Numerical and Experimental Investigations On Various Fin Configurations subjected to Isoflux Heating under the Influence of Convective Cooling

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ABSTRACT
Present paper aims at numerical and experimental assessment of thermal performance of various fin arrays. The numerical results are obtained by FLUENT code which uses standard discretization practices of spatial, temporal and convective derivatives in mass, momentum and energy transport equations. Results are reported in terms of the variation of heat transfer coefficient and Nusselt number with the respect to Reynolds number.

Keywords - Effectiveness, Fins, Nusselt number, Reynolds number,

I. INTRODUCTION
Pinfins are extended surfaces used to enhance heat transfer by availing additional area for heat transport and introducing flow patterns contributing to enhancement of heat transport. Ellison [1] and Kraus and Bar-Cohen [2] have presented the fundamentals of heat transfer and hydrodynamics characteristics of heat sinks including the fin efficiency, forced convective correlations, applications in heat sinks, etc. Electronic cooling is one of the major application of pinfins. Researchers like Iyengar and Bar-Cohen [3] determined the least-energy optimization of plate fin heat sinks in the status of forced convection. Investigations are also carried out regarding shape optimization of array of pin-fins as done by Park et al. [4]

Fin efficiency and convection effectiveness can be examined to minimize any significant conduction resistance though the fins and improve the overall performance of the heat sink. There exists significant work carried out in the thermal analysis of heat sink design. An analytical approach was taken by Keyes [5] who developed formulas for the fin and channel dimensions that provide optimum cooling under various forced convection cooling conditions.

In the present study, the objective is to numerically investigate the performance of fin arrays consisting of circular, square shaped fins. The array of fin is having inline configuration without any staggering. The analysis is carried out using numerical simulations and experimentations.

II. MATHEMATICAL FORMULATION
A base plate of 88 X 88 X 8 mm is subjected to isoflux heating and the fins are attached on the top surface in in-line manner. Fins used are of two types viz. circular and square shaped having 35 mm height. The set of 25 fins in order pattern 5 X 5 are used to enhance the heat transfer. The assembly of plates and fins is enclosed in a rectangular duct sixe of 400 X 88 X 180 mm. It can be observed that dimensions of the duct are maintained sufficiently large such that flow is undisturbed by end effects. In order to make a comparison , some parameters are fixed for all geometries. These are: (i) Same pitch pitch (Px=Py) (ii) the area for air flow passage per base area. The commonly used in-line circular array of fins with 8 mm pin diameter (w) and a 16 mm pitch has been selected to be the base case for comparison.

To analyze the flow and temperature distributions in geometry, set of partial differential equations namely, solve momentum (Navier-Stoke's equations), continuity,energy transport equation are to be solved using numerical algorithms subjected to appropriate boundary conditions alongwith turbulent kinetic energy and turbulent dissipation rate equations.

Continuity Equation:

$$\partial u/\partial x + \partial v/\partial y + \partial w/\partial z = 0 \quad (1)$$

X-direction momentum equation:

$$\rho Du/Dt = - \partial p/\partial x + \partial/\partial x (\mu + \mu_t (2\partial u/\partial x)) + \partial/\partial y(\mu + \mu_t (\partial u/\partial y + \partial v/\partial x) + \partial/\partial z(\mu + \mu_t (\partial u/\partial z + \partial w/\partial x) + \rho g_x \quad (2)$$

Y-direction momentum equation:

$$\rho Dv/Dt = - \partial p/\partial y + \partial/\partial x (\mu + \mu_t (\partial u/\partial y + \partial v/\partial x)) + \partial/\partial y(\mu + \mu_t (2\partial v/\partial y) + \partial/\partial z(\mu + \mu_t (\partial v/\partial z + \partial w/\partial y) + \rho g_y \quad (3)$$

Z-direction momentum equation:

$$\rho Dw/Dt = - \partial p/\partial y + \partial/\partial x (\mu + \mu_t (\partial u/\partial z + \partial w/\partial x)) + \partial/\partial y(\mu + \mu_t (\partial v/\partial z + \partial w/\partial y) + \partial/\partial z(\mu + \mu_t (2\partial w/\partial z) + \rho g_z \quad (4)$$

Energy equation (As applied to solid and fluid):

$$\rho Cp \; DT/Dt = \partial/\partial x (k + k_t (\partial T /\partial x) + \partial/\partial y (k + k_t (\partial T /\partial y) + \partial/\partial z (k + k_t (\partial T /\partial z) \quad (5)$$

The terms of turbulent diffusivities like μ_t and k_t are handled by two equation based turbulence model- Realizable k-ε model. The realizable k-ε model has distinct feature and that is the turbulent dissipation rate equation is derived in such way that conservation of mean square of vorticity is guaranteed [6] . Readers are also directed to ref [7] for further clarification.

Results of turbulent flows strongly depend on near wall treatment. Near wall quantities are modeling with the effective usage of blending functions. This approach is often referred to as – “Enhanced Wall Treatment” in various literature o commercial CFD like reference- [7].

The numerical algorithm to solve the set of partial differential equations is as described by Patankar [8]. The convection terms are discretized using Second Order Upwind scheme and SIMPLE algorithm is used to handle pressure-velocity coupling in Momentum equations.

Boundary conditions are deployed in such way that at the inlet of channel known velocity is specified and the channel is assumed to be open to atmospheric pressure; stream-wise gradient of temperature is assumed to be vanishing at outlet. A constant heat flux boundary condition is specified to represent the assembly of finned plate which is subject subjected to appropriate conjugate heat transfer conditions at the solid-air interfaces. Properties like density, thermal conductivity in Eq (5) are assumed to be varying subjected to solid or fluid continuum zone.

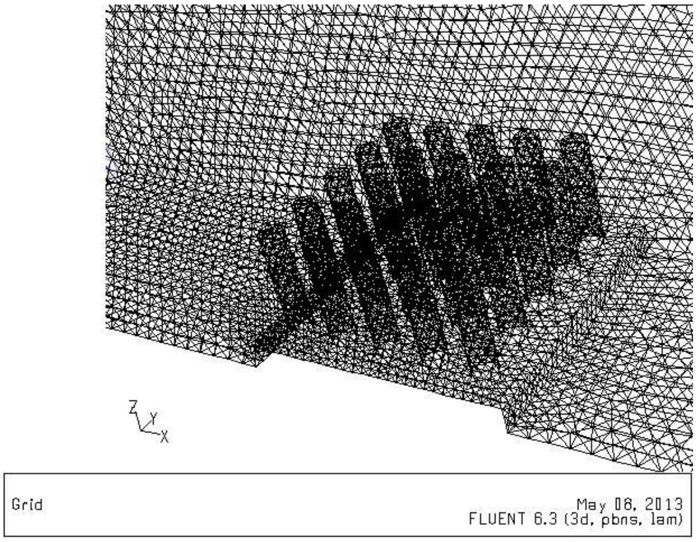

Fig.1 A sample mesh for finned array.

Fig.1 shows a typical mesh for finned array and some part of surrounding duct.

III. EXPERIMENTATION

Forced convection cooling is produced by using a blower, the fin arrays are placed in a duct 90mm X 90mm X 1000mm long. The blower is controlled by using a dimmerstat to adjust the flow rate for each experiment. The air velocity is recorded by using anemometer 100mm upstream of the heatsink. Flat heater pad of nichrome wire were attached to the bottom of heatsink with constant flux heating on which arrays of Aluminium pinfin are placed. Polycarbonate sheets are placed below the heater pad to minimize the heat loss from the bottom. Net Input flux can be calculated by subtracting the heat loss occurring from the bottom surface. The conduction heat loss at the bottom is calculated by sensing the temperatures at the top and bottom of polycarbonate sheet.

IV. RESULTS

Results are reported in terms variation of Nusselt number (Nu) with respect to Reynolds number and Grashof number for all three configurations and heat transfer coefficient and velocity . The length scale chosen for definition of Reynolds and Nusself number is fin length (35 mm) and velocity scale is inlet velocity.

$$Re = \rho VL/\mu; \quad Nu = hL/k \tag{6}$$

To define heat transfer coefficient, average surface temperature of solid-fluid interface (T_{ave}) and the applied flux (q) is taken into account as described:

$$h(Tave - Tin) = q \tag{7}$$

To define Grashof number, characteristic temperature difference is based on heat flux as given below :

$$Gr = g\beta\Delta TL3/v2 \ ; \Delta T = qL/k \tag{8}$$

Since the problem assumes isoflux heating to the fin base, regular definition of effectiveness of fin is slightly modified. Effectiveness is defined as the ratio of heat transfer coefficient with fins to the heat transfer coefficient without fins. The above definition of effectiveness can be named as “Convective Effectiveness”.

3.1 Variation of Heat Transfer coefficient

The variation of heat transfer coefficient with respect to air velocity is depicted in Fig. 2.

In all the configurations studied, it is found that Heat Transfer Coefficient increases as Reynolds number/Air velocity increase. This is due to the fact that Reynolds number signifies the strength of forced convection and, thus, increase of Heat Transfer Coefficient with respect to rise in Reynolds number is justified.

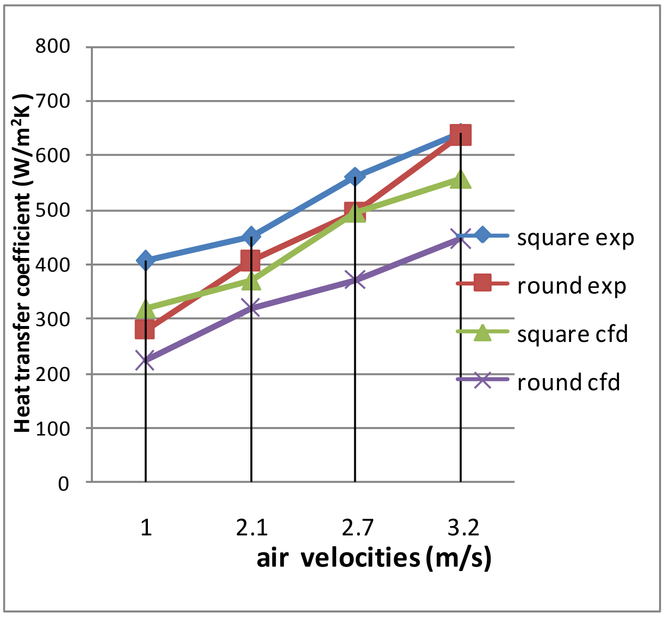

Fig. 2. Variation of Heat Transfer Coefficient

It can be seen from Fig 2 that for plane duct the heat transfer coefficient is relatively smaller than that would occur in other configurations. In plane ducts, heat transfer occurs only via the thermal boundary layer, while if flow is partially obstructed (due to protrusions/fins), the mechanisms like boundary layer separation, reattachement, shedding of vortices become predominant and these mechanisms contribute to the increase in the rate of heat transfer.

3.2 Variation of Nusselt number

Since Nusselt number is directly proportional to heat transfer coefficient, the qualitative variation of Nusselt number against Reynolds number remain same as that of the heat transfer coefficient. As can be seen from the Figs . 3 and 4 Reynolds number increase, the strength of forced convection increase and which in turn results into the rise in Nusselt number.

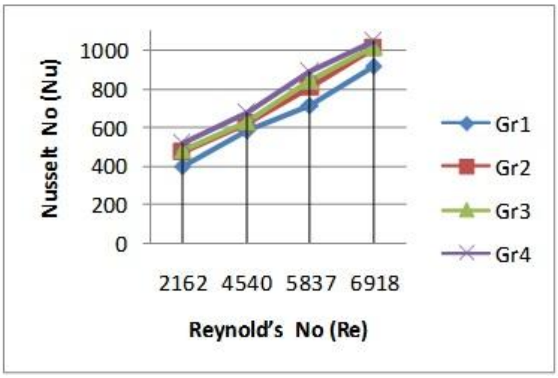

Fig. 3 Variation of experimental Nusselt number with Reynolds and Grashof number (Gr1= 3.3 X 10^7 Gr2= 4.1 X 10^7 , Gr3= 5 X 10^7, Gr4 = 5.72 X 10^7) for round configuration.

It is also observed that increase in Grashof number cause increase in Nusselt number, this is attributed to the fact the buoyancy assists heat transfer. However the dependence of Grashof number is not as strong as that of the Reynolds number. This is attributed to the fact that forced convection washes out the heat directly while naturally convection merely assists forced flow.

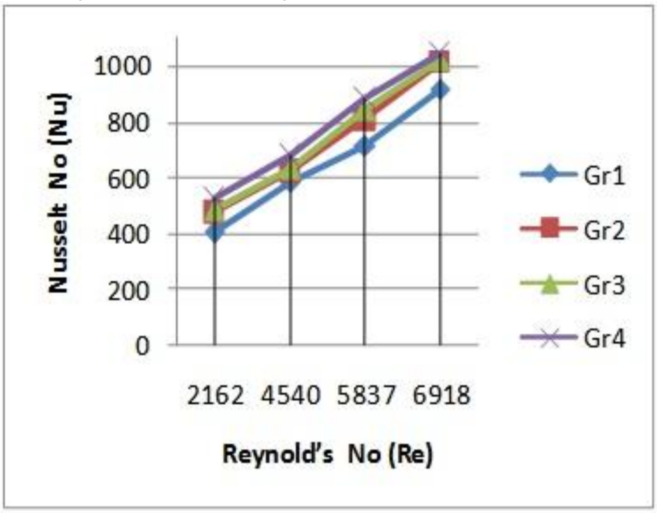

Fig. 4 Variation of CFD Based Nusselt number with Reynolds and Grashof number (Gr1= 3.3 X 10^7 Gr2= 4.1 X 10^7 Gr3= 5 X 10^7, Gr4 = 5.72 X 10^7) for round configuration

Qualitatively, variation of numerically obtained values of Nusselt number with respect to Reynolds and Grashof number is same as observed with experimental results. This is illustrated in Fig. 4.

TABLE. 1 Comparison of Nusselt number for various configurations.

		Gr1= 3.3 X 10^7	Gr2= 4.1 X 10^7	Gr3= 5 X 10^7	Gr4= 5.72 X 10^7
Square Configuration					
Re= 2162	CFD	460	426	406	383
	Expt	585.87	599.1	532.9	519.1
Re =4540	CFD	537	506	483	475
	Expt	648.1	621.85	596.1	570.96
Round Configuration					
Re= 2162	CFD	322	352	406	415
	Expt	402	476	483.1	523.1
Re =4540	CFD	460	506	534	548
	Expt	584.9	622	634	676.96

Out of many results obtained, Table 1 summarizes various results at a glance. Following conclusions can be reached :

1) Nusselt number predicted by CFD methods and that observed by experimentation differ easily by 20 to 30 percent. Without usage of turbulence model, simulation resulted into Nusselt number values 50 per cent lesser than experimental values and at a times convergence difficulties were encountered during simulations.

Choice of turbulence is always a matter of debate, however, the presently chosen model has a capability of blending the wall functions.

2) It is observed that Nusselt number values is are on higher side than that of the circular fins.

3) The trends of variation of Nusselt number with Grashof number for square configuration is opposite than that of the trend observed in round configuration. In round configuration, buoyancy plays assisting role with enhancement of heat transfer coefficient. However, the trends are opposite for square fins.

V. CONCLUSION

The present paper is aimed at qualitative and quantitative investigation of convective cooling of fin arrays using numerical and experimental appraoch. Comparison is made between square shaped and round shaped fins. The results reveal that heat transfer is higher in square configuration that that obtained in round configuration. Surface temperature based heat transfer coefficient and Nusselt number monotonously increase with the increase in Reynolds number, variation of these quantities with Grashof number exhibit complicated nature. The turbulence is well resolved by Realizable k-ε model with blending function approach for near wall treatment. However, still difference of 20 to 30 percentage is seen to be prevalent for experimental and numerical results.

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