International Journal of Advance Research in Science and Engineering
Volume No.07, Special Issue No.03, April 2018
www.ijarse.com
IJARSE
ISSN: 2319-8354

Numerical analysis of Perforated Plate Pin Fin Type of Heat Sink

Akshaykumar Patil [1] , Suresh Sawant[2]

[1] Professor, [2] PG Student

Department of Mechanical Engineering,

Rajarambapu Institute of Technology, Rajaramnagar, MS (India)

ABSTRACT

Pin fins heat sink are used to increase heat transfer in variety of applications such as cooling of electronic component, (Integrated CPUs, chipset and hard disk drives), heat exchanger etc. It is very important to enhance the heat transfer rate and ensure the better cooling of electronic devices. Heat transfer rate in natural convection of parallel plate fin heat sink with circular pin fins placed between the plate fins. Parallel plate fin heat sink with perforated circular pin fins gives more heat transfer than without perforated circular pin fins. Perforated pin fin heat sink gives higher heat transfer rate compared to solid pin fin heat sink by varying the different parameter like shape of pin fin, diameter of Perforations and number of Perforations different heat transfer rate can be achieved. The number of perforation on pin fin increases and its effect on heat transfer rate. The well-known k–ε two-equation turbulence model is employed to describe the turbulent structure and behavior. The parameters include the inlet air flow velocity (U_{in}=6.5, 8.0, 10.0 and 12.2 m/s) and cross cut lengths (L_c=1.5). At given number of cross-cuts, minimum thermal resistance is observed at cross-cut of 1.5 mm along with reduction in pressure drop in comparison with other cross-cut lengths.

Keywords—Heat transfer, Pin fin perforation, Pressure drop, Thermal resistance.

I. INTRODUCTION

Since the rapid development of electronic technology, electronic appliances and devices now are always in our daily life. Under the condition of multifunction, high clock speed, shrinking package size, and higher power dissipations, the heat flux per unit area increased dramatically over 100 W/cm^2.Heat sink is a device which absorbs the heat from heated component and dissipates heat to surrounding air. Air cooling is the most widely used technique for heat rejection. Heat sink dissipates heat to surrounding air by convection. In forced convection heat dissipated by means of some external sources such as fans. . In natural convection, the extended surfaces are used which increases surface area by adding fins to the surface in order to achieve required rate of heat transfer. Heat transfer from a pin fin heat sink depends on many parameters; shape and size of the fins, spacing between the pin fins, number of pin fins, number of perforation, diameter of perforation, thermal conductivity, the type of flow and the fluid, temperature difference between the heat sink and the fluid[1].Besides, the working temperature of the electronic components may exceed the desired temperature level. Thus, the effective removal of heat dissipations and maintaining the die at a safe operating temperature has played an important role in ensuring a reliable operation of electronic components. There are many methods in electronics cooling such as jet impingement [2], heat pipe [3], parallel flow cooling etc. For the parallel flow case, the test section friction factor is lower than that of impinging flow, which is slightly lower than that of reverse impinging flow. Hence overall efficiency of parallel flow cooling is higher in core region [4]. The effect of flow directions and behaviors on the thermal performance of heat sinks is reviewed by Chingulpitak and Wongwises [5]. Some research has been done on optimization of fin height, fin width and inter fin spacing. For given fin width, the thermal resistance of the plate-fin heat sink decreased with an increase in the fin height. This was due to the fact that the heat-transfer area of the higher fin height was larger than that of the lower fin height. For a given fin height, the optimal levels of fin width that provided the lowest thermal resistance were increased with Reynolds number and further increment in the fin width will reduce thermal performance [6]. The effect of fin height on the decrease of thermal resistance became less significant as the fin height exceeded 20 mm [7].The three dimensional numerical study carried by found that the boundary layer development and horseshoe vortices between the fins are dependent on fin spacing to height ratio and Reynolds number. The thermal and hydraulic performance of optimized geometries of plate fin heat sink. Another analytical study to predict the optimum geometry of PFHS showed that 0.8 mm fin thickness and 2 mm fin space are optimum parameters in case of PFHS

With the advent of Computational Fluid Dynamics (CFD) in the recent years, flow and heat transfer computations have become quite readily possible. In particular, with the recent introduction of high-power workstations and personal computers the cost of such computations has been drastically reduced, and as a result many CFD codes have come into the market. More recently, such computations have become very popular throughout the application area of cooling of electronic components. In fact, this popularity has led to several purposes developed CFD codes coming into the market which are specifically tailored for use by heat transfer engineers in the electronic industry. The relatively recent adoption of CFD simulation studies in electronic cooling applications has prompted some interest in validation of these codes for these types of problems.

II. NUMERICAL ANALYSIS

A. *Problem Definition***:**

The introduction of circular pins in between the parallel plates of plate pin fin heat sink (PFHS) gives new design as shown in Fig.1, known as perforated plate pin-fin heat sink (PPFHS) . This improves the thermo-hydraulic performance of heat sink. But still there is a chance to further improve overall performance of PPFHS by working on turbulent flow structure in air flow passage. For that purpose cross cut is introduced on fin wall at centre distance of circular pins as shown in Fig. 1 (b). Due to periodic structure of heat sink and to reduce computational time, only single passage flow is investigated. The detailed geometry parameters for cross cut

International Journal of Advance Research in Science and Engineering
Volume No.07, Special Issue No.03, April 2018
www.ijarse.com
IJARSE
ISSN: 2319-8354

PPFHS are given in Table 1. It is assumed that heat sink is made of aluminum material as continuum with thermal conductivity of 202.4 W/m-K.

Table 1: All dimensions of perforated pin fin type of heat sink

Fin length, l (mm)	Fin width, w (mm)	Fin height, H (mm)	Fin thickness, t (mm)
51	45.5	10	1.5
Fin spacing, δ (mm)	Fin number, N	Pin height, h (mm)	Pin diameter, D (mm)
4	9	10	2
Pin spacing, S (mm)	Pin number, N	Pin perforation dia.(mm)	Number of perforation/pin
20	24	1	3

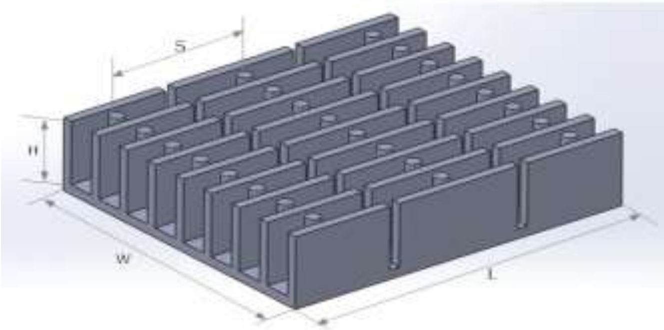

Fig.1 Plate Pin Fin Heat Sink with cross cut

Fig.2 Perforated Plate Pin Fin Heat Sink with cross cut

B. Physical Models:

Based on the Reynolds number, either viscous laminar model or standard k-ε model is used for laminar and turbulent flow respectively. Here the flow is turbulent hence k-ε turbulence model was used. SIMPLE algorithm was imposed to segregate the pressure-velocity coupled equations. The power law discretization scheme interpolates the face value of momentum, k, ε and also second order discretization scheme is applied for pressure values. The coupled wall condition was applied between mesh interface of fluid and solid.

C. Governing Equations:

Momentum and energy conservation equations along with continuity should be solved as governing equations to consider the conjugate heat transfer between fins and fluid flow. To reach the following governing equations, it is assumed that the flow is incompressible and viscous dissipations are also negligible. Due to the flow pattern and vortex generating condition inside the Perforated PPFHS, air flow between the plate fins are assumed to be turbulence. Decomposing the velocities to mean and fluctuating parts and taking the mean from the governing equations will reach to :

Continuity:

$$\frac{\partial \bar{u}_i}{\partial x_i} = 0 \qquad \ldots\ldots. \quad (2.1)$$

Momentum:

$$\rho \bar{u}_j \frac{\partial \bar{u}_i}{\partial x_j} = -\frac{\partial \bar{p}}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu_l \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i}\right) - \rho \overline{u_i u_j}\right] \ldots\ldots. \quad (2.2)$$

Energy:

$$\rho \bar{u}_j \frac{\partial \bar{T}}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\frac{\mu_l}{\sigma_l} + \frac{\mu_t}{\sigma_t}\right)\frac{\partial \bar{T}}{\partial x_j}\right] \qquad \ldots\ldots \quad (2.3)$$

Where, expresses the velocity and is kinematic viscosity and superscript _ stands for mean values.

To continue and close the above equations, k-ε turbulence model has been used and results obtained based on this model. The transport equations for k and ε are given as follows:

Transport equation for k

$$\rho \bar{u}_j \frac{\partial k}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\mu_l + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + \mu_t \left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i}\right)\frac{\partial \bar{u}_i}{\partial x_j} - \rho\varepsilon \qquad \ldots\ldots. \quad (2.4)$$

Transport equation for

$$\rho \bar{u}_j \frac{\partial \varepsilon}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\mu_l + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + C_1 \mu_t \frac{\varepsilon}{k}\left(\frac{\partial \bar{u}_i}{\partial x_j} + \frac{\partial \bar{u}_j}{\partial x_i}\right)\frac{\partial \bar{u}_i}{\partial x_j} - C_2 \rho \frac{\varepsilon^2}{k} \qquad \ldots\ldots.. \quad (2.5)$$

Closures coefficients are set to: =1.44, =1.92, =1, =1.

SIMPLE algorithm was used to solve pressure-velocity coupled equations. The power law discretization scheme interpolates the face value of momentum, k, ε and second order discretization scheme is applied for pressure and energy values. The pressure drop (ΔP) and thermal resistance (R_{th})are calculated as:

$$\Delta P = P_{in} - P_{out} \qquad \text{......... .(2.6)}$$

$$R_{th} = \frac{\Delta T}{Q} \qquad \text{......... (2.7)}$$

Where, P_{in} and P_{out} express the inlet and outlet pressures of the air in single duct, ΔT stands for the difference between temperature on the fin base and the ambient temperature and Q is the constant heating power applied on base plate of heat sink.

D. Boundary conditions:

The heat sink is assumed to be continuity material. This means that the effect of thermal contact resistance on the heat transfer between the base and fin is neglected.

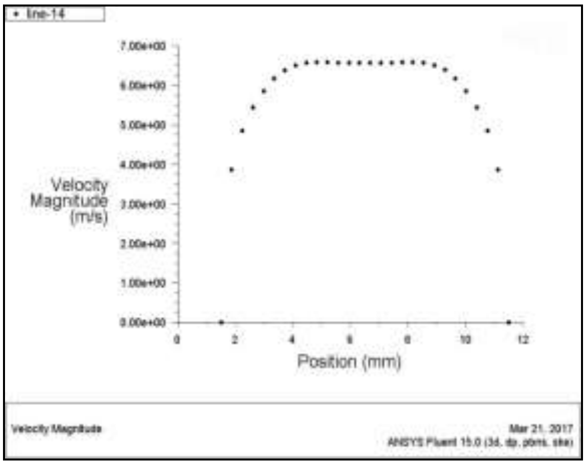

Fig.3 Velocity profile at inlet of heat sink for 6.5m/sS

The flow of fluid is assumed to be fully developed. For that purpose first the entry length is decided by using Eq. 5.1 and then velocity profile was wrote at the inlet of heat sink with velocities of 6.5, 8.0, 10.0 and 12.2 m/s. This velocity profile then read at inlet of flow domain. It is the carried out the thermal performance of a pin fin heat sink.

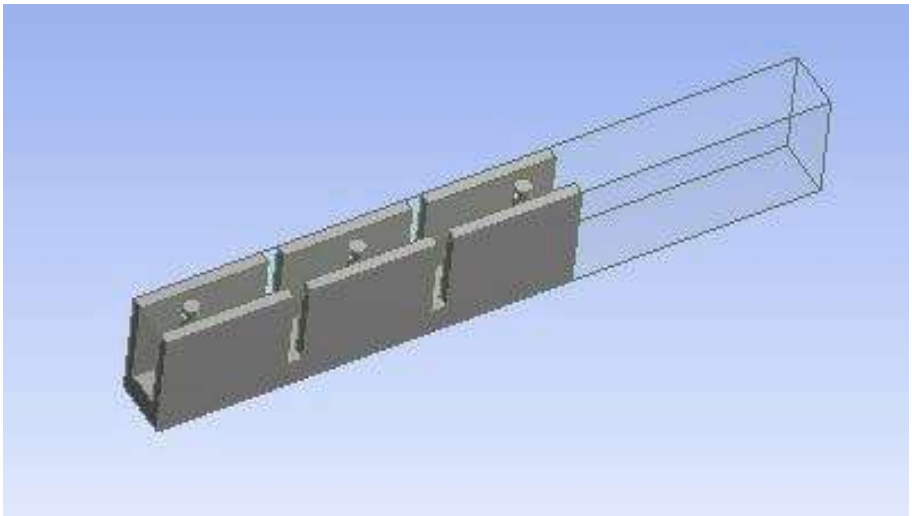

Fig.4 Computational domain of Perforated PPFHS

Fig.4 illustrates the computational domain for the plate fin heat sink. Similar type of computational domains were used for PPFHS and PPFHS with cross cut. It is assumed that the heat sink is made of aluminum with thermal conductivity 202.4W/mK and the flow is fully developed at inlet.

III.RESULTS AND DISCUSSION

Table 2**:** Grid Independency for PPFHS with cross cut with perforation at 6.5 m/s and 10 W

Sr. No.	No. of Elements	Thermal Resistance(K/W)	Pressure Drop(Pa)
1	662651	0.942	98.526
2	1262455	0.880	95.773
3	1924786	0.754	89.799
4	4052869	0.693	90.598

Table 2 grid independency for PPFHS with cross cut with perforation and 1924786 and 4052869 no. of elements shows slight variation in results.

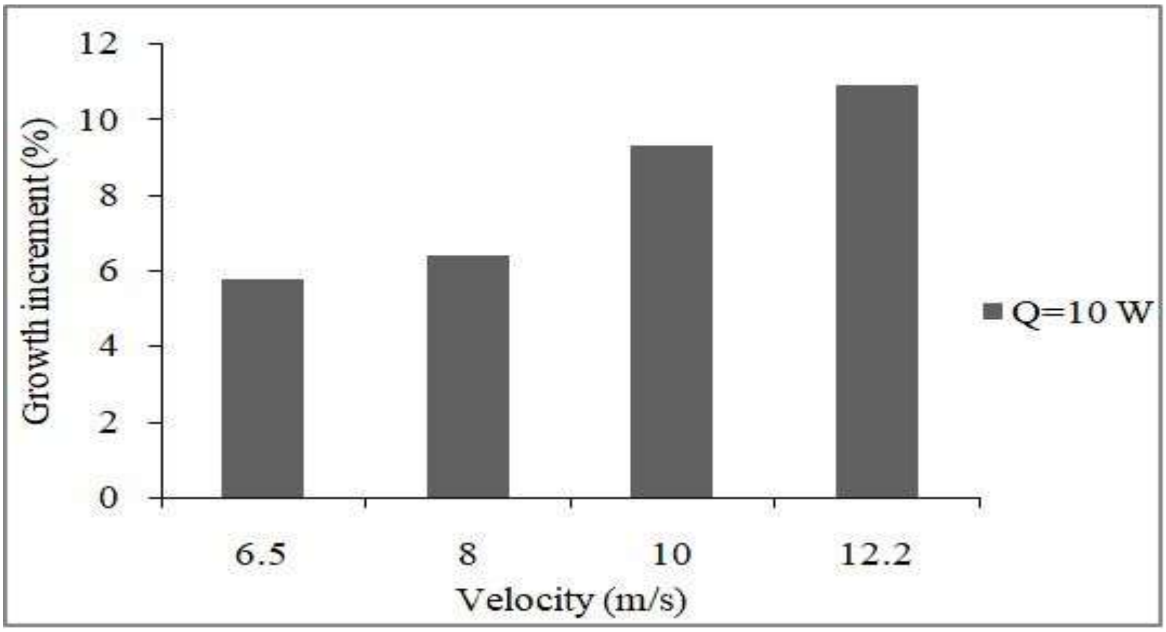

Fig.5 Velocity effect on in perforated PPFHS with Lc= 1.5 mm and PPFHS at 10 m/s

The proper recirculation of air occurs in case of cross cut length of 1.5 mm, shows lesser pressure drop than other cross cut lengths as shown in Fig. 6. Earlier it is seen that CPPFHS

decreases the thermal resistance with simultaneously increasing pressure drop in comparison of PPFHS.

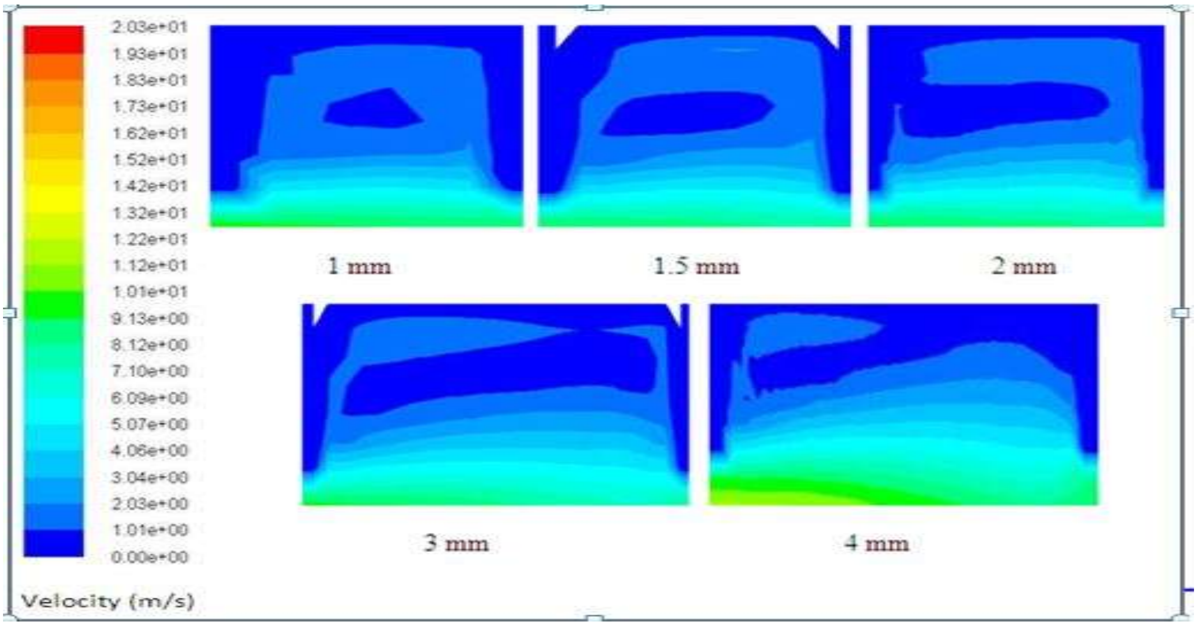

Fig.6 Flow behavior of air inside cross cut lengths at 10 m/s and 10W

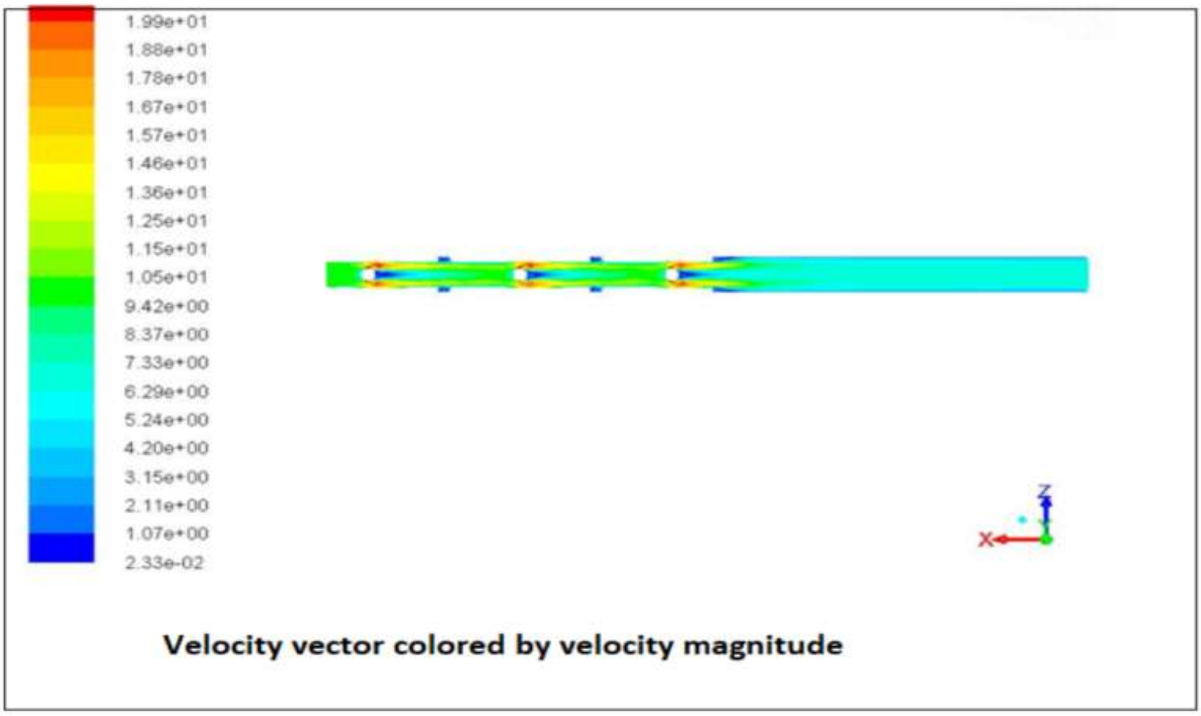

Fig.7 Velocity vector colored by Velocity Magnitude and Contours of Total Temperature

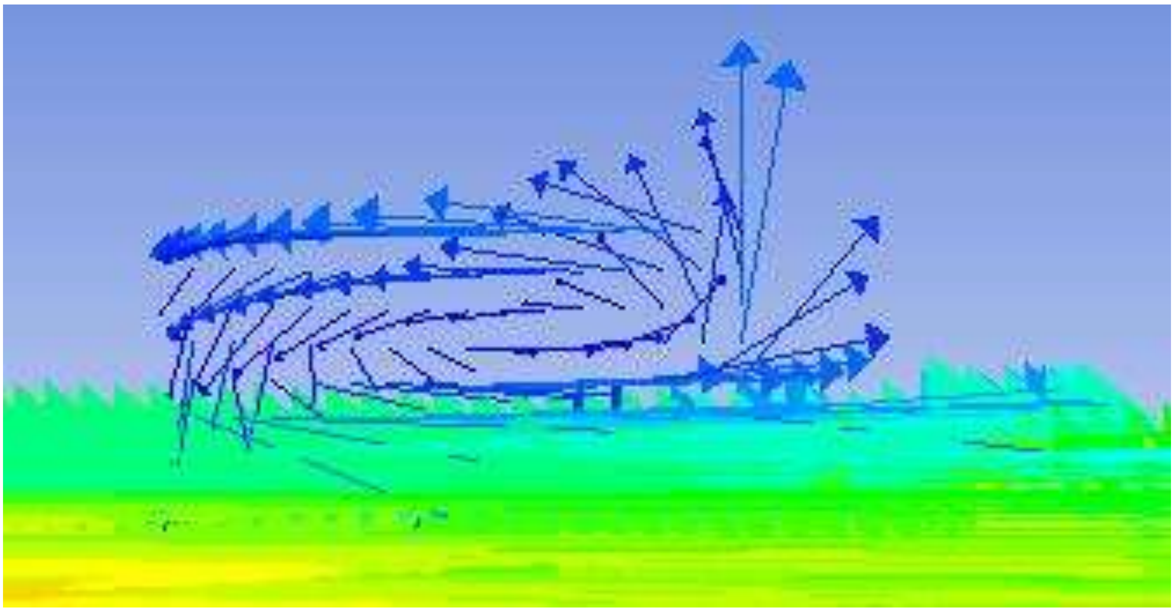

Fig.6 Flow behavior of air inside cross cut lengths at 10 m/s and 10W

IV. CONCLUSIONS

- The perforated pin fin is a light in weight compared to solid pin fin.
- The perforated pin fin having high heat transfer coefficient than solid pin fin.
- The heat transfer enhancement is depending on pin fin dimensions,the perforation Geometry , and number of perforation and thermal conductivity of material.
- Numerical simulation results show that the thermal resistance of a perforated PPFHS is lower than that of a PPFHS.
- The thermal performance of perforated configuration is superior to the solid pin fin geometry for the cases of laminar flow with any working coolant or turbulent flow when air is applied.

V. REFERENCES

[1] I. Mudawar, Recent advances in high-flux, two-phase thermal management, Journal of Thermal Science and Engineering Applications, 5(2) (2013) 021012.

[2] Y. M. Chung, &K. H. Luo, Unsteady heat transfer analysis of an impinging jet, Transactions-American Society of Mechanical Engineers journal of heat transfer, 124(6) (2002) 1039-1048.

[3] K. S. Kim, M. H. Won, J. W. Kim,&B. J. Back, Heat pipe cooling technology for desktop PC CPU, Applied thermal engineering, 23(9) (2003) 1137-1144.

[4] S. A. El-Sayed, S. M. Mohamed, A. A. Abdel-latif,& E. A. Abdel-hamid, Experimental study of heat transfer and fluid flow in longitudinal rectangular-fin array located in different orientations in fluid flow, Experimental Thermal and Fluid Science, 29(1) (2004) 113-128.

[5] S. Chingulpitak,&S. Wongwises, A review of the effect of flow directions and behaviors on the thermal performance of conventional heat sinks, International Journal of Heat and Mass Transfer, 81 (2015) 10-18.

[6] H. Y. Li, &S. M. Chao, Measurement of performance of plate-fin heat sinks with cross flow cooling, International Journal of Heat and Mass Transfer, 52(13) (2009) 2949-2955.

[7] H. Y. Li, M. H. Chiang, C. I. Lee, &W. J. Yang, Thermal performance of plate-fin vapor chamber heat sinks, International Communications in Heat and Mass Transfer, 37(7) (2010) 731-738.

ACKNOWLEDGEMENTS

I must mention several individuals and organizations that were of enormous help in the development of this work. Dr. S. M. Sawant my supervisor, philosopher and personality with a Midas touch encouraged me to carry this work. His continuous invaluable knowledgably guidance throughout the course of this study helped me to complete the work up to this stage and hope will continue in further research. In addition, very energetic and competitive atmosphere of the Mechanical Engineering Department had much to do with this work. I acknowledge with thanks to, Head of Department, Head of Program, and faculty of the department, Central library and Colleagues.