Mass Transfer* 42 (4) (2006) 338–346.