Numerical and Experimental Analysis of Perforated type of Heat Sink

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***Abstract**: -* This paper numerically studied the heat transfer and flow characteristics of plate pin-fin heat sink, cross cut and perforation heat sinks using commercial CFD software FLUENT. The governing equations are discretized by using a control volume based finite-difference method. The coupling of the velocity and the pressure terms of momentum equations are solved by the SIMPLE algorithm. 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}=6.0, 7.0, 8.0 and 9.0 m/s), perforation and cross cut lengths. At given number of perforations, decrease thermal resistance is observed at three numbers of perforation along with increases in pressure drop in comparison with experimentally studied.

***Key-Words**: -* Perforated heat sink, Heat transfer, Pressure drop, Thermal resistance.

1 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 [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 insuring 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]. Also the effect of flow directions and behaviours on the thermal performance of heat sinks [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[7]. The effect of fin height on the decrease of thermal resistance became less significant as the fin height exceeded 20 mm [8]. [10] The three dimensional numerical study found that the boundary layer development and horseshoe vortices between the fins are dependent on fin spacing to height ratio and Reynolds number. [11] The theoretical model is proposed to predict 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 [12]. [13] The entropy generation minimization is another optimization technique used to optimize plate fin heat sink and coefficient of performance is presented using least energy optimization in forced convection cooling [14]. However, some researchers have focused on the flow behaviour in plate-pin fin heat sink. The study showed that pins could disturb the flow field, enhance heat transfer and also increase flow resistance as well by analyzing the influence of number and arrangements of fins. Numerical simulations and some experiments showed that the thermal resistance of a PPFHS is 30% lower than that of PFHS at same air velocity and profile factor of the former is about 20% higher than that of the latter with same pumping power. The variation in pin heights gives better results than equal heights of pins for in-line configuration in case of plate pin-fin heat sink. The key parameter in PPFHS is pin diameter. The increase in pin diameter decreases thermal resistance and simultaneously increases the flow resistance.

Heat shield is a sheet metal placed at top entrance of heat sink to reduce the bypass flow effect and forces more coolant to flow through fin channel. The introduction of shield decreases the thermal resistance for given fin width, fin height and Reynolds number [6]. But this decrease in thermal resistance reduces at high Reynolds number. The thermal resistance increases with increase in angle of shield from 45^0 to 135^0[9]. The best compromise between thermal resistance and pressure drop is achieved at shield angle of 90^0. For low Reynolds number, solid shield gives better performance.

In this work, the modification in design of PPFHS is applied in order to enhance heat transfer by reducing thermal resistance. But this should not be dominated by pressure drop increment so to prove that the synthetically performance of modified design perforation pins is better than PPFHS.

2. Problem Definition:

Plate Fin Heat Sinks (PFHSs) are designed and manufactured for increasing heat transfer from heat source plates as used for electronic devices. A simple PFHS type which is usually used in common applications. This heat sink modified by locating cylindrical pins between plates and introduced Plate Pin Fin Heat Sinks (PPFHSs). As it is locating the cylindrical pins between the plate fins reduces the thermal resistance of PPFHSs by 30% compared to the PFHSs but with a drastic increase in the pressure drop. So, while slightly modification in PPFHS by locating perforation on the per pins and that results we have to reduces the thermal resistance by 10-20% and increases in the pressure drop.

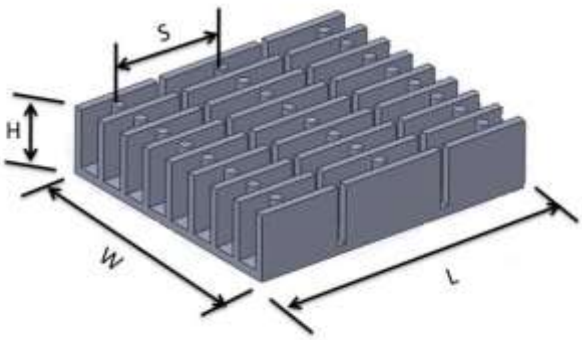

Fig.1 Perforated Plate Pin Fin Heat Sink with cross cut

3. Numerical Analysis:

Numerical methods are extensively used to analyze the performance of the behaviour and also to design the heat sinks for electronic cooling applications. Computational Fluid Dynamics (CFD) is a computer-based numerical tool used to study the fluid flow, heat transfer behaviour and also its associated phenomena such as chemical reaction. A set of mathematical model equations are first developed following conservation laws. These equations are then solved using a computer program in order to obtain the flow variables throughout the computational domain. Examples of CFD applications in the chemical process industry include drying, combustion, separation, heat exchange, mass transfer, pipeline flow, reaction, mixing, multiphase systems and material processing.

3.1 Modeling:

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 aluminium with thermal conductivity 202.4 W/m K and the flow is fully developed at inlet.

Table 1: Geometry parameters 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

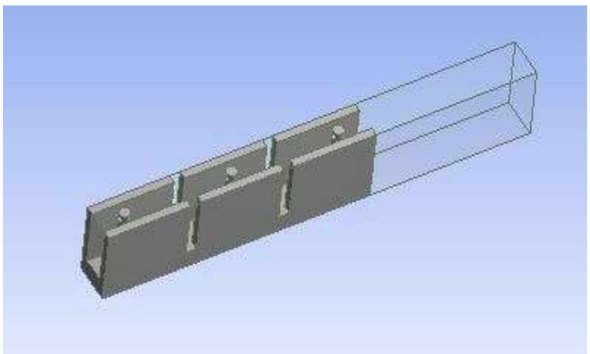

Fig.2 Computational domain of Perforated PPFHS

3.2 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:

$$\frac{\partial \overline{u_i}}{\partial x_i} = 0$$

Momentum:

$$\rho \overline{u_j} \frac{\partial \overline{u_i}}{\partial x_j} = -\frac{\partial \overline{p}}{\partial x_i} + \frac{\partial}{\partial x_j}\left[\mu_l \left(\frac{\partial \overline{u_i}}{\partial x_j} + \frac{\partial \overline{u_j}}{\partial x_i}\right) - \rho \overline{u_i u_j}\right]$$

Energy:

$$\rho \overline{u_j} \frac{\partial \overline{T}}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\frac{\mu_l}{\sigma_l} + \frac{\mu_t}{\sigma_t}\right)\frac{\partial \overline{T}}{\partial x_j}\right]$$

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,

$$\rho \overline{u_j} \frac{\partial k}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\mu_l + \frac{\mu_t}{\sigma_k}\right)\frac{\partial k}{\partial x_j}\right] + \mu_t \left(\frac{\partial \overline{u_i}}{\partial x_j} + \frac{\partial \overline{u_j}}{\partial x_i}\right)\frac{\partial \overline{u_i}}{\partial x_j} - \rho\varepsilon$$

Transport equation for,

$$\rho \overline{u_j} \frac{\partial \varepsilon}{\partial x_j} = \frac{\partial}{\partial x_j}\left[\left(\mu_l + \frac{\mu_t}{\sigma_\varepsilon}\right)\frac{\partial \varepsilon}{\partial x_j}\right] + C_1 \mu_t \frac{\varepsilon}{k}\left(\frac{\partial \overline{u_i}}{\partial x_j} + \frac{\partial \overline{u_j}}{\partial x_i}\right)\frac{\partial \overline{u_i}}{\partial x_j} - C_2 \rho \frac{\varepsilon^2}{k}$$

Closures coefficients are set to: C_1=1.44, C_2=1.92.

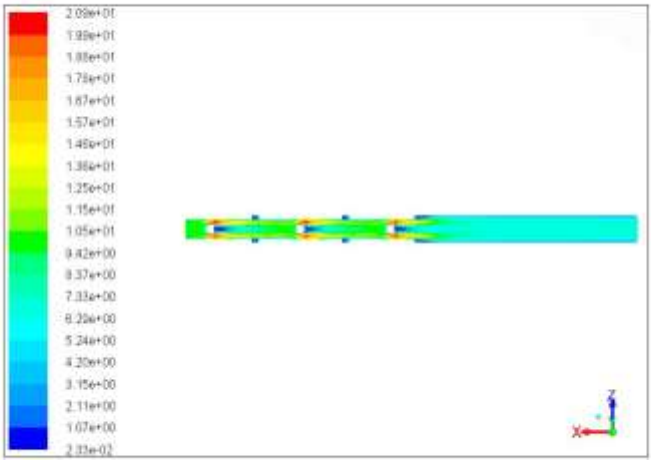

Fig.3 Velocity vector by velocity magnitude

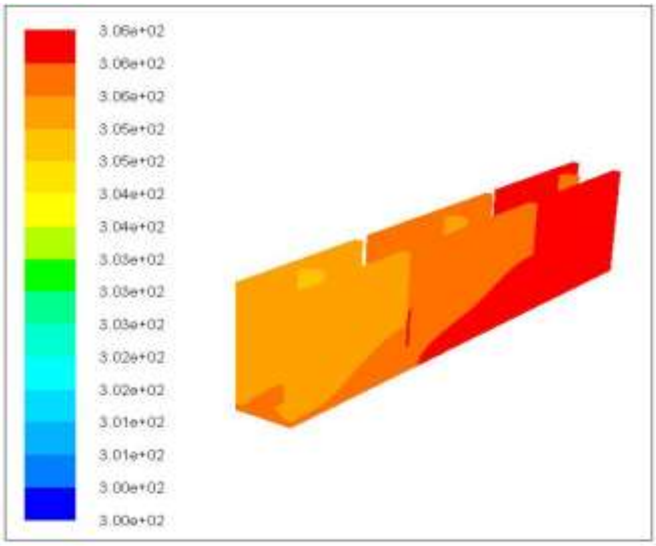

Fig.4 Contour Of total temperature (k)

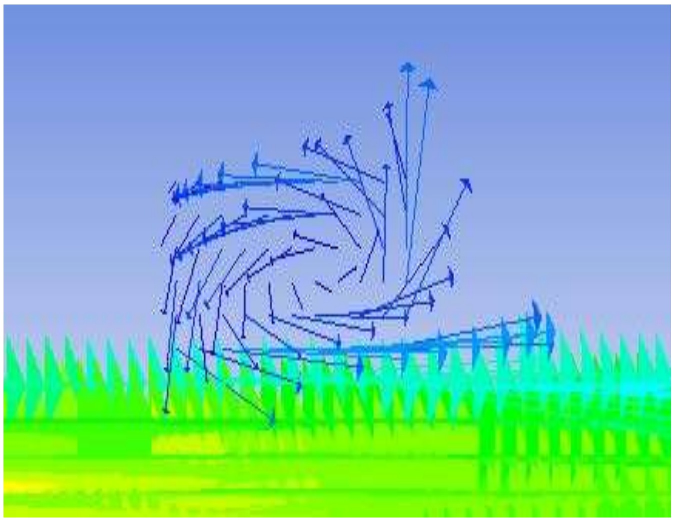

Fig.5 Contours of Total Temperature

4. Experimentation:

The setup is used to test heat sink for its thermo-hydraulic performance. The main design criteria were:
1. Open circuit wind tunnel.
2. Good flow quality.
3. Test section aspect ratio of 1 and the maximum test section length possible in the available space.
4. Flow speed in the test section between 1 m/s to 5 m/s.
5. Low noise level.

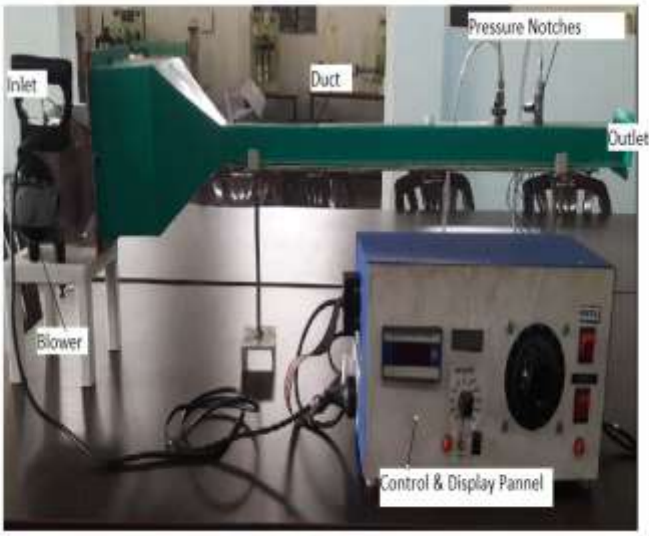

Fig.6 Experimental setup for testing heat sink

4.1 Results and Discussion:

4.1.1Calculation of thermal resistance:

The heat transfer rate from heat sink to air is calculated by following equation:

$$Q_a = m_a C_{p,a}(T_{a,avg,out} - T_{a,avg,in})$$

Where, m_a is mass flow rate of air, $C_{p,a}$ is specific heat of air, and $T_{a,avg,out}$ and $T_{a,avg,in}$ are average air temperature at the outlet and inlet respectively. The average heat transfer rate between heat supplied to test section and heat absorbed by air can be determined as $Q_{avg,,}$

$$Q_{avg} = (Q_a + Q_{heater})/2$$

Then thermal resistance of air flowing through heat sink can be calculated as:

$$R_{th} = \frac{(T_{b,avg} - T_{a,avg})}{Q_{avg}}$$

4.2 Plate-pin fin heat sink with Perforation (Cross cut 1.5mm)

The experiment is carried out to found the thermal resistance and pressure drop in plate pin-fin heat sink Perforation which is calculated mathematically

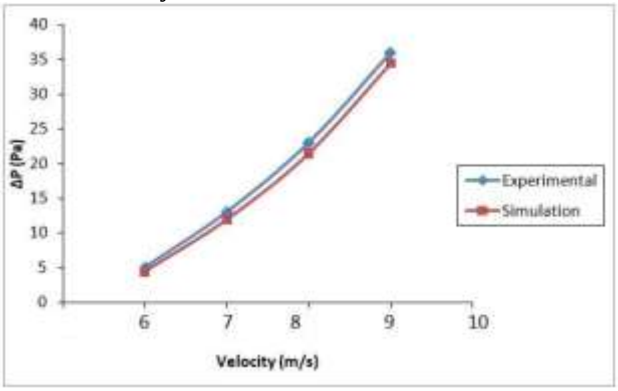

Fig.7 Comparison of experimental and simulation results of pressure drop in PPFHS with Perforation.

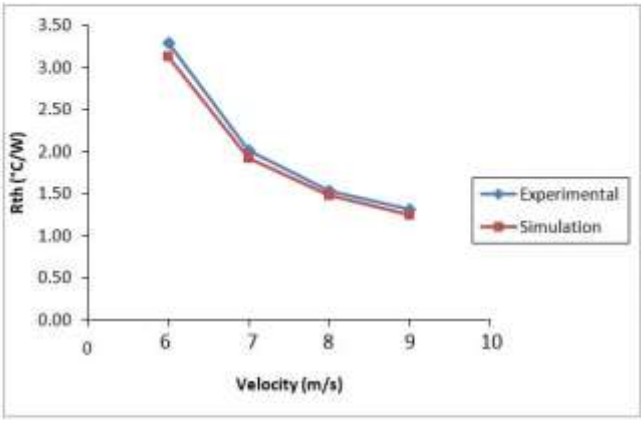

Fig.8 Comparison of experimental and simulation results of pressure drop in PPFHS with Perforation.

Fig.7 and Fig.8 shows the comparison between thermal resistance and pressure drop in case of PPFHS with perforation of cross cut 1.5 mm. From fig.7 results of simulation and the experimental results are nearly same for low velocity. For higher velocity there is slight change in results and the difference is about 6% and similarly for pressured drop the difference in simulation and experimental values are of 12%.

5. Conclusions:

Numerical procedure has been used to investigate the effect of cross cut lengths on thermal and hydraulic performance of perforated plate pin-fin heat sink (PPFHS). Also to combine the effect of thermal resistance and pressure drop. Comparing the results showed that addition of perforation and cross cut in pin, fin wall decreases the thermal resistance by increasing pressure drop simultaneously. But overall performance of perforated PPFHS is higher than PPFHS and would be considered as an efficient alternative for PPFHS. From numerical analysis and experimental analysis we can conclude that

1. Increasing the cross cut length does not affect more on thermal resistance but pressure drop for Lc=1.5 mm is low due to flow stabilization in cross cut region.
2. The best performance of Perforation occurs at high air flow velocity and high heat fluxes.
3. The introduction of perforation and cross cut ensure the upper limit temperature of 358K for CPU, in all heating powers below 50 W and in all air flow velocities. Also, for 100 W, with flow velocity beyond 12.2 m/s guarantees the peak temperature of 358K**.**
4. While comparing the results of numerical and experimental analysis, the results are seen to be appropriate. As continuously decrease in thermal resistances, simultaneously increases in pressure drop.

NOMENCLATURE

Latin symbols	
U	Flow velocity [m/s]
Q	Heating Power [W]
R	Overall thermal resistance [K W^{-1}]
P	Pressure [Pa]
Re	Reynolds number [-]
C_p	Specific heat [J/kg K]
T	Temperature [K]
k	Thermal conductivity [W/m K]
C_1,C_2,C_μ	Turbulent constants
Greek symbols	
Δ	Differential
ε	Dissipation rate of turbulent energy [m^2/s^3]
μ_t	Eddy viscosity of the air [kg/s m]
ρ	Fluid density [kg/m^3]
μ	Fluid dynamic viscosity [Pa s]
μ_l	Kinematic viscosity [m^2/s]
σ_ε, σ_k	k-ε turbulence model constant for ε and k
Subscript	
in	Inlet
out	Outlet

ABBREVATIONS

PPFHS	Plate Pin Fin Heat Sink
3D, 2D, 1D	Three, two and one dimension

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