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Enhancement of Heat Transfer by Natural Convection from Discrete Fins

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ABSTRACT

Heat transfer enhancement in fins can be increased by using discrete fins. Numerical studies were conducted to analyzed natural convection heat transfer from solid and discrete fins. Numerical investigation was made for three-dimensional fluid flow and convective heat transfer from an array of solid and discrete fins that were mounted on a flat plate. Commercial computational fluid dynamics (CFD) code, Fluent was used to perform simulations. Incompressible air as working fluid was modeled using Navier–Stokes equations and standard based k-ε turbulent model was used to predict turbulent flow parameters. Temperature field inside the fins was obtained by solving Fourier's conduction equation. The conjugate differential equations for both solid and gas phase were solved simultaneously by finite volume procedure using SIMPLE algorithm. The problem was analyzed varying parameters like, number of cut along the length of fin and width of cut was 2 mm constant while heat input was 15 W, 30 W and 45 W. As increasing the heat input, Nusselt number increases. Numerical computations were validated with experimental studies of the previous investigators and good agreements were observed Results show that fins with longitudinal cut, have remarkable heat transfer enhancement in addition to the considerable reduction in weight by comparison with solid fins.

Keywords:**Discrete fins, Natural Convection, enhancement, Navier–Stokes equations.**

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I. INTRODUCTION

Extended surfaces (fins) were frequently used in heat exchanging devices for the purpose of increasing the heat transfer between a primary surface and the surrounding fluid. Various types of heat exchanger fins, ranging from relatively simple shapes, such as rectangular, square, cylindrical, annular, tapered or pin fins, to a combination of different geometries, have been used. The study of improving heat transfer performance was referred to as heat transfer augmentation or enhancement. The heat transfer augmentation was very important subject in industrial heat exchangers and other thermal application. There were many techniques which were available for augmentation for single or two-phase heat transfer in natural or forced convection.Akyol and Bilen [1] conducted an experiment on surface with hollow rectangular finned array with in line and staggered arrangement. They found that the heat transfer significantly improves compared to surface without fin whereas staggered fin arrangement gives slightly better heat transfer than in line arrangement. Baskaya et al. [2] carried out parametric study of natural convection heat transfer from the horizontal rectangular fin arrays. They investigated the effects of a wide range of geometrical parameters like fin spacing, fin height, fin length and temperature difference between fin and surroundings; to the heat transfer from horizontal fin arrays. Bassam and Abu [3] numerically analysed heat transfer of permeable fins and resulted in significant enhancement over solid fins.They stated that under no condition did the increase of number of permeable fins result in decrease in Nusselt number as opposite to solid fins. They used certain assumptions to make the analysis simple that the fins were made up of highly conducting material.

AlEssa et al [4] have examined the enhancement of natural convection heat transfer from a horizontal rectangular fin embedded with rectangular perforations of aspect ratio of two [5] and a fin by triangular perforations of bases parallel and toward its base by using finite element technique. Pise and Awasarmol [6] used permeable fins for augmentation of natural convection heat transfer from cylinder. The experiment was conducted to compare the rate of heat transfer from solid and permeable fin. Shaeri et al [7] they studies convective heat transfer from an array of solid and perforated fins that were mounted on a flat plate. They state that fins with longitudinal pores, have remarkable heat transfer enhancement in addition to the considerable reduction in weight by comparison with solid fins. Sandikar et al. [8] carried out numerical investigations of the heat transfer by natural convection from horizontal base with vertical fin array for solid and perforated fins. Base inclination and base temperature was varied as 0^0 to $90^0$and 40 to 100^0C respectively. Results show that fins with longitudinal perforations, have remarkable heat transfer enhancement and considerable reduction in weight in comparison with solid fins.

The aim of the present study was to determine thermal performance of a new type of discrete fins and comparing their results with solid fin and the flat surface without fins for the same conditions. For this consideration, three-dimensional turbulent fluid flow and convective heat transfer around an array of solid and discrete rectangular fins wereanalyzed numerically. Fins have slots with square cross section with 2 mm and No. of slots, like 1 to 5. These cutting slots were along the fin width and their cross section was perpendicular to the fluid flow direction. Selecting smaller size makes the flow laminar in the channel and increasing more channels was not possible due to fin dimension and shape. As far as the authors were aware, detailed experimental or theoretical work on flow and heat transfer for an array of three-dimensional discrete fins with the shape presented in this study mounted on a flat plate with the selected perforation seems to be limited and the present analysis maybe the first study with turbulent flow regime.

II. PROBLEM DESCRIPTION

Fin array used in this investigation was shown in Fig.1. This type of arrays will be attached to a base plate. Typical practical arrangements of similar array of solid fins were presented and described in [9]. The airflow was considered to be steady and turbulent with constant properties. The fin material was aluminium with thermal conductivity of 202 (W/mK) that was used widely for heat sinks. In this model, square cross sectional slot of dimensions 25mm×2mm×2mm were made in lateral surfaces of rectangular plate fins, along the thickness of fins. The fin length was used as the characteristic length for calculation purpose. Studies were made for 15-45 W heat.

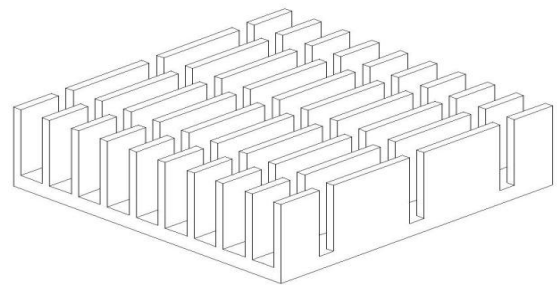

Fig.-1 Physical model of discrete fin block

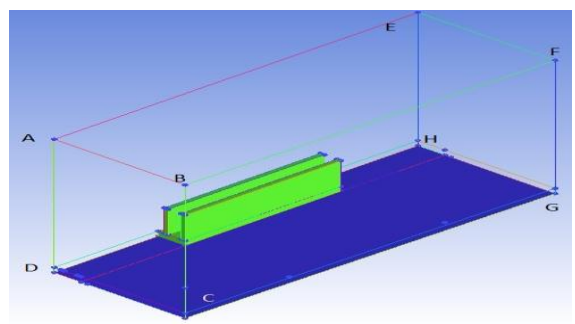

Fig.-2 Computational domain for solid fins

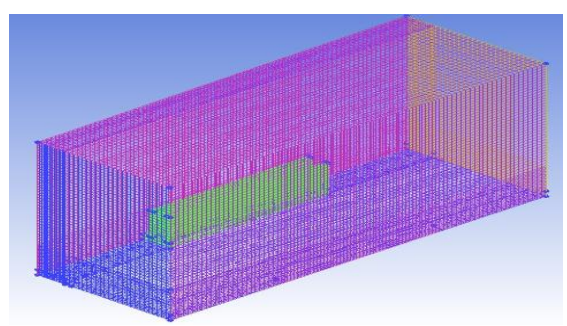

Fig.-3 Computational domain with mesh for solid fins

A. Assumptions for analysis

The analysis and results reported were based on the following assumptions:

(i) Steady and one-dimensional heat conduction.

(ii) Homogeneous and isotropic fin material with constant thermal conductivity.

(iii) No heat sources/sinks in the fin body.

(iv) Uniform base and ambient temperatures.

(v) Side area of the fin was much smaller than its surface area ($w \gg t$).

(vi) Uniform heat transfer coefficient over the fin surfaces (perforated or solid).

(vii) Negligible radiation effects.

B. Initial and boundary conditions

The plane CDHG was assumed adiabatic plane and to make advantage of symmetry boundary condition plane ADHE was symmetry plane. All other remaining planes i.e. pressure outlet boundary condition $P=P_\infty$, $T=T_\infty$, and $\rho=\rho_\infty$was imposed. Although this type of boundary condition was imposed for fully developed cases but since the outlet boundary was sufficiently far from the plate, its usage will not affect the results considerably. Also in these planes velocity component w = 0. No slip boundary condition was applied to all remaining planes that were walls. Free stream temperature was assumed 27^0C and the fin base, has a constant heat input varies from 15-45 W. In this study heat transfer due to radiation was neglected. Commercial software ICEM CFD was used for grid generation. The cell quantity amounted to 2.2 million cells. The range of parameters was listed in Table-1.

Table-1 Range of parameters

Fin length (L), mm	75
Fin thickness (t), mm	2
Fin array length (B),mm	74
Fin spacing (S), mm	6
Number of fins (N)	10
Fin width (W),mm	25
Ambient temperature T_∞	27 0C
Heat Input, W	15, 30 , 45

C. Flow and energy equations

Commercial fluid dynamics (CFD) code, Fluent 13 was used to perform simulations. The governing equations used for simulations were listed below. By applying time- averaging procedures to conservation equations, the basic equations that govern the mean flow quantities for the three-dimensional steady state incompressible turbulent flow can be described as follows:

$$\frac{\partial u_i}{\partial x_i} = 0 \qquad (1)$$

Momentum equation:

$$\frac{\partial}{\partial x_i}\left(\rho u_j u_i - \tau_{ij}\right) = -\frac{\partial p}{\partial x_i} \qquad (2)$$

Energy equation:

$$\rho C_p \frac{\partial (u_i T)}{\partial x_i} = \frac{\partial}{\partial x_i}\left[\frac{\partial T}{\partial x_i}(\lambda + \lambda_t)\right] \qquad (7)$$

For heat conduction in the fin, Fourier's equation was applied as:

$$\frac{\partial^2 T}{\partial x^2} = 0 \qquad (8)$$

To determine temperature field in the fin's surfaces and perforation's walls, the Fourier's steady state heat conduction equation was solved in solid region simultaneously with convection in the fluid.

D. Solution method

The governing Eqs.(1), (2), (7), (9), and (10) were solved using finite volume method and using SIMPLE algorithm[13] The convective terms were discredited using Second Order discretization technique. Velocity components were first calculated from the Navier – Stokes equations using a guessed pressure field, and then the pressure and velocities were corrected, to satisfy the continuity equation. The procedure continues until the sum of the residuals of continuity equations in each cell was less than 1×10^{-4}. Continuity, momentum, k and ε equations were solved to determine the flow field and then the energy equation to find the thermal field in the computational region. The convergence criterion for energy equation was taken 1×10^{-8}.

III.CALCULATIONS AND VALIDATION

The present numerical model was compared with the experimental study of Pise and Awasarmol [9].

$$q_1 = \int_0^L \left[-\frac{\partial T}{\partial x}\bigg|_{y=0}\right] dx_1 \; Q_1 = (LW) q_1 \qquad (11)$$

$$q_2 = -\frac{1}{Gr^{1/4}} \frac{\partial y}{\partial x}\bigg|_{x=0} \; Q_2 = (SW) q_2 \qquad (12)$$

$$q_3 = \int_0^L \left[-\frac{\partial T}{\partial y}\bigg|_{y=0}\right] dy_1 \; Q_3 = (LW) q_3 \qquad (13)$$

andwere heat fluxes from the fin and the base respectively. And Q_2were the heat flow rates from the fin and the base. The heat flow rate from the outer

face of the end fin (either first or the last in the array) of half-thickness t. Further the heat transfer rates from the end fins were assumed to be equal. The heat transfer rate from the fin array (Q_T) was given by

Q_T = (N-1) (2 Q_1 +Q_2) + 2 Q_3 (14)

Total heat transfer area for the N-fin array having breadth B was given by,

AT = (B - N t) W + 2 N L W

B = (N - 1) S + N t

The average heat transfer coefficient, h_m for the N-fin array for combined convection was defined by the following equation,

$$h_m = \frac{Q_T}{A(T_{W,Q} - T_\infty)} \quad (15)$$

In this paper same finned array with same configuration was simulated. The results obtained were shown in figure-4. It was found that the result shows good agreement between present numerical work and the experimental work.

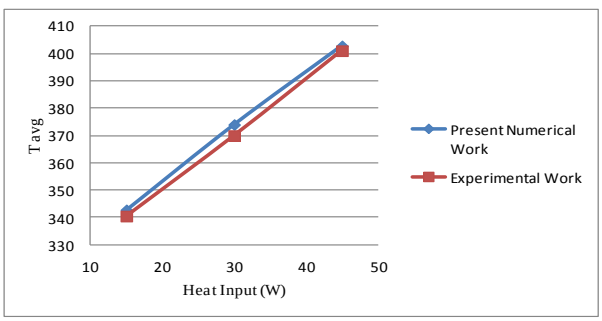

Fig 4 Comparison of present numerical data with the experimental data (Pise and Awasarmol, 2010)

IV. RESULTS AND DISCUSSION

The numerical results were obtained for a fin array of different heat input using validated model. Figures show the effect of Nusselt number at different heat input for solid and discrete fins respectively. Both the fins show a small increase in Nusselt number for all heat input range. Figure-5 and 6 shows temperature and velocity vector for 4 number of cut respectively. Figure-7 and 8 shows the comparison of solid and discrete fins for heat input. It was seen that Nusselt number was more for discrete fins as compare to solid fins. Discrete fins show 8% enhancement in Nusselt number. Figure 9 show the effect of heat input on heat transfer rate at various number of cut on solid and discrete fin array respectively, whereas Itwas seen that the Nusselt number for 45 W heat input shows improvement in Nusselt number that means high heat transfer.

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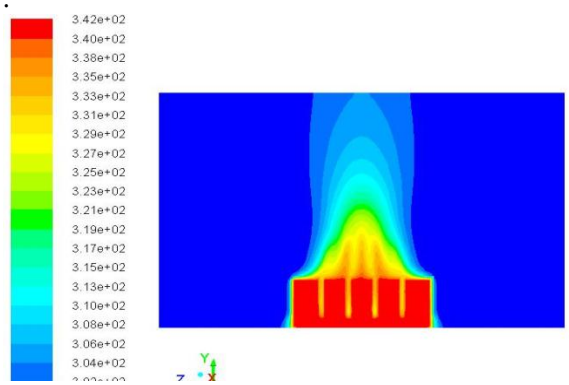

Fig.-5 Temperature counters of discrete fins of 4 No. of cut

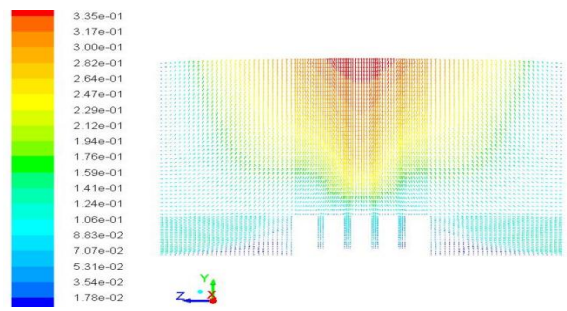

Fig.-6 Velocity vectors colour by velocity magnitude of discrete fins of 4 No. of cut

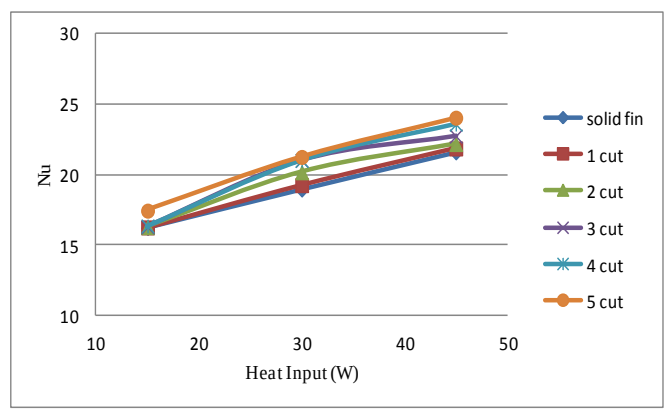

Fig.-7 Effect of No. of cut on of Nusselt number at various heat input

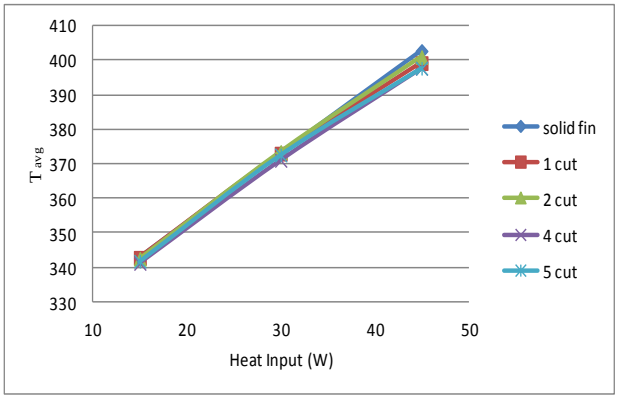

Fig.-8 Effect of No. of cut on fin temperature at various heat input

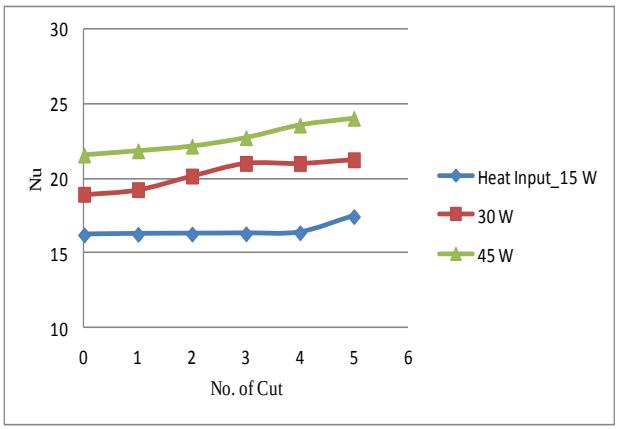

Fig.-9 Effect of heat input on Nusselt number

V. CONCLUSION

From the above discussion, following conclusions can be drawn:

1. Temperature profiles show that the base temperatures of solid fins were more elevated as compared to discrete fins. It means that for the same heat flux the discrete fins runs cool that shows that heat transfer rate was more in discrete fins as compared to solid fins.
2. Heat transfer rate of the block with discrete fins was enhanced by about 10% as compared to that of the cylinder block with solid fins
3. By using the discrete fins, the average heat transfer coefficient has been increased by significant value.
4. There was a reduction in area, which means the reduction in cost of the material, which was about 10-30 %. The material removed by mass in discrete fins was about 10 to 30%. Thus the cost of material saved was considered approximately 10 to 30%.

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