International Journal of Engineering & Scientific Research
Vol.4 Issue 11, November 2016,
ISSN: 2347-6532 Impact Factor: 5.900
Journal Homepage: http://www.ijmra.us, Email: editorijmie@gmail.com
Double-Blind Peer Reviewed Refereed Open Access International Journal - Included in the International Serial Directories Indexed & Listed at: Ulrich's Periodicals Directory ©, U.S.A., Open J-Gage as well as in Cabell's Directories of Publishing Opportunities, U.S.A

THERMAL PERFORMANCE ANALYSIS OF FIN ARRAYS WITH PERFORATION AND CROSS FIN AT THE CENTER

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Abstract

Increasing the heat transfer area enhances the heat dissipation rate, but at the same time increase of resistance to fluid flow causing decrease in heat transfer rate. In order to dissipate the heat of very high heat flux densities, the required heat sink must often be larger than device. Consequently, the heat sink performance is reduced. The inter fin resistance may be reduced by adding the notches or by adding the perforation to the fins.

The present paper reports, the validation of experimental results by modeling and simulation in CFD, on the fluid flow and heat transfer characteristics of a cross fin arrays with circular perforation equipped on horizontal flat surface and its external dimensionally equivalent non perforated fin arrays with and without cross fin. The simulation is carried out using the ANSYS 12.0, fluid flow (CFX) workbench. In this study, results shows that adding a cross-fin at center helps to increase the heat dissipation area but it forms the stagnant layer around the fin array. The fluid flow rate of the cross-fin array can be eased by adding perforation to the fins. The new

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perforated fins have an improvement in convective heat transfer coefficient, over its external dimensionally equivalent non perforated fin array. This is new era for an efficient heat sink design consideration.

Keywords:Heat transfer;CFD simulation;Cross fin;Perforated fins;Heat sink,Natural convection.

1. Introduction

The enhancement of heat transfer is an important subject of thermal engineering. Removal of excessive heat from system components is essential to avoid the damaging effects of burning or overheating. The heat transfer from a surface may in general, be enhanced by increasing the heat transfer coefficient between a surface and its surrounding, or by increasing heat transfer area of the surface, or by both.

With the increase in heat dissipation from electronic devices and the reduction in overall form factors, it became an essential practice to optimize heat sink designs with the least trade-offs in material and manufacturing costs. Heat dissipaters are not only chosen for their thermal performance but also for design parameters like weight, cost and reliability. Depending on the application and different priorities will influence design parameters.

Investigated experimentally [1] heat dissipation from a fin array with an inverted notch at the central bottom portion of the fin to modify its geometry for the enhancement of heat transfer on normal and inverted notched fin arrays (INFAs). They found that the average heat transfer coefficient for INFAs is nearly 30–40% higher as compared with a normal array. Studies on [2] rectangular notched fin arrays reported that total heat flux as well as the heat transfer coefficient increases as the notch depth increases.

Iinvestigation [3] of the plate fin arrays with and without cross fin at centre of the horizontal fin arrays. They noticed that, the average Nusselt number can be improved by introducing a cross-fin at the center of a horizontal array for larger fin spacing. The flow visualization studies [4] of perforated fin arrays and non perforated fin arrays, the flow pattern observed in case of non perforated fin array was a sliding chimney and that of the perforated fin array was of single chimney in nature, a significance of enhanced heat dissipation rate due to perforation. Also they noted that heat dissipation rate increases with the size of the perforation because of more free convection.

The present paper reports [5], an experimental analysis of the fluid flow and heat transfer characteristics of a rectangular fin arrays with and without cross fin at center equipped with circular perforation on a horizontal flat surface and its external dimensionally equivalent non perforated fin arrays.

The geometric parameters (Table 1) considered in this study were the different diameters (4mm, 6mm, 8mm and 12mm) of circular perforations. Geometric dimensions of the perforated fin arrays both with and without cross fin at center and its external dimensionally equivalent non perforated fin arrays were same and that were Height = 65, Length=100 mm, fin spacing = 30mm, fin thickness 3mm and fin base size 132x100x10 (length x breadth x thickness). Repeated steady state readings were taken at different heat inputs. Also, CFD modeling and simulation is carried out using a workbench – fluid flow (CFX) under ANSYS 12.0.

Table 1 Geometry Details of Fin Arrays Tested

Sr. No.	**Type of Fin Array**	**Dia. of Perforation mm**	**Fin Height x thickness mm**	**No. of Holes**	**Surface Area m^2**
1	CRPL - Non Perforated With Cross Fin	-	65x3	0	0.08119
2	CR4PR - Perforated With Cross Fin	4	65x3	128	0.082799
3	CR8PR - Perforated with Cross Fin	8	65x3	128	0.0779717
4	CR12PR - Perforated with Cross Fin	12	65x3	128	0.0667077

5	WCRPL - Non Perforated Without Cross Fin	-	65x3	0	0.06676
6	WCR4PR - Perforated Without Cross Fin	4	65x3	96	0.0679668
7	WCR8PR - Perforated Without Cross Fin	8	65x3	96	0.0643462
8	WCR12PR - Perforated Without Cross Fin	12	65x3	96	0.0558982

2. Experimentation

The primary requirement of the natural convection heat transfer experiment is controlled environmental conditions. The experiment must be away from fans and flow of outside air. Hence to provide natural convection conditions, the experiment were conducted in a room where fan was off and keeping windows closed so that readings should not affect by the outside atmosphere and to provide similar atmospheric conditions for all experiments.

Figure1 shows the schematic of experiment with locations of thermocouple junctions. The supply is taken from mains through stabilizer. The dimmer-stat was adjusted to make the desired supply to the electric heater. Ensured that all thermocouple junctions are intact and functioning properly. The heating was continued till the steady state reached. It took 4 to 5 hours to reach steady state, took only one set of reading very carefully on one day. After reaching the steady state temperature readings were noted. This procedure was repeated for all heater inputs. Observations were repeated to confirm the validity and the readings, obtained were found to be similar.

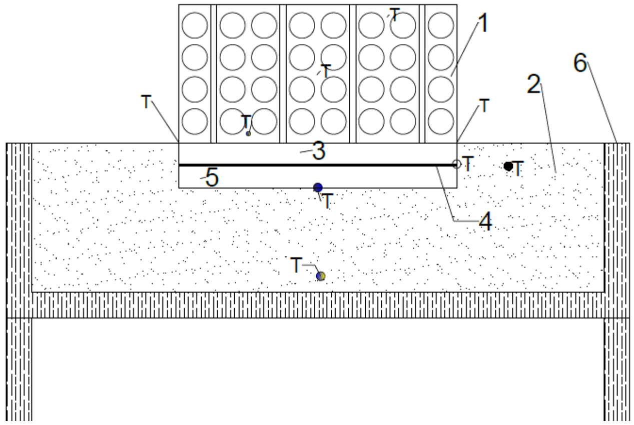


Figure1 Sectional View of experimental set up

1) Fin array 2) Alumina insulating brick 3) Base plate for mounting of fins 4) Heating coil 5) Lower base plate 6) Wooden Frame, T – Some locations Thermocouple junctions

3. CFD Modeling and simulation

Computational Fluid Dynamics (CFD) Modeling and simulation was carried out in an ANSYS 12.0, Workbench environment with the system of fluid flow CFX [6].

Simulation starts with creating the geometry, and it was created using ANSYS Design Modeler available in the workbench. A fluid domain is required around the fin to study mass flow and thus the heat transfer from the fin because the area of interest is the outside of the fin, the interface between air and the fin surface. Hence, connections are required between the solid fin surface and the fluid consisting of air. In this simulation, fluid domain (Enclosure) having size 172x140x130mm and solid domains (Fin with base) are created to the requirements as shown in Figure 2 The setup was modeled with full geometry so that maximum physics of the experimental analysis can be included. Hence it is logical to assume that the behavior of the created system domains is similar to the behavior of the experimental system.

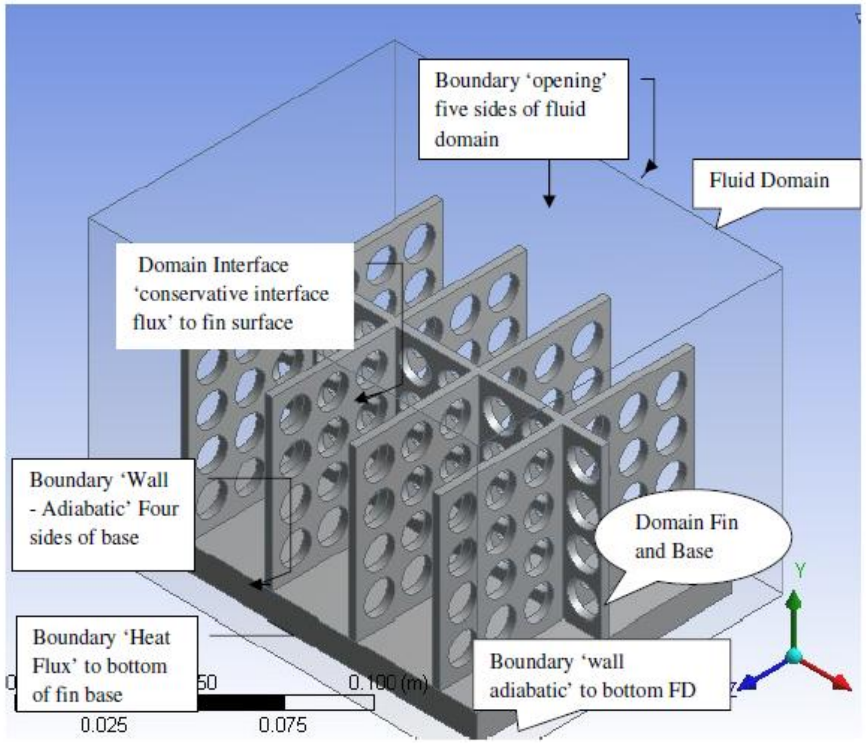


Figure 2 Fluid and Solid Domains and boundary conditions applied,

CFD simulations of type CR12PR

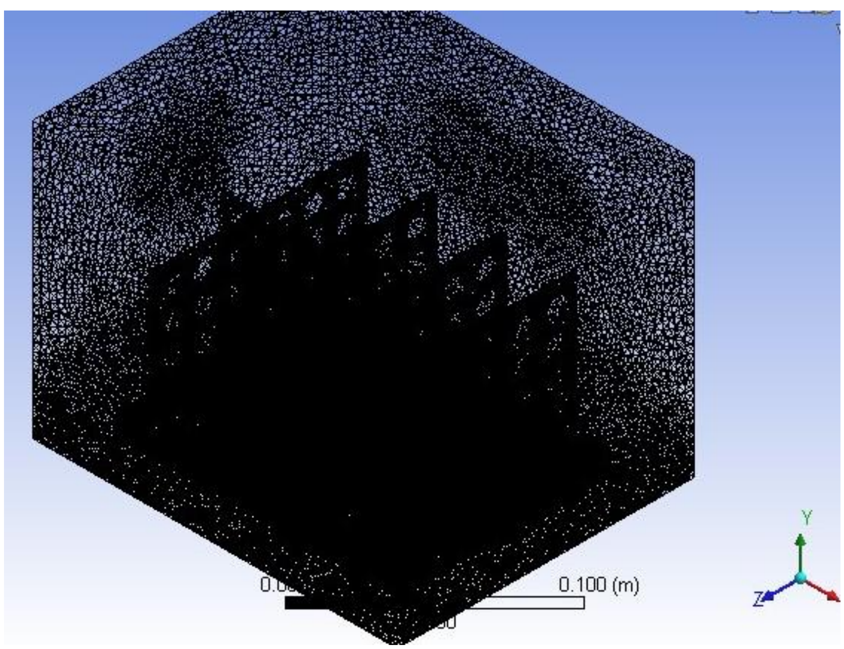


Figure 3 Wireframe model of meshed domains, CFD simulation of type CR12PR

Next step is to generate the mesh; wire frame model of meshed domains is shown in Figure 3. Meshes are generated in standard volume mesher in a CFX-Mesh, which has Advancing Front Volume Mesher. It enables an automatic tetrahedral mesh generation using efficient mesh generation techniques, meshes were created with high contact sizing relevance dense meshing near the fin surface, inflation growth rate 1.2 and total number of tetrahedral elements between 2.8 to 3.4 million.

After generating the mesh successfully meshed model is taken for setup to apply required boundary conditions. Interface between solid fin surface and fluid (air) of fluid domain was created. Through the use of the boundary condition of (refer Figure2) 'Opening' to all outer faces of the enclosure (Fluid domain) except the bottom face which was set to adiabatic. Hence, the size of the fluid domain can be reduced to a great extent and it can be assumed to be as the atmospheric conditions. The bottom surface of base plate was set to the boundary condition to 'Heat Flux'. The heat flux was applied equal to the ratio of Heat transfer by convection to the Area of bottom of a base of the experimental readings. Other vertical sides of base plate set to boundary condition as 'Adiabatic'. After applying the all boundary conditions setup sent to the solver in which, required target values of the root mean square (RMS) of the residual were set to $1.0\text{x}10^{-6}$. In a few

cases, residuals of the energy equations remain constant (about $4.8\text{x}10^{-6}$) Most solutions were getting converged between 240 to 270 iterations. It took 8 to12 hours to get solutions converged.

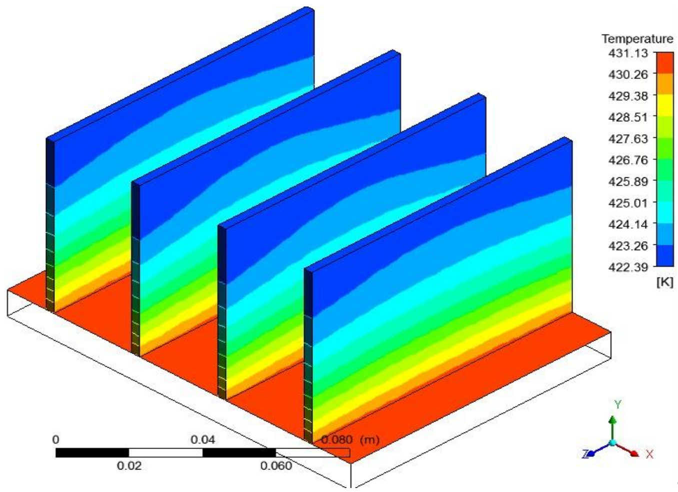
4. Results and Discussion

4.1 Visualization of CFD Simulation results

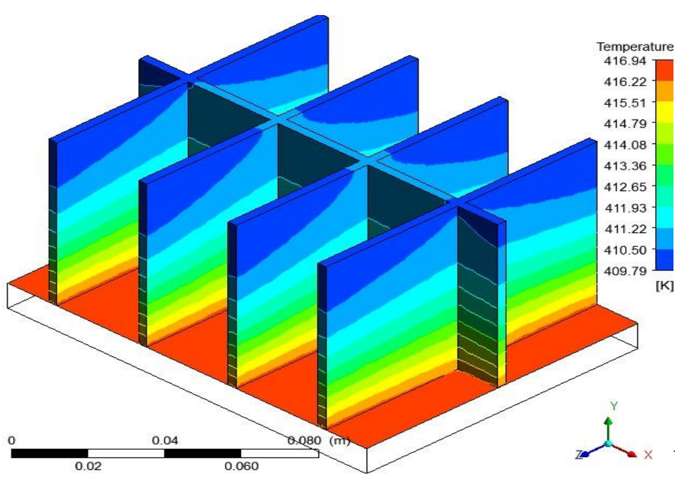
When the solver was terminated, the results were examined which is the post processing part. The CFX result viewer has state of the art viewing 3D results, with the help of the 3D viewer, it is able to view the Temperature contours, velocity vectors, 3D streamlines, particle tracking etc. Average wall heat flux along the fin surface as well as parameters like Nusselt Number, heat transfer coefficient, Rayleigh number and changes in other parameters can also be predicted by computational analysis.

4.1.1Contures of temperature solid domains

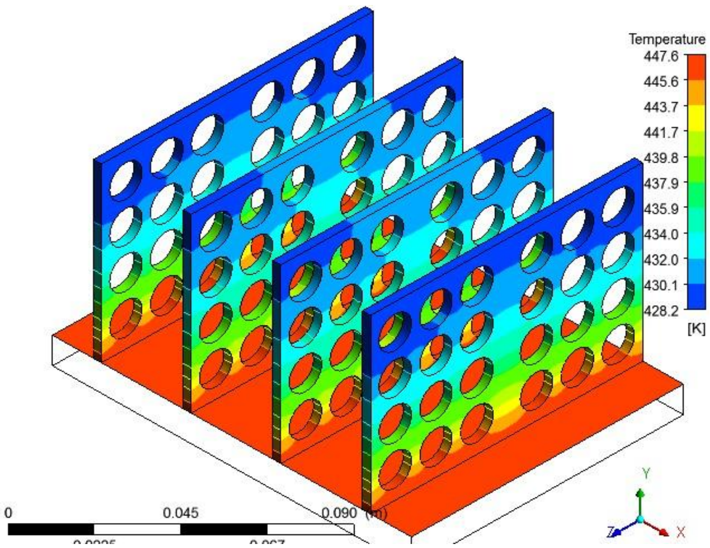
Figure 4 (a-d) shows the temperature contours at 80Watt heat input and which are clipped to their local range of temperature over the convective surface area of a) non perforated fin arrays without cross fin (WCRPL), b) non perforated fin arrays with cross fin(CRPL), c) perforated fin arrays without cross fin (WCR12PR), d) perforated fin arrays with cross fin(CR12PR). It is seen that the temperature difference (base to fin tip) of non perforated without cross fin, and non perforated with cross fin, perforated fin arrays without cross fin and perforated fin arrays with cross fin are 8.71, 7.15,19.4, 16.13 degrees respectively. The 12mm perforated fin arrays without cross fin are showing the highest difference and non perforated fin arrays with cross fin showing the lowest. Hence perforated fin arrays are having better performance than that of the equivalent non perforated fin arrays with or without cross fin.



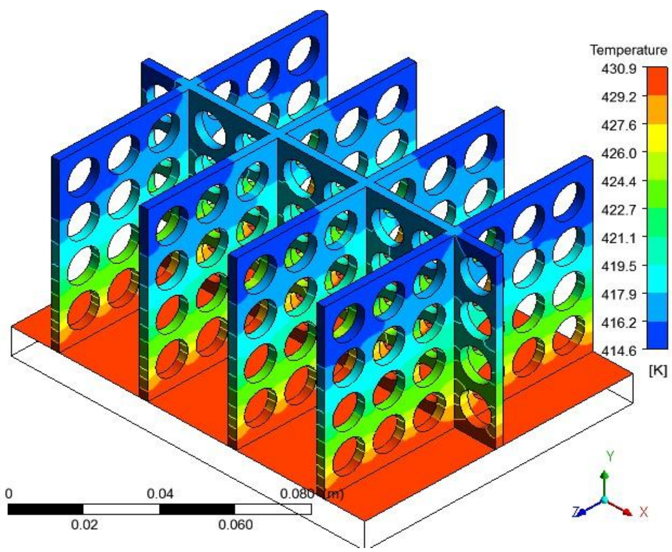
a)



b)



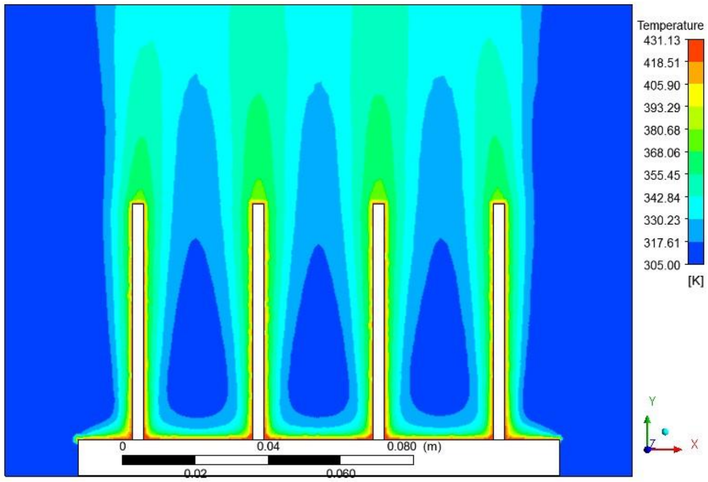
c)



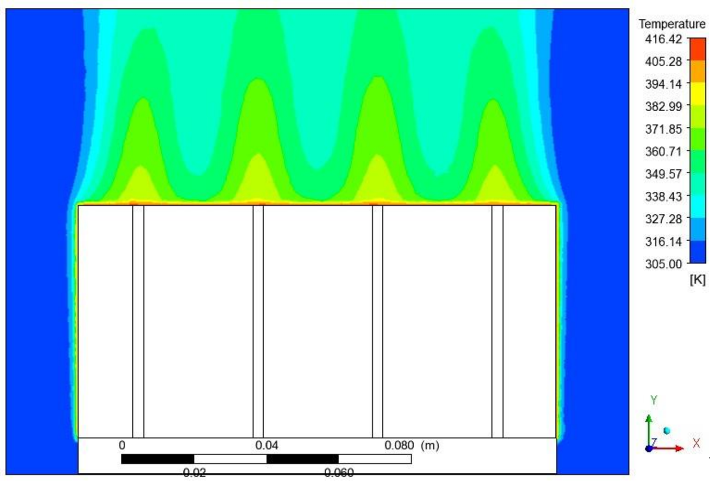
d)

Figure 4 Contours of temperature over a convective surface area at 80W heat input a) non perforated without cross fin(CRPL) b) non perforated with cross fin(WCRPL) c)12mm perforated without cross fin(WCR12PR) d)12mm perforated fin arrays with cross fin(CR12PR)

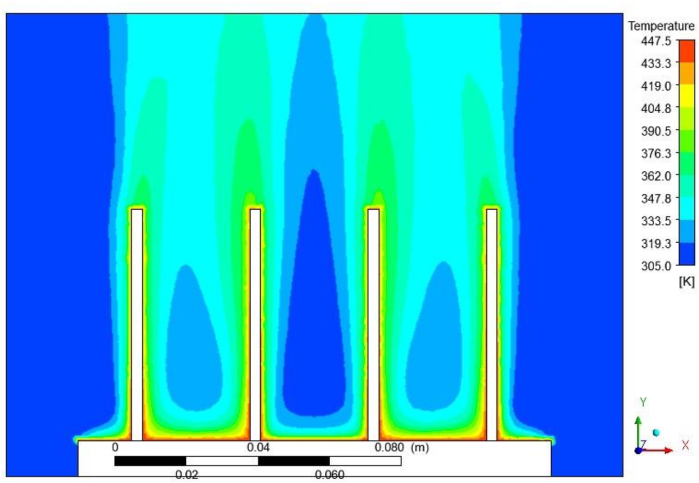
4.1.2Contures of temperature of fluid domains


Temperature
431.13
418.51
405.90
393.29
380.68
368.06
355.45
342.84
330.23
317.61
305.00
[K]
0
0.04
0.080 (m)
0.02
0.060
Y
Z
X

a)


Temperature
416.42
405.28
394.14
382.99
371.85
360.71
349.57
338.43
327.28
316.14
305.00
[K]
0
0.04
0.080 (m)
0.02
0.060
Y
Z
X

b)


Temperature
447.5
433.3
419.0
404.8
390.5
376.3
362.0
347.8
333.5
319.3
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[K]
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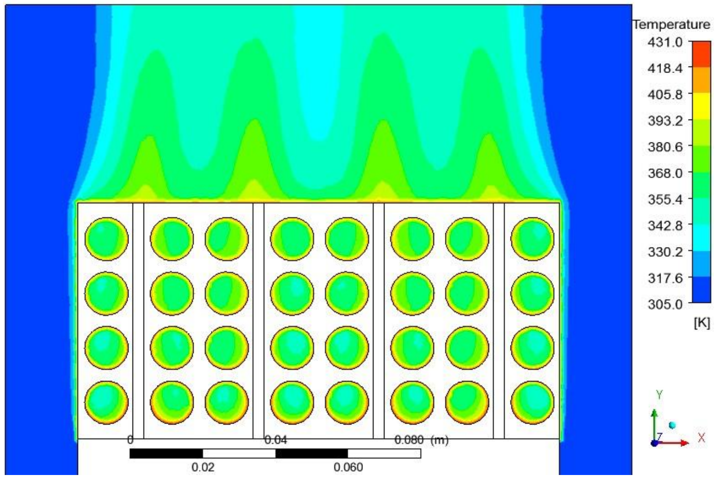
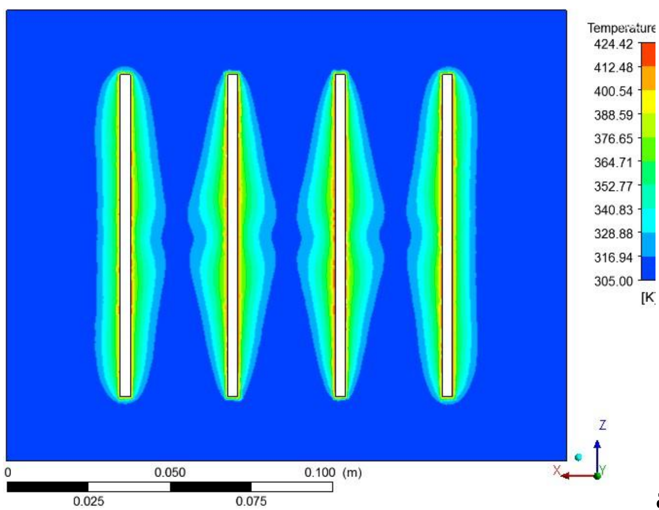
c)



d)

Figure 5.Contours of temperature of fluid domains clipped on XY plane at Z=0.05m, All 80 Watt heat input, a) non perforated without cross fin(WCRPL) b) non perforated with cross fin (CRPL) c)12mm perforated without cross fin(WCR12PR) d) 12mm perforated fin arrays with cross fin (CR12PR)



a)

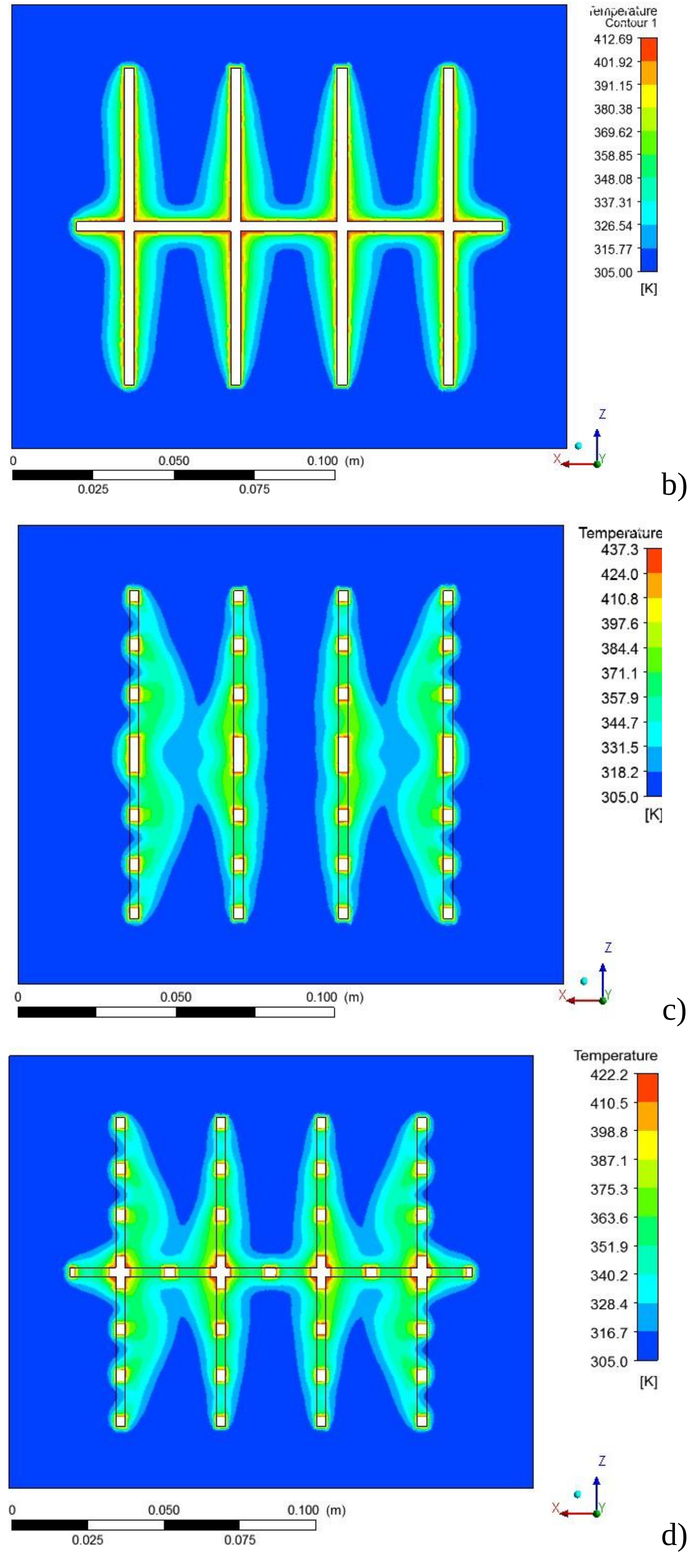


Figure 6.Contours of temperature of fluid domains clipped on ZX plane at Y=36mm, All 80 Watt heat input, a) non perforated without cross fin, b) non perforated with cross fin, c)12mm perforated without cross fin, d) 12mm perforated fin arrays with cross fin

Figures 5 (a-d) Shows temperature contours of the fluid domain of the different types of fin arrays, clipped on the XY plane, passing through middle of the domains. And Figures 6 (a-d) shows temperature contours of fluid domain clipped on the ZX Plane at Y=36mm. From figures 5and 6, it is seen that Outer part of the fluid domain is at ambient temperature and layer adjacent to fin at high temperature. Hence, heat is dissipated to the environment. The stagnant layer of hot air around the non perforated fin array can be eased by adding the perforation, which provides cross ventilation, resulting in an increase in the rate of heat dissipation to the surrounding when perforation is added. Also Figure 7 vectors of velocity around fin on XY plane it is seen that the inward flow of air in the fluid domain from the bottom, moving up along sides of cross fin and leaving the domain from the top and from Figure 8 enlarged isometric view of combined temperature contours and vectors of velocity around the perforated fin it seen that air is getting circulated through perforation. Hence it confirms that the area removed by the perforation from the fin is compensated by the entry of more fresh cold air through the perforation and also from the ends of the fin resulting in enhancement of the heat transfer coefficient.

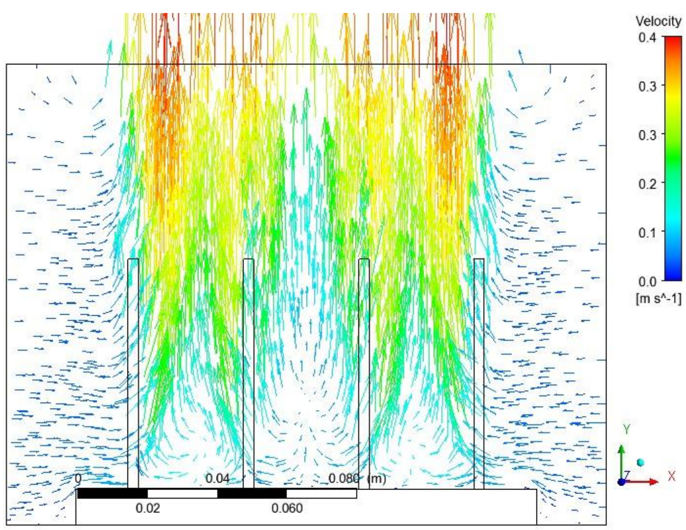


Figure 7 Vectors of velocity 12mm perforated fin array without cross fin at center

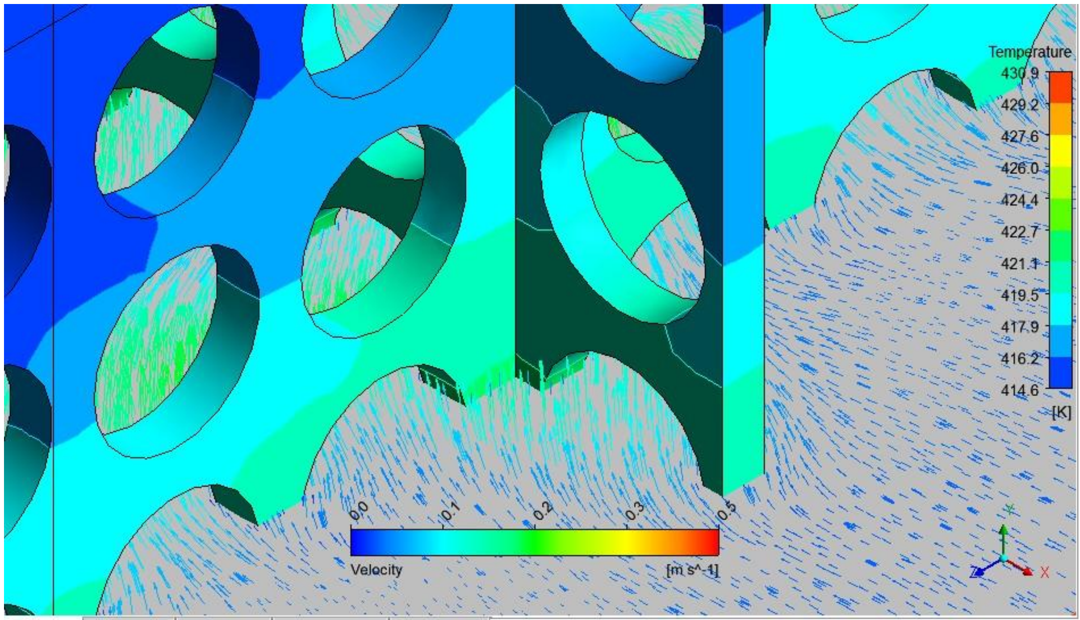


Figure 8 Enlarged isometric view of combined the vectors of velocity and temperature contours of solid domain of the12 mm perforated fin array CR12PR

4.2.1 Effect of Cross Fin

Figure 9, shows the plot of heat transfer coefficient (ha) Vs Heat input (Q) for perforated fin arrays having cross fin and without cross fin of 12mm size of perforation, from actual experimental analysis. It shows that average heat transfer coefficient, of the perforated fin arrays having cross fin at center is less than that of corresponding perforated fin array of without cross fin at the center at the same heat inputs. Similar trend occurs for the plots of comparison among having cross fin and without cross fin for other type of sizes of perforation. Hence, Cross fin is not advisable for shorter fin spacing.

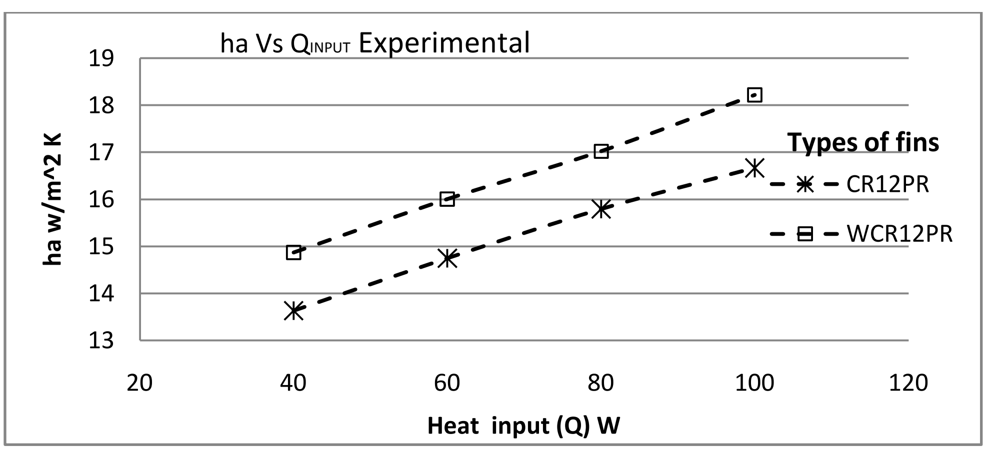


Figure 9 Average heat transfer coefficient ha Vs Heat input of perforated fin arrays with and without cross fin

4.3 Effect of perforation

Plot of Nusselt no (Nu) Vs Raleigh no (Ra) for the different size of perforation of fin arrays without cross fin is given in Figure 10. From figure it seen that 12 mm perforated fin arrays are having the highest Nu among all other type, and Nu of 4mm size of perforation is even lower than solid fin arrays. Similar trend is observed for the plot of average heat transfer coefficient ha Vs heat input Q of fin arrays having cross fin Figure 11 hence conclusion can be drawn that enhancement of heat transfer occurs with increase in size of perforation except fin arrays of 4mm size of perforation. Next section explains more about this.

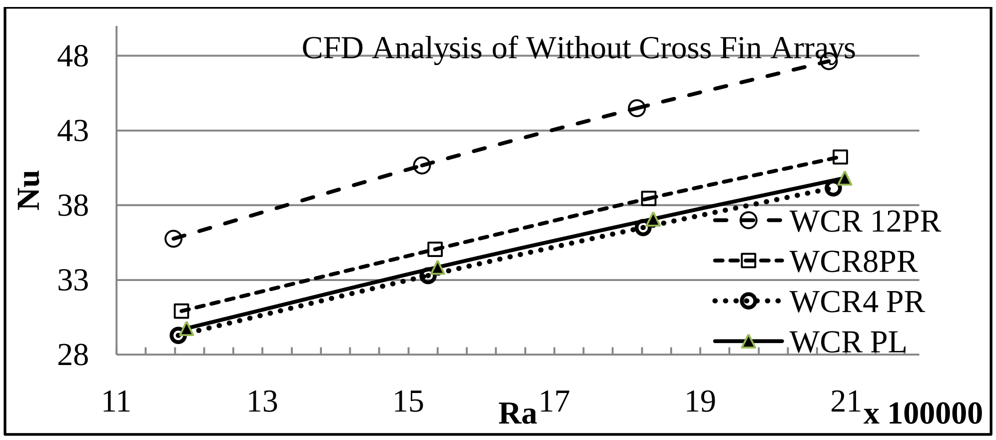


Figure 10 Performance of Nusselt number with Rayleigh number for perforated fin arrays (without cross fin, 80W heat input)

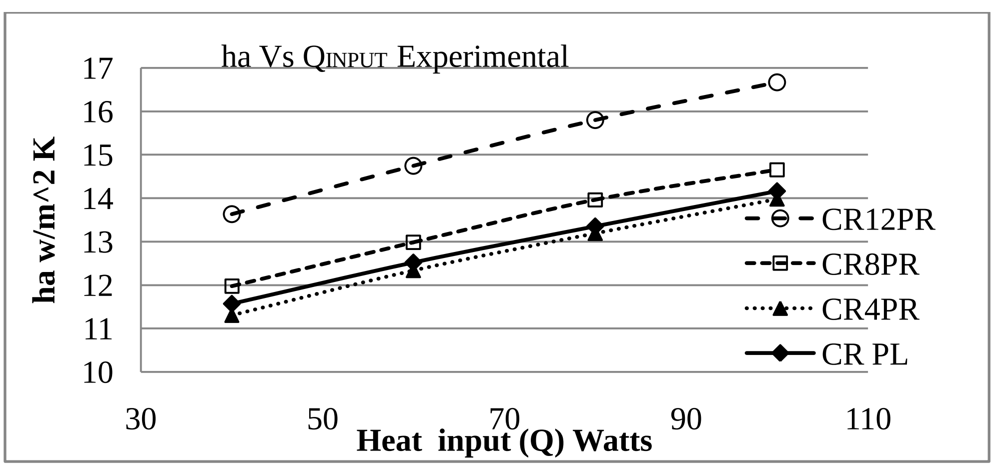


Figure 11 Performance of Average heat transfer coefficient with heat input for perforated fin arrays (with cross fin, 80W heat input)

4.4 Minimum Size of Perforation

Figure 12 shows the performance of average heat flux (q) with increasing size of perforation for fin arrays with and without cross fin. Initially graph is showing decreasing trend up to 3mm size of perforation and then shows increasing trend. Also from Figures 10 and 11 it reveals that least performance of 4 mm perforated fin arrays, even poor than the non perforated fin arrays for both arrays with cross fin as well as arrays of without cross fin. Hence, from CFD analysis as well as experimental analysis it is found that when the Heat Transfer Area of the perforated fin array (Ap) must be less than the Heat Transfer Area of Non perforated Fin array (As), to get the considerable effect of perforation in the heat transfer coefficient as well as in the conserved heat flux.

$$Ap \leq As$$

Hence, after calculation of Ap and As it can be proved that radius of perforation must greater than thickness of fin ($r \geq t$) then only improvement in the performance of the perforated fin array over its non perforated fin array will occurs.

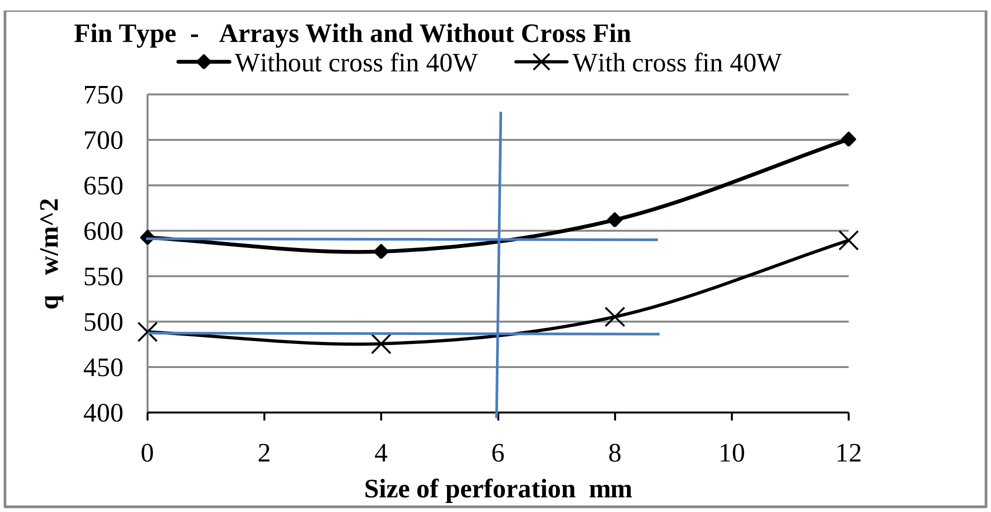


Figure12 Average heat flux vs. size of perforation with and without cross fin

Also this is confirmed by the Figure 13 which is magnified view contours of temperature from top combined with vectors on the XZ Plane, passing through the bottom row of 4 mm perforated holes having cross fin. It is seen that a hot layer is formed inside the hole which is restricting to flow of the cold air through holes. Figure 14 shows similar simulation of 12 mm size of perforated fin arrays with cross fin shows the clear circulation of the air through the perforated holes.

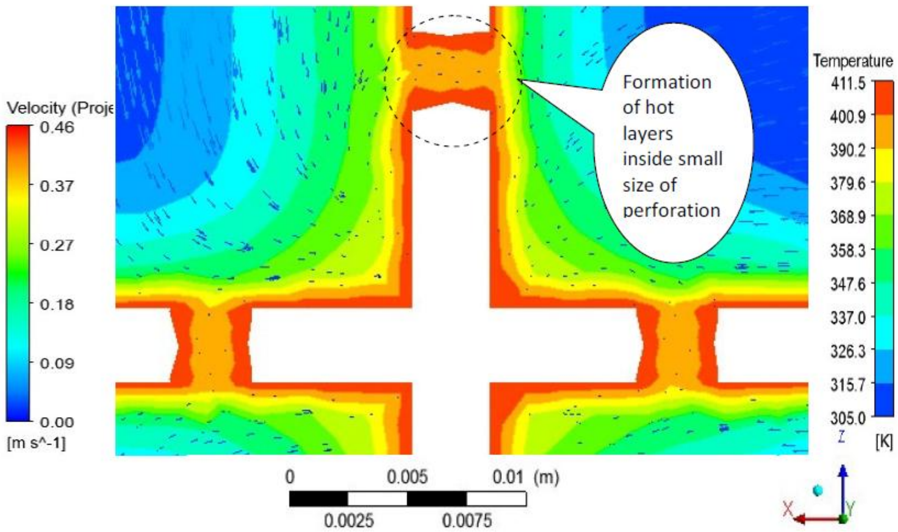


Figure 13 Magnified view of the temperature contours combined with vectors on the XZ Plane from simulations of CR4PR

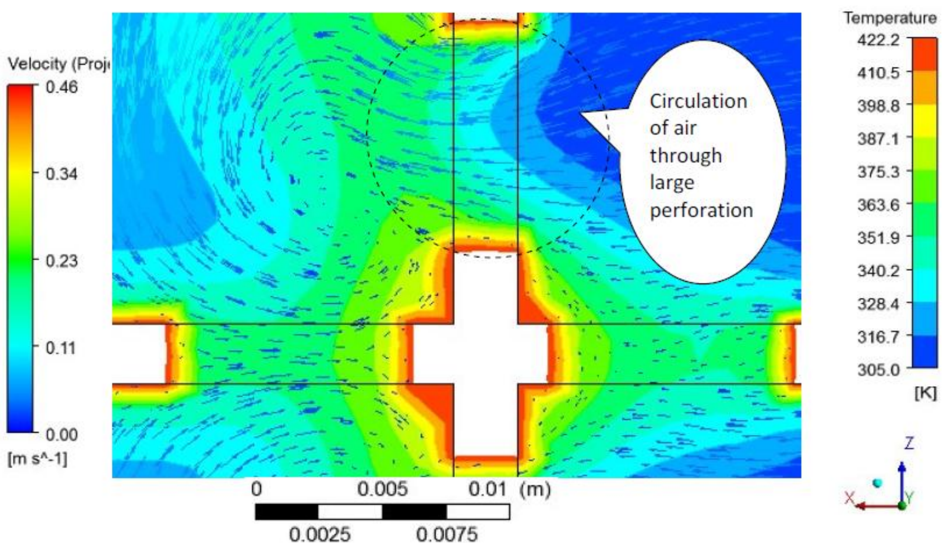


Figure14 Magnified view, the temperature contours combined with vectors on the XZ Plane, simulations of CR12PR

5. Grid independency and validation of results

A Grid independency test was carried out by varying the number of elements from 2.4 million to 3.4 million. It was observed that, a very minute variation (< 1^oC) in various output temperatures and < $3w/m^2$ in conserved heat flux values existed. Even though, as time was not a constraint and for accurate results the total number of tetrahedral elements in a grid were chosen to be approximately 2.8 to 3.4 million.

It is observed that no variation more than 2% in any of the quantity generated outputs from the simulation. Hence, with further working out it concludes that a CFX mesh would give acceptable precision in predicting the essential characteristics of heat transfer through different arrays of fins, particularly for comparative purpose of this study.

Comparison among the CFD simulation results and experimental values of various temperatures, heat transfer coefficient and Nusselt no (Nu) and Raleigh no (Ra) with a maximum variation of -8% to +6% was observed. This may be due to uncertainties during the experimentation and idealistic conditions in simulation and uncertainties of experimental results are below 5%.

6. Conclusions

- It was verified as a good agreement among the experimental and the CFD analysis (ANSYS Fluid Flow CFX -Workbench) for laminar regimes in all the experiments.
- CFD Simulation and followed experimentation have revealed that perforated fins with the big size of perforation can be applied efficiently to augment the rate of heat transfer. As perforation to the fins allows more amount of outside cold air get sucked inside from the base and comes in contact with the heated fin surface in the turns which gets heated and goes out from top resulting faster cooling the fin surface.
- Improvement in the performance of the perforated fin array than its corresponding solid fin array will start to occur when the radius of perforation (r) is more than the thickness of the fin (r > t).
- CFD simulation and succeeding experimental analysis have revealed that average heat flux and average heat transfer coefficient of the non perforated fin arrays with cross fin at center is less than that of non perforated without cross fin at the center for tested sizes of perforation

and fin parameters. It may be due to the restricted passage for movement of air. Hence, adding a cross fin at the center to a fin array with or without perforation for shorter spacing is not advisable.

- The results of the perforated fin arrays with and without cross fin, non perforated fin arrays with and without cross fin at center shows that the average heat transfer coefficient rises with increasing with heat input which has proved by both experimental as well as CFD analysis. Finally, one of the most important benefits of the utilization of perforated fins without cross fin is reduction of fin's weight. Low weight certifies saving material of fins and related equipments such as heat sinks. From the experimental trials new design of the fine structure with perforation can be maximize the rate of heat transfer.

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