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Numerical Simulation for Studying Heat Transfer in Multi Jet Impingement on Sparse and Dense Pin Fin Heat Sinks

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Abstract*— Heat sinks, in electronic systems, are devices that transfer heat from the hotter body by dissipating heat to fluid medium, generally air. These are basically heat exchangers that are used to transfer heat from the components to the surrounding so as to keep away from the problem of overheating. This project presents the numerical simulation of the sparse and dense design pin fin heat sinks when multi jets (3x3 nozzle arrays) are impinging on it. The simulation is carried out to study the effect of Reynold's number (Re) on heat transfer rate at z/D=6 at minimum cross flow condition. The study is carried for three values of Re (7000, 9000 and 11000) and it is observed that as the Re increases the heat transfer rate also increases. It is also observed that the denser design produces higher rate of heat transfer than the sparse design in all the cases. It is noticed that the symmetry of heat transfer patterns occur in the minimum cross flow condition.

Index Terms— **Pin fin, Multi jet -Impingement, Fin Density, Flow simulation.**

I. INTRODUCTION

The advancements in computing technology led to higher data processing rates at tremendous speeds and smaller chip size. This is leading to higher processor temperatures and thus higher heat dissipation necessity. Higher processor temperatures lead to malfunctioning of CPU. Impinging jets, which are used in many applications, can produce high heat transfer rates. It is easy to adjust the location of interest and to remove large amount of heat on the impingement surface.

There is a large class of industrial processes in which jet impingement cooling is applied. For example, jet impingement cooling is used in cooling of blades or vanes in a gas turbine [10], food processing industries [6, 8] and improvement of cooling efficiency in the electronic industry [13-14]. This method is widely used in industry because of features like simple equipment, flexibility of control of mass flow rate and achievement of high heat transfer rates. Compared to single jet impingement, multi-jet impingement gives higher average heat transfer coefficient and uniformity of heat transfer over impingement shell [1]. Over the past 30 years, experimental and numerical investigations of flow and heat transfer characteristics under single and multiple impinging jets remain a very dynamic research area. However, use of multiple jets is much more difficult due to the possibility of interference between nearby jets prior to their impingement on the objective surface. To defeat the above difficulties, appropriate design of multiple jets and proper arrangement to take away exhausted air after impingement is very significant.

Obot and Trabold [14] investigated the result of spent air exit scheme or cross flow scheme on the heat transfer by impinging jets on a flat surface by unrestricting/restricting the spent air: minimum, intermediate and maximum cross flow schemes (or two-way and one-way exits, respectively). This study aims to understand the degree of degradation due to the strength of the cross flow. They proposed that the minimum cross flow scheme shows the greatest performance followed by intermediate and maximum cross flows. The range of Reynolds number for this work is 1000 to 10000.

Kanokjaruvijit et al. [5] experimentally studied 8 x 8 jet array impinging on the staggered array of dimples at Re = 11000 with the help of transient broad band liquid crystal method. The space between the perforated plate and the target plate (Z/d) was adjusted to be 2, 4 and 8 jet diameters to study its effect on the heat transfer performance. Similar trends as observed by Obot and Trabold [7] are also noticed in this work.

Thakare and Joshi [11] evaluated twelve versions of low Reynolds number k-ε models and two low Reynolds number Reynolds stress models (RSM) for heat transfer in turbulence pipe flows. Their comparative study between the k-ε models and RSM models for the Nusselt number prediction is in support of the applicability of the k-ε models even though the RSM model overcomes the assumption of isotropy and the steadiness of turbulence Prandtl number.

Shi et al. [9] presented simulation results for a single semi-confined turbulence slot jet impinging normally on a flat plate. The effects of turbulence models, near wall treatments, turbulence intensity, jet Reynolds number, as well as the type of thermal boundary condition on the heat transfer were calculated using the standard k-ε and RSM models. Their results indicate that both standard k-ε and RSM models forecast the heat transfer rates insufficiently, particularly for low nozzle-to-target spacing. For wall bounded flows, large gradients of velocities, temperature and turbulence scalar quantities exist in the near wall region and thus to incorporate the viscous effects it is required to integrate equations through the viscous sub layer using finer grids with the help of turbulence models.

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Leon F.G. [7] investigated velocity field and turbulence fluctuations in an array of circular jets impinging normally on a plane wall. The dimension indicated that the contact between the self induced cross flow and the wall jets resulted in the formation of horseshoe type vortices that circumscribe the outer jets of the array. It is also observed that center jet has shortest core and highest turbulence kinetic energy, indicative of the strong interaction by a large number of adjacent jets.

II. PROBLEM DESCRIPTION

Our idea in the present work is therefore to study the multiple (3x3) circular air jets impinging on sparse and dense design pin fin heat sinks with minimum cross flow arrangement. The schematics of the arrangement are as shown in Figure 1. The effect Reynolds number (Re) on heat transfer for a given value of z/D (=6) ratio are studied by numerical method. The parameters considered for the cross flow scheme are z/D = 6, Re = 7000, 9000 and 11000. The flow is assumed to be steady, incompressible and three dimensional during the study. Radiation heat transfer effects are neglected. Properties of the fluid such as density, specific heat and thermal conductivity are assumed to be constant.

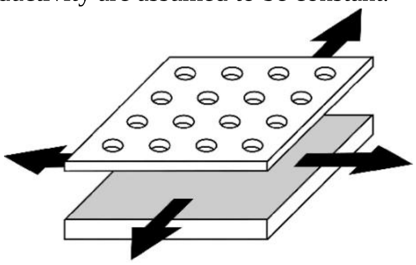


Fig 1 Minimum crosses flow schematic

III. NUMERICAL DETAILS

This section details the model that was formulated to study the effect of changing fin density for various Re values. Figure 2 shows the image of the computational volume considered for analysis. 9 nozzles each 5mm (D) diameter and 25mm long are assumed. Spacing between each nozzle is 15mm. The nozzle plate is positioned above the heat sink such that z/D=6, where z is the height of the nozzle plate base from the heat sink base.

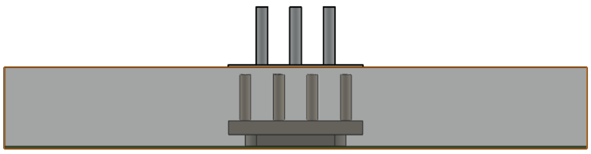
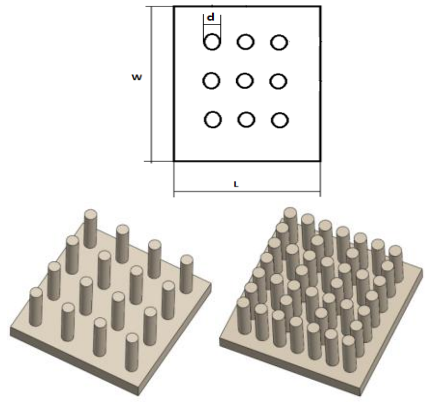


Fig 2 Computational domain with boundary condition for minimum crosses flow condition.

Figure 3 shows the design of sparse and dense heat sinks designs used during simulation. Each fin is 5mm in diameter and 20mm in length. In sparse design, 4 X 4 array of fins is considered with a spacing of 15mm between each fin. In dense design, a 7 X 7 array with no fin in the jet impingement location is considered. Spacing between each fin is 7.5mm. The base plate of the heat sink is 60mm X 60mm X 5mm. Table 1(a) and 1(b) gives the meshing details of sparse as well as dense designs. The processor considered is 45mmX45mmX5mm in dimensions. Heat generation of 30W is applied at the center of the processor [16].



(a) (b) (c)

Fig 3 (a) Nozzle Geometry, (b) Sparse design, (c) Dense Design

Table 1 Mesh Details of (a) Sparse and (b) Dense designs

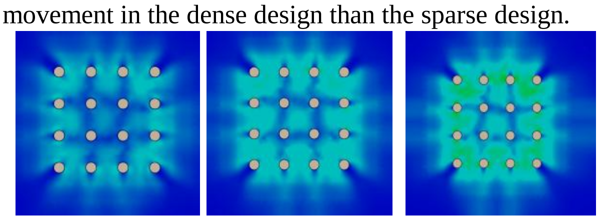
MESH DETAILS FOR SPARSE DESIGN	
Total cells	489754
Fluid cells	171072
Solid cells	84732
Partial cells	233950
Irregular cells	0
Trimmed cells	0

MESH DETAILS FOR DENSE DESIGN	
Total cells	1760049
Fluid cells	980078
Solid cells	339104
Partial cells	440867
Irregular cells	0
Trimmed cells	0

During analysis inlet flow velocities are varied to study the effect of variations of Re in both the designs. The values of velocities and h for various Re values are taken as discussed by N.K.Chougle [2]. Initial wall temperature is taken as 300K. Base of the control volume is defined as adiabatic wall. Steady state analysis is carried out and results are presented in the next section.

IV. RESULTS AND DISCUSSION

Figure 4 & 5 shows velocity contours for multiple-jet impingement for Re = 7000 to 11000 at z/D = 6. From the horizontal cut plots, it can be seen that there is larger region of lower velocity (blue region) in sparse design when compared to dense design. This is mainly because the impinging jet of air is having larger area to disperse in the former design. The restriction to flow in the directions perpendicular to impinging jet resulted in the motion of air till base with higher air velocity without getting dispersed. This resulted in higher air movement in the dense design than the sparse design.



Re =7000 Re = 9000 Re = 11000

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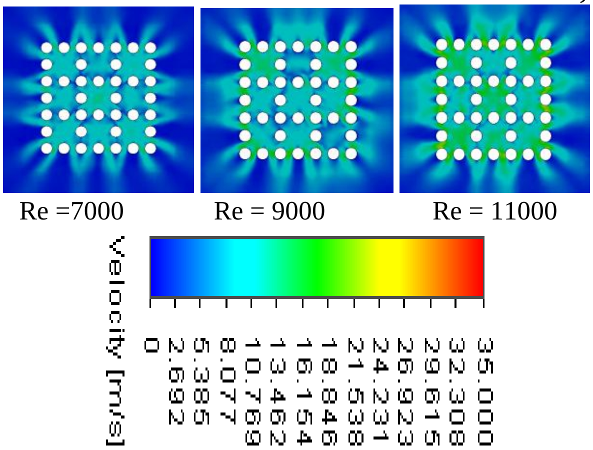


Fig 4 Velocity contours along the horizontal plane 1mm above base plate (Re = 7000-11000).

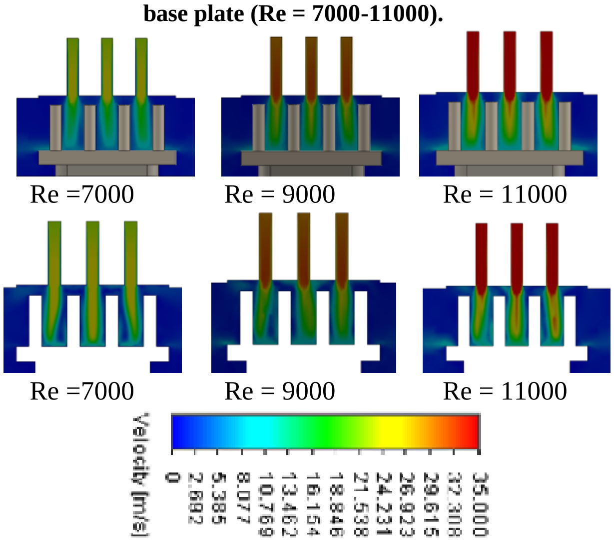
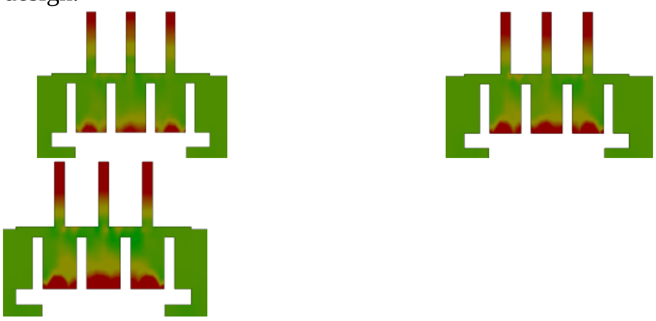


Fig 5 Velocity contours along the Diagonal Vertical plane (Re = 7000-11000)

Figure 6 shows the pressure distribution plots. The cut plots are taken along the diagonal vertical plane. From the plots, it can be seen that higher pressure is at the bottom of the heat sink in both the cases. This indicates that most of the heat transfer occurs at the base and lower portion of the pink fins. It can also be observed that the pressure drop is less in the dense design when compared to sparse design. This is also because of more restriction to air movement in the perpendicular direction to impingement direction in denser design.



Re =7000 Re = 9000 Re = 11000

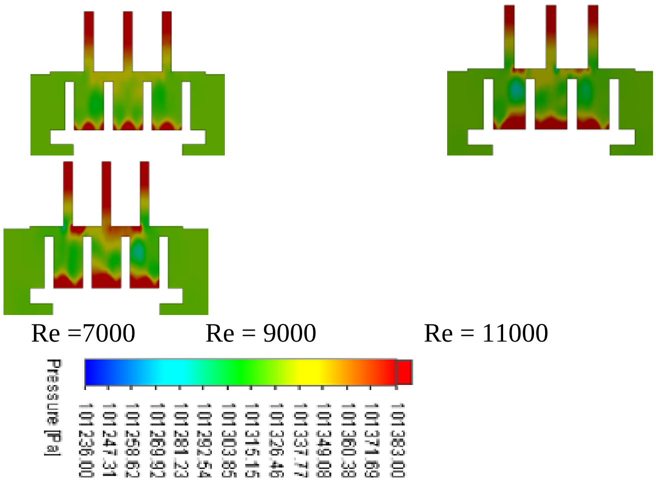


Fig 6 Pressure Contours for Re = 7000-11000

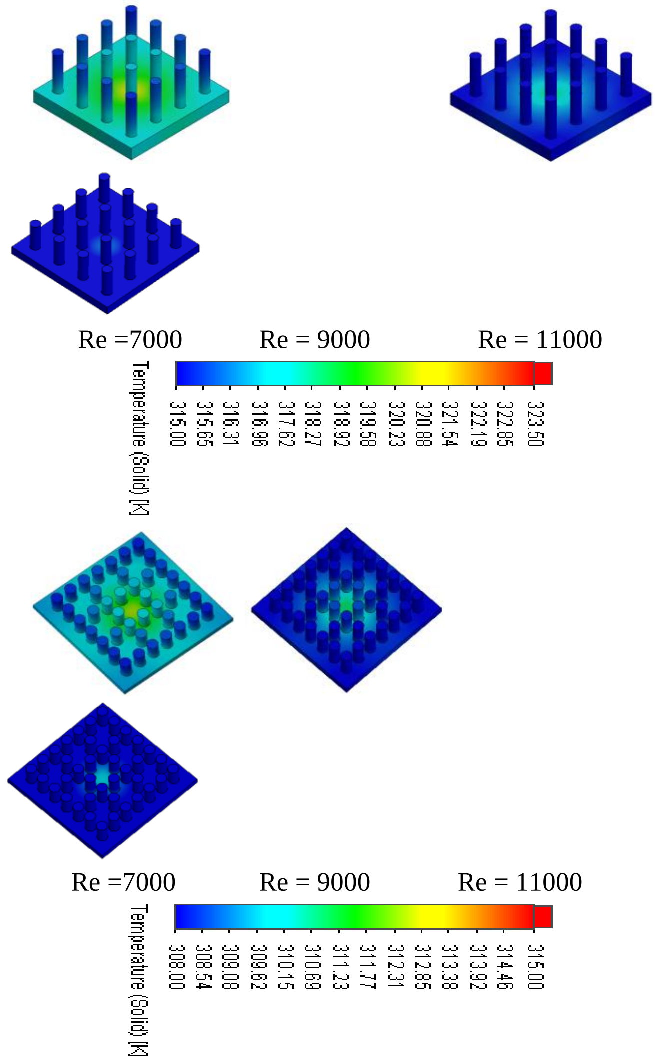


Fig 7 Temperature Contours (Re = 7000-11000)

The heat is transferred from the processor to the base of the heat sink and is dissipated by convection through fins. It is observed that as we increase Reynolds's number, heat sink surface become cooler. Temperature distribution contours of the heat sink for multiple jet impingements on sparse and dense design pin fin heat sink at different Re are shown in

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Figure 7. Since the heat generation is at the center of the processor, it can be observed that the temperature decreases radially outwards from the center of the heat sink in all cases. Also it can be observed that the average temperature is lower in case of dense design than when compared to sparse design. The max temperature for dense design for Re=7000 is 315K while in the sparse design it is found to be 323K. In all the three cases (Re=7000, 9000, 11000) it is observed from the analysis that dense design of heat sink is 10K lower in temperature when compared to sparse design. This is because of the reasons discussed previously:

i) higher air velocity at the base in dense design when compared to sparse design

ii) Lower pressure drop in dense design when compared to sparse design

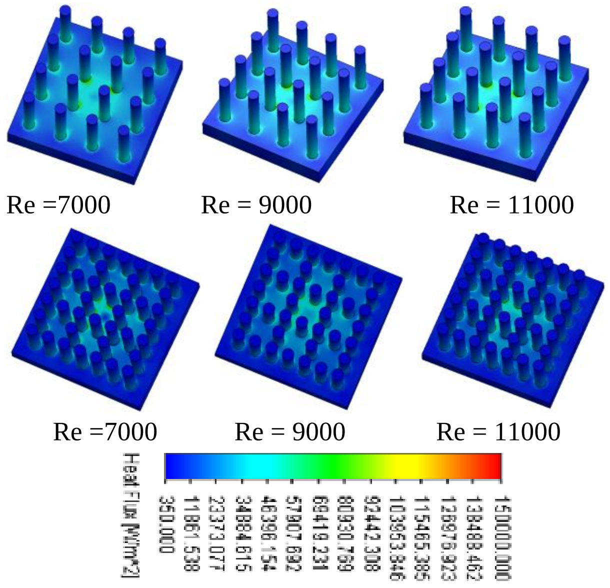

Fig 8 shows the variation of heat flux for sparse and dense designs at Re = 7000 to 11000.

V. CONCLUSION

Jet impingement technique is an efficient way of cooling electronic equipment. Many studies are being carried out in this area. In this paper numerical investigations are carried to study the effect of multi-jet (3X3) impingement on sparse and dense design pin fin heat sinks. Geometry and boundary conditions are described. 30W heat generation is considered and the study is carried for three values of Re (7000, 9000, 11000) for z/D=6. From the results it is observed that the denser design of pin fin heat sinks gives better results (10K lesser temperature) than sparse design in each case. This is because of higher air velocity at lower pressure drop in denser design due to greater restriction of flow in direction perpendicular to the impingement direction. It is also observed that the heat dissipation is increasing with increase in Re value in both designs.

NOMENCLATURE

D	Diameter of the nozzle, m
Re	Reynold's number based on jet diameter = ρvd/μ
h	Heat transfer coefficient, w/m^2K
v	Velocity of the jet, m/s
z	Distance between nozzle exit and target plate, m
z/D	Dimensionless jet to target plate spacing.

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