ISSN 2395-1621

Experimental and computational investigation of forced convection analysis of plate circular pin fin heat sinks over vertical base

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ABSTRACT

It is essential to provide adequate cooling in electronic components to avoid overheating and increase the performance of the system. Recognizing the poor heat transfer performance of air-cooling, secondary fin surface is usually exploited to solve the increased demand of heat dissipating. For further improvement of the overall heat transfer performance, augmentation via fin pattern is often employed. Among the fin configurations being implemented in practical applications, pin fin is regarded as one of the most effective heat transfer augmentation methods for providing boundary layer restarting and periodic vortex shedding. A comparative study of pin fin heat sinks having circular, elliptic, and square cross-section was done by Kai-Shing Yang A, Wei-Hsin Chu B [4], where they found that for an inline arrangement, circular pin fin shows an appreciable influence of fin density whereas virtually no effect of fin density is seen for square fin geometry. This is associated with the unique deflection flow pattern accompanied with the inline circular configuration. For the staggered arrangement, the heat transfer coefficient increases with the rise of fin density for all the three configurations. For rectangular plate fin heat sinks it has been observed that there is boundary layer growth in forced convection, which decreases the heat transfer rate. To overcome this problem, some circular fins are inserted in the passage of plate fin. By doing so an average heat transfer coefficient of the rectangular plate pin fin heat sink (PPFHS) was increased and the thermal resistance was decrease by 30% compared to rectangular plate fin heat sink (PFHS) under the condition of equal wind velocity, and the profit factor of the former is about 20% higher than that of the latter with the same pumping power as proposed by Xiaoling Yu and JianmeiFeng [2]. Up till now all the researchers have analyzed the PFHS and PCPFHS for horizontal orientation but this project aims to find the heat transfer enhancement of PFHS and PCPFHS for its vertical orientation at different range of heat fluxes and Reynolds's number.

Keywords— **Plate fin heat sink (PFHS), Plate circular pin fin heat sink (PCPFHS), Vertical orientation, Forced convection**.

ARTICLE INFO

Article History

Received :18th November 2015

Received in revised form :

19th November 2015

Accepted : 21st November , 2015

Published online :

22nd November 2015

I. INTRODUCTION

Heat transfer enhancement is an active and important field of engineering research since improvement in the effectiveness of heat sinks through suitable heat transfer augmentation techniques can result in considerable technical advantages and savings of costs. Considerable enhancements were demonstrated in the present work by using small cylindrical pins in the passage of plate fins as compared to the

plate fin heat sinks. Planted pin fins acts as a turbulence generator by disturbing the laminar boundary layer growth

in the plate fin passage. As turbulence mixes the fluid, in the case of a hot surface and a cold fluid, turbulence in the boundary layer brings cold fluid from the bulk flow near to the wall and takes hot fluid from near the wall and transports it away into the bulk flow where it mixes with cold bulk fluid. Both these aspects of the mixing process promote heat transfer. Conversely in laminar flow, the flow is essentially parallel to the surface, so the heat transfer mechanism is just laminar conduction through the boundary layer and into the bulk. Experiments are reported that demonstrate the effectiveness of the results of the proposed approach. It is shown that the suggested method of heat transfer enhancements is much more effective than existing methods,

since it results in an increase in heat transfer area (like fins) and also an increase in the heat transfer coefficient. Plate fin heat sinks (PFHS) are widely used in electronic equipment cooling because of their many advantages, such as easy machining, simple structure, and low cost. Various forms of PFHS are manufactured and supplied to markets in large quantity and they can achieve excellent solutions for many thermal issues in electronic equipment. Many publications studied the optimization of the PFHS and attempted to define general rules for optimizing it. All these researches focused on optimization of design parameters and the operating condition of a cooling system. However, there exists an intrinsic shortcoming in structures of PFHS, i.e. parallel plate fins make airflows passing through heat sinks smoother. This is undesirable for enhancing heat transfer performances of heat sinks. In some cases, it will be helpful to reduce the cost of the total electronic device, if users can modify the heat sink structure and improve heat transfer performance of it by themselves. Based on the reasons mentioned above, an idea of how to make the flow in a PFHS more turbulent and heat transfer performance of the heat sink stronger was formed, and a plate circular pin fin heat sink (PCPFHS) was constructed from an existing PFHS and analysed for its performance in forced convection in vertical orientation.

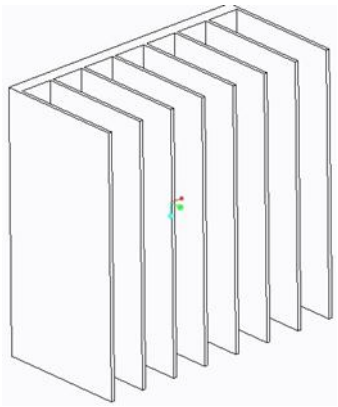

Fig.1 Vertical Plate Fin Heat Sinks (PFHS)

NOMENCLATURE

Q_{in} : Heat input, W
$\dot{Q}$: Heat transfer rate, W
l : Characteristic length
P : Density of air.
V : Velocity of air
μ : Dynamic viscosity
H : Fin height, m
L : Fin Length, m
S : Fin spacing, m
T : Temperature, K
H : Heat transfer coefficient , $W/m^2.K$
t : Fin thickness, m
D_h : Hydraulic diameter
h_{av} : Average heat transfer coefficient, $W/m^2.K$
Re : Reynolds number = $\rho \cdot V \cdot D_h/\mu$
Nu : Nusselt number = $h_{av} \cdot l/K$
K : Conductivity of air.
Γ : Diffusion coefficient of air

Abbreviations

PFHS : Plate Fin Heat sinks
PPFHS : Plate Pin Fin Heat Sinks

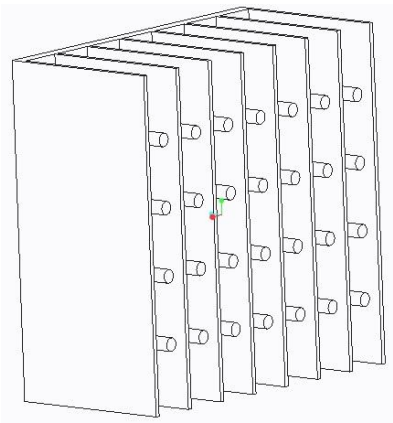

Fig.2 Vertical Plate Circular Pin Fin Heat Sinks (PCPFHS)

II. PREVIOUS RESEARCH

Avram Bar-Cohen and Warren M. Rohsenow [1] conducted one of the earliest studies about the heat transfer performance of rectangular fin arrays. In their experiments, the preceding development of analytic relations for the optimum spacing between printed circuit boards or cards, modeled as thick isothermal or isoflux surfaces, reveals significant departures from values associated with negligibly thick elements. Xiaoling Yu, JianmeiFeng, Quanke Fen and Qiuwang Wang [2] proposed a special solution for improving heat transfer performance of a PFHS by planting some columnar into flow passages of the PFHS to disturb airflows passing through the heat sink. So a PPFHS was constructed. Numerical simulation and experimental results show that the thermal resistance of a PCPFHS is 30% lower than that of a PFHS used to construct the PCPFHS with the same blowing velocity, and the profit factor of the former is about 20% higher than that of the latter with the same pumping power. N. Sahiti, F. Durst and A. Dewan [3] made an attempt to improve the performance of heat exchangers by using pin fins was undertaken. Kai-ShingYang , Wei-Hsin Chu , Ing-Yong Chen , Chi-Chuan Wang [4] performs an experimental study of pin fin heat sinks having circular, elliptic, and square cross-section. A total of twelve pin fin heat sinks with inline and staggered arrangements were made and tested. The effect of fin

density on the heat transfer performance is examined. For an inline arrangement, the circular pin fin shows an appreciable influence of fin density whereas no effect of fin density is seen for square fin geometry. This is associated with the unique deflection flow pattern accompanied with the inline circular fin configuration. For the staggered arrangement, the heat transfer coefficient increases with the rise of fin density for all the three configurations.

III.EXPERIMENTAL SETUP

The experimental setup mainly consists of flow control valve, u-tube manometer, 2 hp blower, honey comb mesh for streamed flow over fin pattern, duct, fin pattern, heater assembly and control panel. The fins are made from aluminum having conductivity of 202 W/Mk. The rectangular fins are cut from 2mm aluminum fin and the fin base is cut from 6mm aluminum fin. The pin fins are then mounted between the rectangular fin passage with the help of screw and the contact surface are covered with thermal solution to reduce contact resistance. Three cartridge heater were placed in the holes drilled in the base portion of the fin array. Heaters were connected to the mains via a dimmerstat. A calibrated wattmeter was connected to measure the heater input. A bakelite (0.233 W/Mk) sheet which acts as an insulator were placed to cover the back of the heater which facilitate the heat to travel only in the fin array direction. Thermocouple (copper-constantan) were used to measure the temperature at various locations on the fin array and the surroundings.

IV.EXPERIMENTAL PROCEDURE

Six different set of rectangular plate fin and rectangular plate circular pin fin array were tested for different Reynolds numbers for turbulent flow through non circular duct ranging from 4500 to 10,000 at different heater input starting from 50 W, 75W.100W.125W and 150W.Radiation losses and contact losses are calculated to find out exact amount of heat loss through fin array due to convection

Fig 3. Experimental setup

V.NUMERICAL MODEL

The schematic diagram of PFHS and PPFHS are shown in Fig. 1 and 2. Geometric parameters of both the heat sinks are listed in Table 1. The different sets of PFHS and PPFHS are taken for computational analysis. As the number of plate fins increase, the fin spacing between the rectangular fins decreases. The pin fins are centrally aligned in the passage of plate fin as the heat transfer rate of inline arranged pin fin is maximum as compared to staggered fins[4]. The Pin fins are equally spaced along the length of heat sink and its density is varied from 4 to 8 pin fins in the passage. With the rapid development in the field software technology, **CFD** (**C**omputational **F**luid **D**ynamics) is becoming a popular tool for analysis. The major advantage of CFD analysis is the use of parametric programming, as it is not feasible to have experimental setups for all the different geometries. This saves time and cost of experimentation. In the present paper, an attempt is made to study and analyze forced convection heat transfer from vertical rectangular plate fin array and plate fin pin fin array using CFD analysis. The results are compared with the experimental and numerical work done earlier.

Table 1 : Parameters of Numerical Study

Parameter	SET 1	SET 2	SET 3	SET 4	SET 5	SET 6
Area of base plate (mm^2)	22500	22500	22500	22500	22500	22500
Length of Fin Array (mm)	150	150	150	150	150	150
Height of Fin Array (mm)	70	70	70	70	70	70
Fin Spacing (mm)	15	15	20	20	28	28
Pin Fin Diameter(mm)	-	6	-	15	-	10
No of Plate Fin	8	8	7	7	6	6
Arrangement of Pin Fin	In Line and Central					

The computational domain and the coordinate system are illustrated in Fig. 4. The flow passage between two rectangular fins in each heat sink was selected as the computational domain. In order to ensure no back flow in outlets, the computation domains cover 1L downstream of the fin arrays in negative z direction[2]. Two sides of the computational domains in x direction are the center planes (symmetry planes) of plate fins. The flow is assumed as three-dimensional, turbulent, incompressible, and steady flow. Buoyancy and radiation heat transfer effects are negligible. Thermodynamic properties are assumed to be constant. The k–ε turbulence model is used to describe the characteristics of air flow through heat sinks [5]. Assuming that the flow variables can be written in the form $f = \bar{f} + f'$ where $\bar{f}$ is the mean value and f' is a fluctuation about the mean.

VI.GOVERNING EQUESTION

The forced convection flow was modeled by a set of partial differential equations describing the conservation of mass, momentum and energy in three rectangular Cartesian coordinate directions

- Conservation of mass:

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

- Conservation of momentum:

$$\frac{\partial(\rho u^2)}{\partial x}+\frac{\partial(\rho uv)}{\partial y}+\frac{\partial(\rho uw)}{\partial z}=-\frac{\partial P_m}{\partial x}+\rho v\left(\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 u}{\partial y^2}+\frac{\partial^2 u}{\partial z^2}\right) \qquad (2)$$

$$\frac{\partial(\rho vu)}{\partial x}+\frac{\partial(\rho v^2)}{\partial y}+\frac{\partial(\rho vw)}{\partial z}=-\frac{\partial P_m}{\partial y}+\rho v\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right) \qquad (3)$$

$$\frac{\partial(\rho wu)}{\partial x}+\frac{\partial(\rho wv)}{\partial y}+\frac{\partial(\rho w^2)}{\partial z}=-\frac{\partial P_m}{\partial z}+\rho v\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right) \qquad (4)$$

- Conservation of Energy :

$$\frac{\partial(\rho uT)}{\partial x}+\frac{\partial(\rho vT)}{\partial y}+\frac{\partial(\rho wT)}{\partial z}=\frac{v}{Pr}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}\right) \qquad (5)$$

The above governing equations can then be solved for our given condition and the obtained equation can be written as

$$\frac{\partial}{\partial x_i}(\rho\bar{u}_j)=0 \qquad (6)$$

$$\frac{\partial}{\partial x_i}(\rho\bar{u}_j\bar{u}_i)=-\frac{\partial\bar{P}}{\partial x_i}+\frac{\partial}{\partial x_j}\left(\frac{\partial\bar{u}_i}{\partial x_i}+\tau_{ij}\right) \qquad (7)$$

whereτ_{ij} is the Reynolds stress term given by

$$\tau_{ij}=\mu_t\left(\frac{\partial\bar{u}_i}{\partial x_j}+\frac{\partial\bar{u}_j}{\partial x_i}\right)-\frac{2}{3}\rho\delta_{ij}k \qquad (8)$$

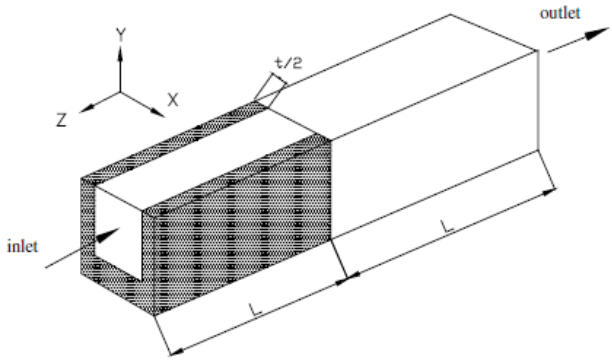

Fig. 4 Computation domain for PFHS [2]

Where,μ_t is the turbulent viscosity and $k=\frac{1}{2}(\overline{u'^2}+\overline{v'^2}+\overline{w'^2})$is the turbulent kinetic energy. The energy equation solved for the fluid flow is $\bar{u}\frac{\partial\bar{T}}{\partial x_i}=\frac{\partial}{\partial x_i}\left(\Gamma\frac{\partial\bar{T}}{\partial x_i}-\overline{T'u'}\right)$where $\Gamma$is thediffusion coefficient of air. The energy equation solving conduction heat transfer within the heat sink is

$$\frac{\partial}{\partial x_i}\left(\lambda_s\frac{\partial T_s}{\partial x_i}\right)+\dot{q}=0$$

where$\dot{q}$ is the heat generated per unit volume of the heat sink, λ_s is the heat sink thermal conductivity and Ts is the temperature within the heat sink. The solution algorithm used in this paper is the SIMPLE method described in [5]. The discrete schemes of convection items are QUICK scheme with three-order precision. The flow velocity, k and ε were specified at the inlet, and they were assumed to have zero gradient in the z direction at the outlet. Non-slip boundary condition for velocity was provided on walls. The standard wall functions were used to treat near wall domain. Uniform heat flux condition was applied on the bottom surface of fin base. On the symmetric planes, adiabatic conditions were provided to the velocity u, k, and ε. Grid structures were fine enough to give results independent of grids. Properties of the working fluid are the same as those of air at 294 K, and the material of heat sinks is aluminum with thermal conductivity of 202 W/(mK).

VII.NUMERICAL BOUNDRY CONDITION

All boundary conditions were implemented by the inclusion of additional source and/or sink terms in the finite volume equations for computational cells at the boundaries. In forced convection flows the velocity at the inlet is known as it can be calculated by height of water column in the manometer. Again the heater input in watt is known which is applied to the base of the fin array. Pressure at outlet is atmospheric. Since governing equations are invariably coupled, the temperature field causes the velocity field to develop and in turn the velocity field affects the temperature field with the promotion of convective heat transfer. Figure 3 shows the plate fin array under investigation with the coordinate system used and the relevant dimensions. The computational domain is also shown on this figure. Because only one fin channel is investigated and because of symmetry conditions only one quarter of this fin is simulated.

VIII. RESULTS, COMPARISONS AND DISCUSSION

The six different set of Plate fin heat sink (PFHS) and Plate pin fin heat sink (PPFHS) with different fin spacing and pin density in this study are listed in Table 1. The effects of the wind velocity and the types of PPFHS on the thermal performance are investigated. The parameters used in this study include U_{in} (U_{in} = 0.5, 0.6,0.7,1.0 and 1.20 m/s), the types of pins (Type-1–Type-6, containing 3 types of PFHS and 3 types of PCPFHS) and the arrangements of pins (in-line). In the PCPFHS, the center distance of two neighbor pins is set such that all the pins are equally spaced. The fin base plate area is heated with heating power ranging from 50W to 150W in step size of 25 W.The mechanism of the flow in this type of fin array is complicated; hence, it is difficult to obtain physically meaningful solutions. Therefore, before the start of the parametric study, trial simulations were conducted for the verification of the computational domain and numerical procedures applied.Figure 11 compares present results with experimental data and computational results by XiaolingYu[3]. In this case, the length of the fin is 51 mm, height 10 mm, and fin number of 9 , fin thickness of 1.5 mm and pin in very good agreement with experimental values and numerical results of Xiaoling Yu[3].].The maximum difference between experimental values of Xiaoling Yu [3] and present numerical values is 10.0% between numerical values of Xiaoling Yu [3] and present study is 11.7%.The six different sets that are taken for experimentation have same height and length, but are of different spacing and pinfin density. Out of these sets three are Plate Fin Heat Sinks (PFHS) and other three arePlate Circular Pin Fin Heat Sinks (PCPFHS), having rectangular fin density ranging from 6 to 8 and pin fin density ranging from 20 to 28 with centrally aligned arrangement. The analysis is carried out by considering three different variables that are, Reyndols number, heat flux i.e applied to the base of fin array and different fin arrangements. All the six sets are analysed at different values of Reynolds number (ranging from 4000 to 10000) and different heat flux (50 W to 150 W) and the heat transfer coefficient is calculated. As it is seen from figures(Fig. 4a to 9b), by planting pin in the passage of plate fin the heat transfer rate increases as compared to the plate finarray. But by increasing the pin sizethere seems to be decrease in heat transfer rate for same fin spacings. This is because of chocking of fin passage between the rectanguar fin by pin fins.For lesser diameter and higher density of fin between the rectangular passage,there is increase in heat transfer rate.This is because planted pin fins disrupt the thermal boundary layer growth that occurs over the fin wall , which makes the flow turbulant, and facilates the increase in heat transfer rate. From the experiment it is observed that, for the same rectangular fin passage, if we go on increasing the diameter of planted pin fin the heat transfer rate starts decreasing.So there is a need to find out the optimum value of pin fin density and pin fin dimensions at which there is maximun heat transefer rate for particular pattern. In this study the six differnet sets of fin pattern areanalysed computaionally and experimentally validatedusing ICEM-CFD softwere for mesh generation and ANSYS fluent software is used to carry out simulations,

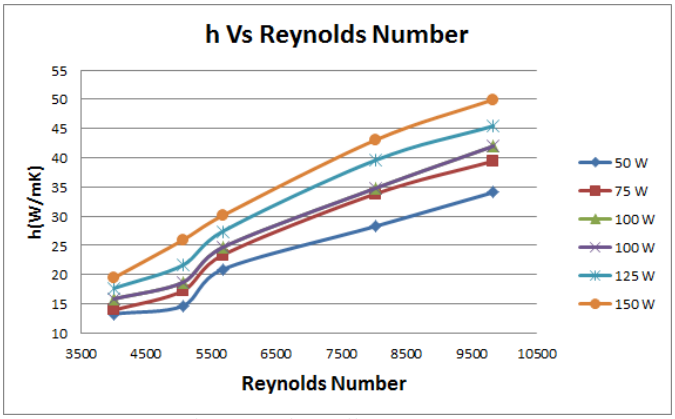

Fig 5a. Variation of heat transfer coefficient with Reynolds number at different heat input(PFHS SET-1)

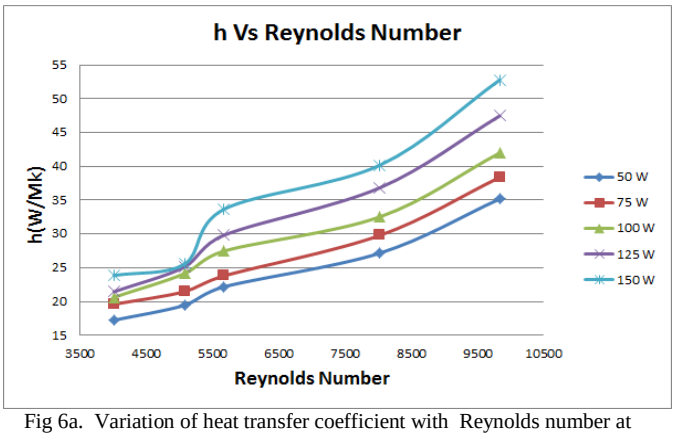

Fig 6a. Variation of heat transfer coefficient with Reynolds number at different heat input(PCPFHS SET-2)

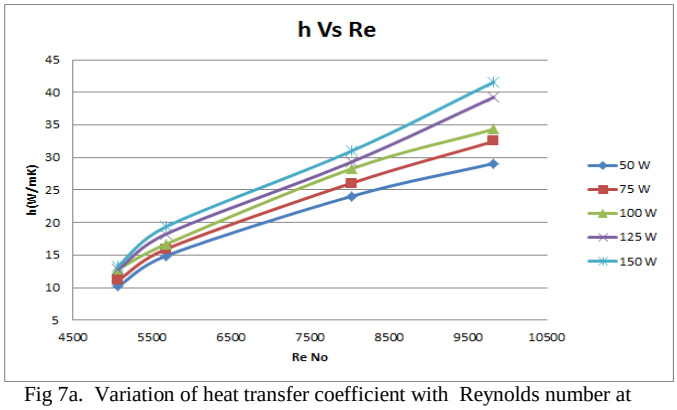

Fig 7a. Variation of heat transfer coefficient with Reynolds number at different heat input(PFHS SET-3)

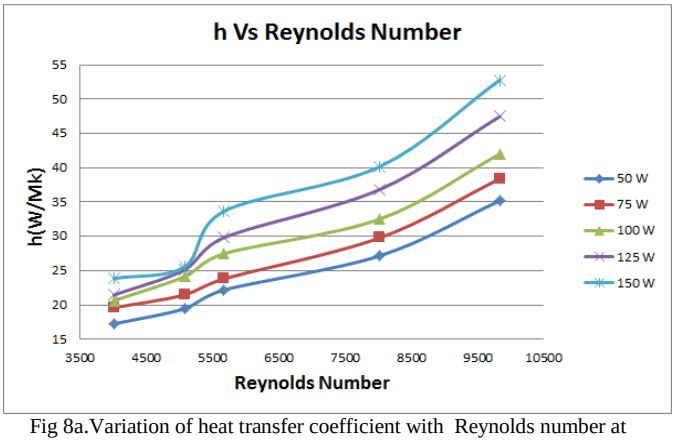

Fig 8a.Variation of heat transfer coefficient with Reynolds number at different heat input(PCPFHS SET-4)

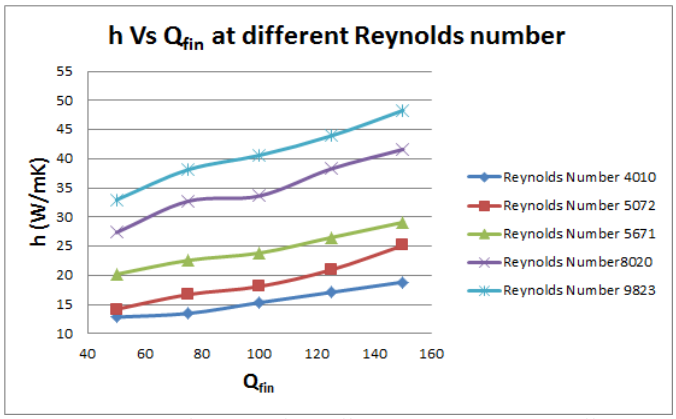

Fig 5b. Variation of heat transfer coefficient with heat input at different Reynolds number(PFHS SET-1)

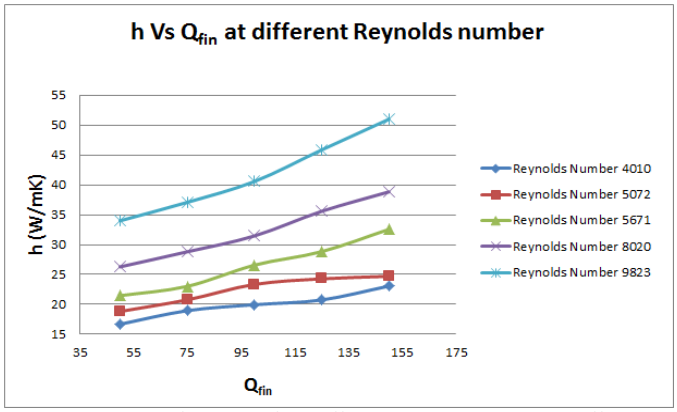

Fig 6b. Variation of heat transfer coefficient with heat input at different Reynolds number(PCPFHS SET-2)

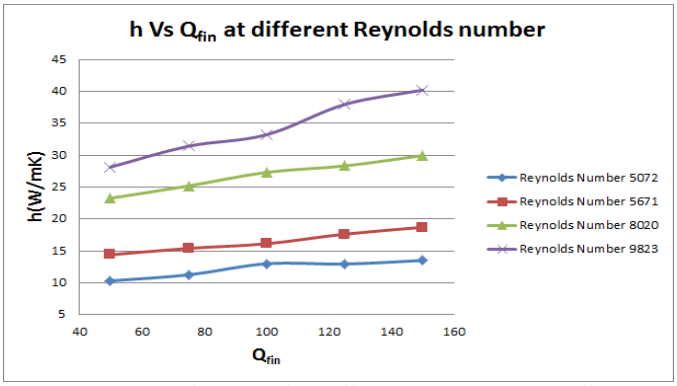

Fig 7b. Variation of heat transfer coefficient with heat input at different Reynolds number(PFHS SET-3)

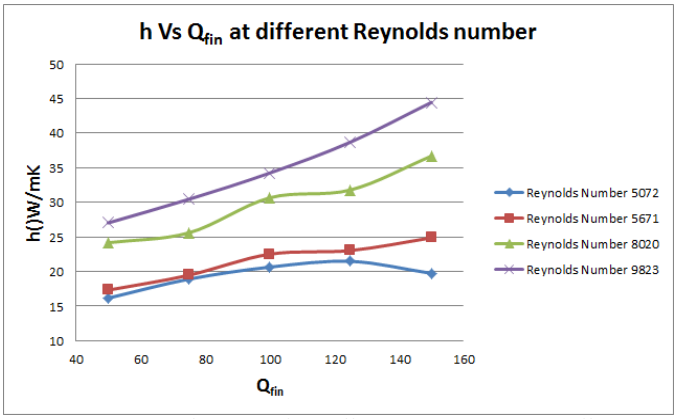

Fig 8b. Variation of heat transfer coefficient with heat input at different Reynolds number(PCPFHS SET-4)

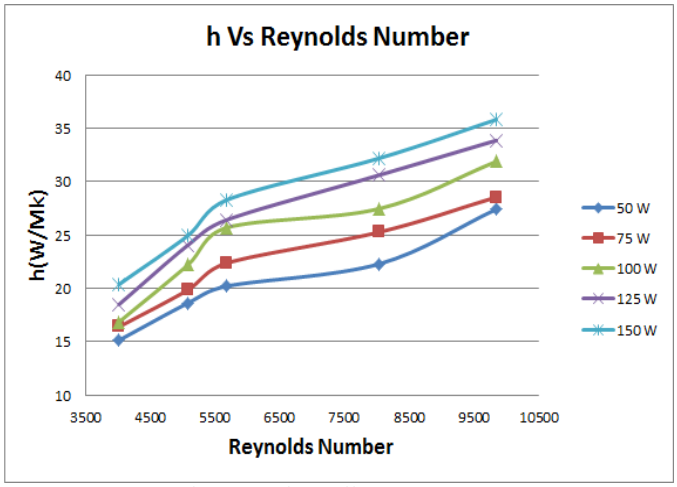

Fig 9a. Variation of heat transfer coefficient with Reynolds number at different heat input(PFHS SET-5)

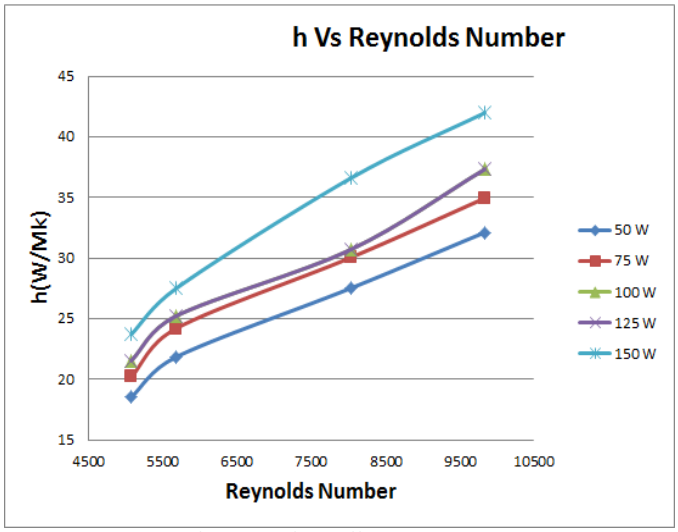

Fig 10a. Variation of heat transfer coefficient with Reynolds number at different heat input(PCPFHS SET-6)

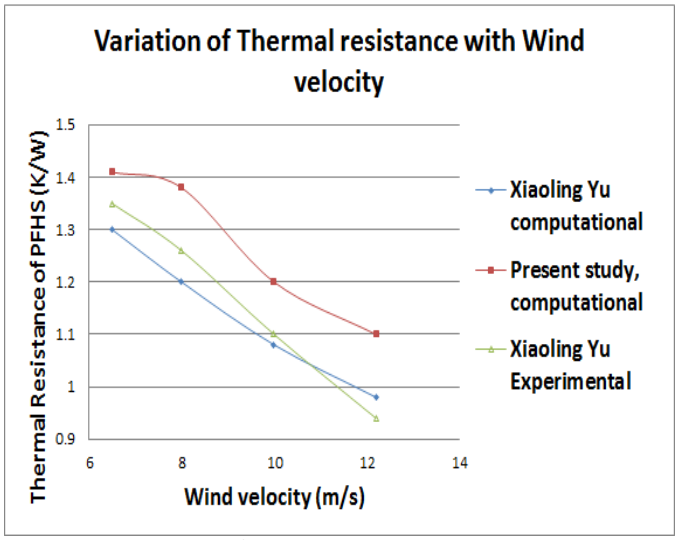

Fig 11.Variation of Thermal resistance with wind velocity

IX. COMPUTATIONAL DOMAIN AND MATHEMATICAL MODEL

The computational domain considered in the present study is the same as that presented by Sahiti*et al.* [18] and

Dewan*et al.*[19]. A three-dimensional rectangular duct of size

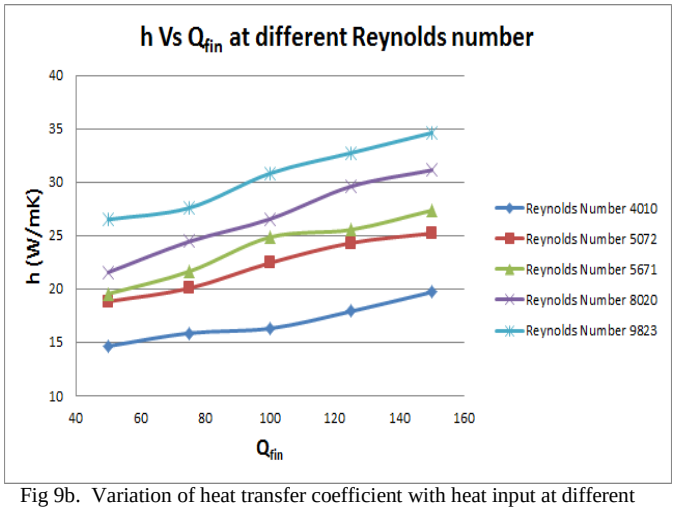

Fig 9b. Variation of heat transfer coefficient with heat input at different Reynolds number(PFHS SET-5)

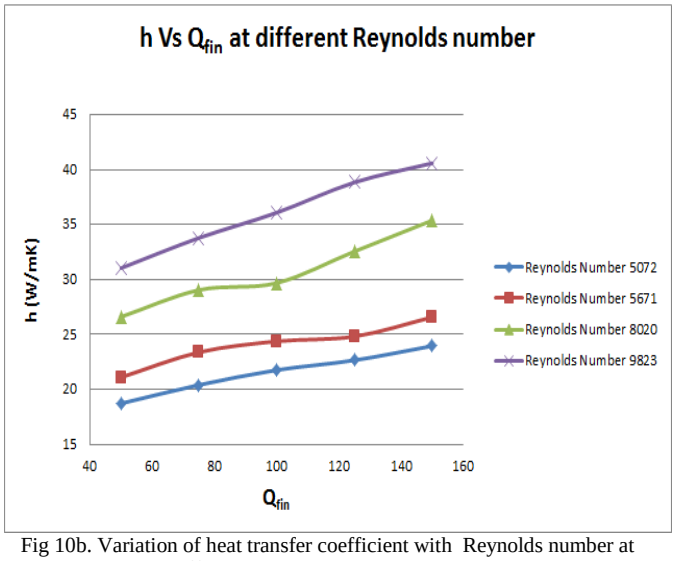

Fig 10b. Variation of heat transfer coefficient with Reynolds number at different heat input(PCPFHS SET-6)

880.0×100.0×200.0 mm^3 is considered. An in-line array of pin fins mounted on the heated bottom wall maintained at different range of heat input starting from 50W to 150W with step size of 25W is considered. The inline arrangements with 4 rows of pins areconsidered in the present work. For the flow to be fully developed, the inlet block length was taken as $5D_h$ to avoid the influence of backflow streams and the outlet block length was assumed to be $15D_h$[18]. Here D_h denotes the hydraulic diameter. The unidirectional flow was assumed at the inlet with a constant atmospheric temperature. The computational domain used in the present work for solid pin fins is shown in Fig. 1 (here D_h ~0.133m). Two symmetry planes are considered through the middle of the fins of two consecutive rows. The rectangular box minus the solid fins is the fluid domain considered in the present work. The fluid was chosen to be air. Aluminum was chosen to be the fin material. The computational domain for the plate circular pin fins was the same as that for solid rectangular plate fins with solid pin inserted in between rectangular fins. The height of the pin fins was 70.0 mm and diameter of fin was 6.0mm, 10.0mm, 15.0mm. The pin fins are equally spaced aligned with the direction of the flow, as shown in Fig. 12. Usually the flow with Reynolds number less than 4000 is considered to be laminar. However in flow over tube banks (such as the one considered in the present study) large scale vortices are present in the recirculation region behind pins and these large scale vortices make the flow into transition regime[23]

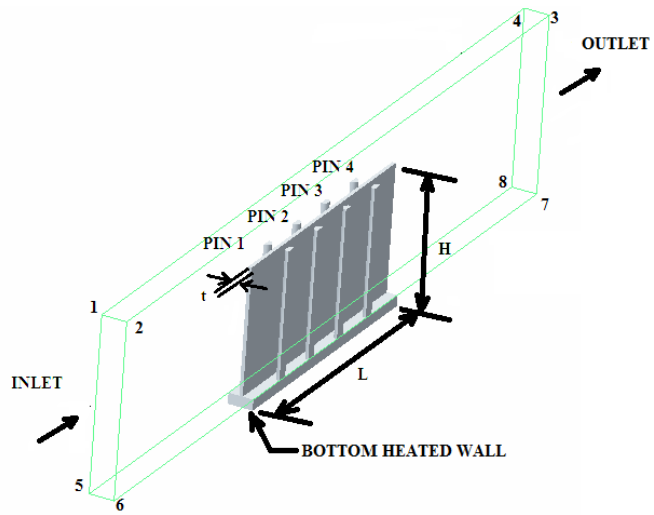

Fig 12. Computational domain for solid plate circular pin fins heat sink (PCPFHS).

X.ASSUMPTION

The assumptions considered due to complexity of flow and to simplify the analysis are: (a) The fin tips are adiabatic; (b) There is no airflow bypass;(c) The approaching airflow is normal to the pin axis; (d) The approach velocity is uniform for each row in a heat sink; (e) The flow is steady; (f) Radiation heat transfer is negligible; (g) The fluid is considered incompressible with constant properties; and (h) Body forces are negligible. Most assumptions used are in accordance with the conditions encountered in the practical situations, except the one of steady flow. The flow was assumed to be steady mainly because engineers are primarily interested in steady behavior. In addition, symmetry boundary condition was applied in the computational domain. This assumption reduces the computational effort and is consistent with that employed in the literature[18]

XI. COMPUTATIONAL BOUNDARY CONDITIONS

The inlet (velocity inlet), outlet (pressure outlet), wall and symmetry boundary conditions were applied in the computational domain. The boundary conditions referring to Fig. 12 are

(a) For the inlet section 1-2-6-5:

$u(0, y, z) = u_{in}$, $v(0, y, z) = 0$, $w(0, y, z) = 0$ and $(0,y,z) = T_{in}$ 293K

(b) For the bottom heated wall (for Q_{in}=50W):

$u(x, y,0) = v(x, y,0) = w(x, y,0) = 0$ and $T(x, y, 0)=T_w= 334K$

(c) For the section 5-6-7-8:

$u(x, y,0) = v(x, y,0) = w(x, y,0) = 0$ and $\left(\frac{dT}{dZ}\right)_{(x,y,0)} = 0$

(d) The top wall 1-2-3-4 was considered to be adiabatic, where the no slip condition for the velocity components was applied and in the outlet section 4-8-7-3, the pressure outlet boundary condition was used.

XII.COMPUTATIONAL CODE

The commercial finite volume code FLUENT 14.5.0 was used to solve the governing equations. The second order upwind scheme was used to discretize the governing equations. The RNG k-εturbulence model[21] with standard wall functions was used. The SIMPLE algorithm was used to relate velocity and pressure corrections to enforce mass conservation and to obtain the pressure field.

XIII. COMPUTATIONAL MESH

The unstructured meshes were used for all the geometries. The prismatic volume mesh was used for the solid circular pin fin geometries and tetrahedral volume meshes for the perforated pin fin geometries. A prismatic volume mesh was obtained from the triangular meshes with pave scheme on the surfaces and Hex/Wedge element with cooper scheme on the volume.

XIV. CONVERGENCE AND GRID INDEPENDENCE STUDIES

In the present work, two different convergence criteria of 10^{-5} and 10^{-6}were used and it was observed that the numerical values of the flow parameters corresponding to two different convergence criteria were almost identical. The grid independence study was performed for solid Rectangular plate fin heat sink and Rectangular plate circular pin fins heat sink (Figs. 13 and 14) by using the RNG k-ϵ model with the standard wall functions for the inlet velocities ranging from 0.5 m/s to 1.2 m/s. From grid independence study (using different types of mesh, namely, MT1, MT2, MT3, and MT4, each having different number of cells) mesh with 323424 cells for solid Rectangular plate fin structure and mesh count with 111526 for Rectangular plate circular pin fin structure was used for further simulations. The validation of the present computational model was performed by comparing the present computations for solid rectangular platfins with theexperimental data of XiaolingYu[2] for solid plate finand an average 10.42% deviation was observed (Fig. 11).

XV. GLOBAL NUSSELT NUMBER AND HEAT TRANSFER

The global Nusselt number was calculated for different inlet velocities ranging from 0.5 m/s to 1.2 m/s. It is seen from Fig. 15 that the global Nusselt number of plate circular pin fin heat sink (PCPFHS) is larger than that of the solid rectangular plate finsheat sink (PFHS) at the same Reynolds number.This behavior is because of the insertion of solid circular pin in the passage of plate fin which provide more wetted surface area for heat transfer compared to that of PFHS and also create turbulence by disturbing the boundary layer growth which increases the heat transfer rate.The Nusselt number is seen to increase with an increase in Reynolds number for both cases. This is due to the fact that a higher fluid inlet velocity leads to larger convective heat transfer resulting in a larger increase of heat transfer rate with increasing Re. Heat transfer is high forPCPFHS compared to that for the solid PFHS due to increase in Nusselt number and wetted surface area.

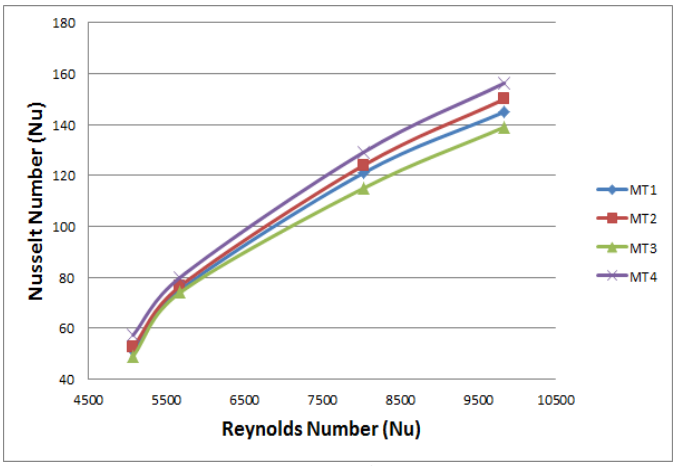

Fig 13.Grid independence study for solid PFHS.(SET-3)

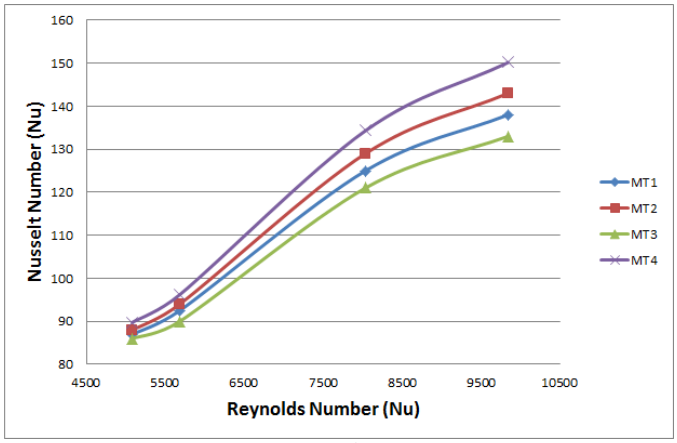

Fig 14.Grid independence study for solid PCPFHS. (SET-4)

XVI. CONCLUSIONS

In the present paper the effect of insertion of solid circular pin fins in the passage of plate fins in inline manner on turbulent forced convective heat transfer in a 3D channel been experimentally and computationally examined. The study shows that plate circular pin fins considerably enhanced the heat transfer for the range of Reynolds number considered in the present study. The use of circular pin fin although increase the weight and the cost but the enhancement in heat transfer is considerably enough to compensate the cost of manufacturing. Therefore it can be concluded that the use of plate circular pin heat sink is beneficial in the fins of air cooled engine as well as in many other industrial applications wherever there is a necessity of faster heat removal.

It can be concluded from fig 15, fig 16, fig 17, fig 18, fig 19 that fin spacing and pin density plays an important role in increasing the coefficient of heat transfer. Selecting larger pin diameter or pin diameter not always

going to increase heat transfer rate. Selecting larger pin diameter may cause chocking of flow which reduces complete contact of air with fin surface thus reducing heat transfer rate. Increasing fins spacing not always going to increase heat transfer rate. Hence for achieving higher heat transfer rate for PCPFHS a proper combination of pin density and fins spacing should be made.

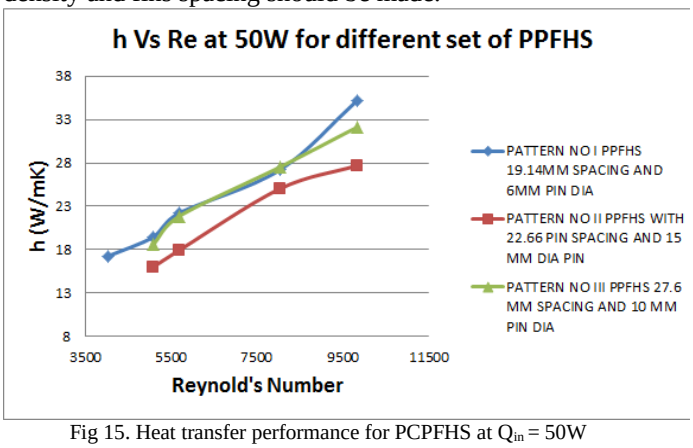

Fig 15. Heat transfer performance for PCPFHS at Q_{in} = 50W

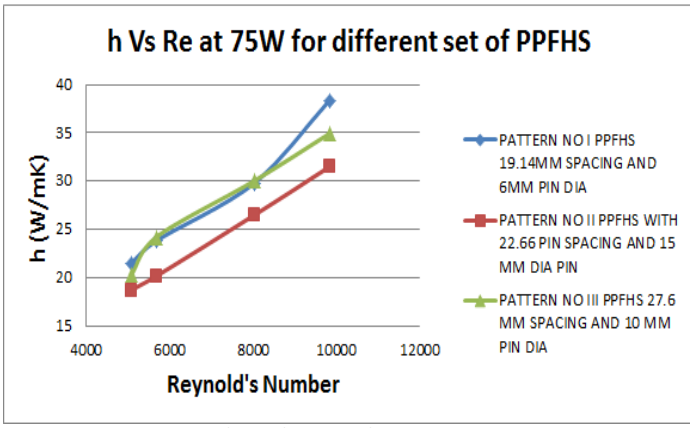

Fig 16. Heat transfer performance for PCPFHS at Q_{in} = 75W

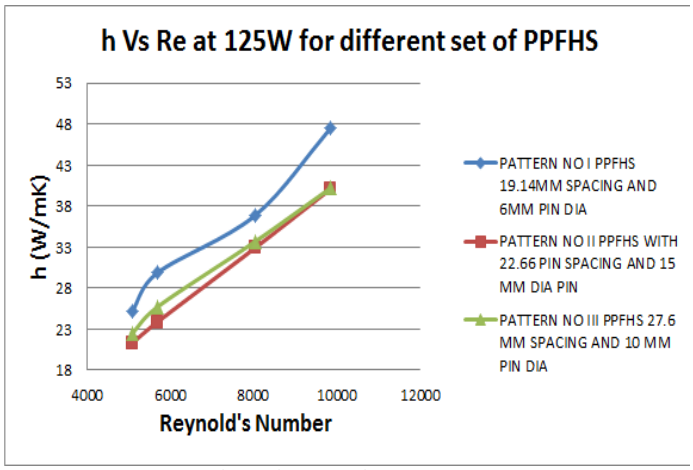

Fig 17. Heat transfer performance for PCPFHS at Q_{in} = 125W

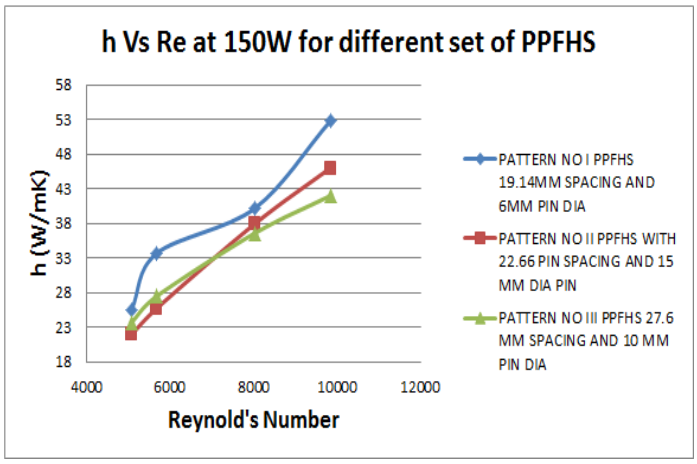

Fig 18. Heat transfer performance for PCPFHS at Q_{in} = 150W

ACKNOWLEDGEMENTS

Gaurav Kamde gratefully acknowledges the financial support received from BCUD, ShriSavitribaiPhule Pune University through the research grant to carry out the research work reported in this paper and MITCOE Mechanical staff for its technical support.

REFERENCES

[1]. Avram bar-cohen and warren m. Rohsenow, , "Thermally optimum arrays of cards.and fins in natural Convection", IEEE transactions on components, hybrids, and manufacturing technology, vol. Chmt-6, no. 2. june1983

[2]. Xiaoling Yu, JianmeiFeng, QuankeFeng And Qiuwang Wang, "Development Of A Plate-Pin Fin Heat Sink And Its Performance Comparisons With A Plate Fin Heat Sink", Applied Thermal Engineering, Elsevier.Vol 25,Pp 173–182,2005,

[3]. N. Sahiti, F. Durst , A. Dewan, "Heat Transfer Enhancement By Pin Elements", International Journal Of Heat And Mass Transfer Elsevier,Vol 48,Pp 4738–4747, 2005.

[4]. Kai-Shing Yang A, Wei-Hsin Chu B, Ing-Yong Chen B, Chi-ChuanWang ,"A Comparative Study Of The Airside Performance Of Heat Sinks Having Pin Fin Configurations", International Journal Of Heat And Mass Transfer, Elsevier, Vol 50,Pp 4661–4667, 2007.

[5]. R.F. Babus'Haq, K. Akintunde, S.D. Probert, "Thermal performance of a pin fin assembly", Int. J. Heat Fluid Flow 16 ,50–55, 1995.

[6]. Yue-Tzu Yang, Huan-SenPeng, "Investigation Of Planted Pin Fins For Heat Transfer Enhancement In Plate Fin Heat Sin", Microelectronics Reliability,Elsevier.Vol 49 Pp 163–169, , 2009 .

[7]. Jiazeng Liu1, Jianmin Gao1 and TieyuGao, "Forced convection heat transfer of steam in a square ribbed channel", Journal of Mechanical Science and Technology, Springer, 26 (4) 1291~1298, , 2012.

[8]. MeysamAtashafrooz and SeyyedAbdolrezaGandjalikhanNassab, "Combined heat transfer of radiation and forced convection flow of participating gases in a three-dimensional recess", Journal of Mechanical Science and Technology 26 (10), 3357~3368,2012.

[9]. Swee-Boon Chin , Ji-Jinn Foo , Yin-Ling Lai •Terry Kin-Keong Yong, "Forced convective heat transfer enhancement with perforated pin fins", Heat Mass Transfer , 49:1447–1458.

[10]. Kays,W. M., "Pin fin heat exchanger surfaces". Trans. ASME, 77, 471–483,2013.

[11]. Dewan, A., Mahanta, P., SumithraRaju, K., and Suresh Kumar, P, "Review of passive heat transfer augmentation techniques". Proc. IMechE, Part A: J. Power and Energy, , 218(A7), 509–527,2004.

[12]. Maudgal, V. K. and Sunderland, J. E., "An experimental study of forced convection heat transfer from inline pin fin arrays". In Proceedings of the 13th IEEE Semi-Therm Symposium, Austin, Texas, USA, pp. 149–157, 1997.

[13]. Behnia, M., Copeland,D., and Soodphakdee, "A comparison of heat sink geometries for laminar forced convection: numerical simulation of periodically developed flow". In Proceedings of the InterSociety Conference on Thermal phenomena, Seattle, Washington, USA, pp. 310–315,1998.

[14]. Leon, O., Mey, G. D., Dick, E., and Vierendeels, J, "Comparison between the standard and staggered layout for cooling fins in forced convective cooling". J. Electron. Pack., 125, 442–446, 2003.

[15]. Short, Jr, B. E., Price, D. C., and Raad, P. E. , "Design of cast pin fin coldwalls for air-cooled electronics systems". J. Electron. Pack. 126, 67–73, 2004..

[16]. Kobus, C. J. and Oshio, T, "Development of a theoretical model for predicting the thermal performance characteristics of a vertical pin-fin array heat sink under combined forced and natural convection with impinging flow". Int. J. Heat Mass Transf,48, 1053–1063,2005,.

[17]. Tahat, M., Kodah, Z. H., Jarrah, B. A., and Probert, S. D, "Heat transfers from pin-fin arrays experiencing forced convection". Appl. Energy, 67, 419–442, 2000.

[18]. Sahiti, N.,Durst, F., and Dewan, , "A. Strategy for selection of elements for heat transfer enhancement". Int. J. Heat Mass Transf,, 49, 3392–3400, 2006.

[19]. Sahiti, N., Durst, F., and Dewan, A,,"Heat transfer enhancement by pin elements". Int. J. Heat Mass Transf., 48, 4738–4747,2005.

[20]. Sparrow, E. M., Ramsey, J. W., and Altemani, C. A. C., "Experiments on inline-pin fin arrays and performance comparison with staggered arrangement". ASME J. Heat Transf., 102, 44–50,1980.

[21]. Choudhury D., "Introduction to theRenormalization Group Method and Turbulence Modeling, Fluent Inc", Technical Memorandum TM-107, 1993.

[22]Dewan, A., "Tackling Turbulent Flows in Engineering", First edition, Springer, Berlin, Germany, p. 68, 2011.

[23]. Bergelin. P., Brown G.A. and Doberstein.S.C., "Heat transfer and fluid friction during flow across banks of tubes - IV. A study of the transition zone between viscous and turbulent flow", Trans. ASME 74, 953-960,1952.