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Heat Transfer Augmentation Factor with Square Pin Fin Arrays

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***Abstract*—Theoretical studies and analysis on laminar natural convection heat transfer from a plain plate and square pin finned plate have been carried out in this study to predict the average Nusselt number in stream-wise, span-wise directions, fin aspect ratio and its inclination.**

Objective: The objective of the present patents is to provide more understanding on the distributions of local and average Nusselt numbers on the vertical plate with fins (conductive as well as non-conductive) in laminar natural convection. In this patent we also predict the average Nusselt number of the plate as a function of fin spacing in stream and span wise direction, aspect ratio of fins, fin height, fin thickness and angle of inclination of fins.

Methodology/approach: A finite volume numerical scheme is used in this patent to predict the flow field and the Nusselt number with the simultaneous computation of temperature field near the plate. Heat transfer behavior with both the in-line and staggered arrangement of fins have been analyzed by examining variations of the local and average Nusselt numbers.

Results: The maximum increase of the average Nusselt number is found to be around 36% for pin finned plate as compared to a plain plate for the same operating conditions. This is obtained with optimal fin spacing of $S_v/L = 0.2$ and $S_h/W = 0.25$, fin height of 24 mm (H/t=8), and angle of inclination of 45^o. The average Nusselt number decreases with increase in angle of inclination and also increases with increase in aspect ratio. Present study reveals that in-line and staggered arrangements do not yield appreciably different results. The maximum average Nusselt number difference between conductive and non-conductive fins is around 5 % for $S_h/W= 0.33$, Sv/L =0.2 at $\theta = 45^o$, fin height of 6 mm (H/t=2).

Research limitations/implications: A correlation has been developed to predict the average Nusselt number of the plate as a function of fin spacing in stream and span wise direction, aspect ratio of fins, fin height, fin thickness and angle of inclination in this patents.

***Index Terms*— Augmentation factor, heat transfer, natural convection, plate-fin heat exchanger, square pin fin, aspect ratio.**

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I. INTRODUCTION

Fins are the most effective instruments for increasing the rate of heat transfer from the heated surfaces. As the heat transfer surface area is increased by attaching fins to the heated surfaces, the heat transfer increases. Natural convection is a convenient, noise free and inexpensive way of cooling a hot surface by an adjoining fluid. Several attempts have been made to analyze the natural convection over a pin finned surfaces because of its theoretical interest and practical applications such as vaporization and condensation, It finds application in annular finned-tube heat exchangers, heat exchangers, hot water and steam pipes, heaters, refrigerators, chemical processing systems and electrical conductors, spray drying, nuclear reactor design, solar energy collectors, and in many electrical and electronic components, automobile engines, computer chips etc. Numerous analytical, numerical and experimental investigations have been conducted on natural convection heat transfer from pin finned rectangular plates. Other studies have considered factors such as the temperature distribution on the fin surface and the fin efficiency.

Abdul majeed S. A. Ghamdi.[1] analytically investigated Numerical Transient Heat Transfer in Rectangular Porous Fin Using MATLAB. They adopted Darcy's model to derive the energy equation governing the heat transfer process through the porous fins.

Baskaya et al. [2] carried out a numerical study of natural convection heat transfer from horizontal rectangular fin arrays. They utilized a CFD code to solve the three dimensional elliptic governing equations and found that the overall heat transfer is enhanced with increase in the height of the fin (*H*) and decrease in the length of the fin (*L*). In addition, for maximum heat transfer, optimum values of the fin spacing(*S)* was obtained. Fluid flow and heat transfer from horizontal rectangular thick fin arrays with short length were studied numerically by Dialameh et al. [3]. They found that natural convection heat transfer coefficient increases with increasing temperature difference and fin spacing, and decreases with fin length. Desrayaud and Fichera [4] studied numerically laminar natural convective flows in a vertical isothermal channel with two rectangular ribs. They specified that in the case of channel, increasing the length of the rib has only a limited influence on the heat transfer while increasing its width decreases the heat transfer drastically. Kim et al. [5] experimentally investigated the heat sinks with vertically oriented plate-fins. They presented the optimized dimensions and the fin efficiency, thus enabling the best fin to be designed for any practical use. In all of the above references, the optimum fin shape was determined based on either the minimum weight or the maximum heat transfer rate through the fin base. Fin-design problems based on the desired fin efficiency for a specified fin volume are very limited in the literature. Honda and Wei [6] made a noticeable progress in nucleate boiling enhancement by use of micro-pin-finned surface. Chin, Foo, Lai and Yong[7] studied numerically Forced convective heat transfer enhancement with perforated pin fins, B. Singh and S. K. Dash [8] studied numerically the natural convection heat transfer from a finned sphere in both laminar ($10^5 \leq Ra \leq 10^8$) and turbulent ($10^{10} \leq Ra \leq 10^{12}$) regimes by varying the fin-height-to-sphere-diameter ratio (H/D) and the fin-pitch-to-sphere-diameter ratio (P/D) in the range of 0.017–0.200 and 0.131–0.393 respectively. They found that for the sphere having conductive (Al) fins, with increasing number of fins, Nu decreases for laminar heat transfer and increases for turbulent heat transfer and for the sphere having non-conductive fins, Nu decreases with increasing number of fins in both laminar and turbulent heat transfer. Also, Nu increases with increasing fin height for conductive fins over the sphere and decreases for non-conductive fins. They developed correlations of Nusselt number for natural convective heat transfer from a finned sphere are developed with the pertinent input parameters like Ra, P/D and H/D in the range stated above. Although the fins are non-conductive but it will break the boundary layer for flow and heat transfer and hence can augment heat transfer from the surface by bringing fresh air from the ambient. So it is desired to find out the heat transfer augmentation factor for different configurations of the pin fins which are non-conductive. Sahoo et al. [9] Theoretically Analyzed the Steady Laminar Natural Convection Heat Transfer from a Pin Finned Isothermal Vertical Plate The non-conductive pin fins have an added advantage over the conductive pin fins. The advantage is their low weight which makes it possible to fix the pin fins through glue to any heat transfer surface where augmentation is required.

II. PHYSICAL DESCRIPTION OF THE PROBLEM

A vertical plate of height 0.7 m and width 0.6 m is studded with pin fins having base of 3 mm x 3 mm which are Non conductive either in a staggered manner or in in-line condition as has been shown in Fig.1.

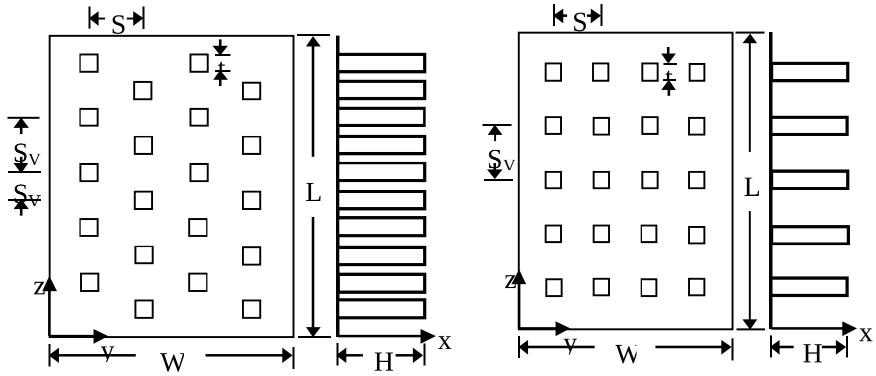

Fig :1 Physical configuration (a) staggered square pin fin array (b) in-line square pin fin array

III. MATHEMATICAL FORMULATION

The mathematical simulations for the present problem will call for the solution of the continuity equation, momentum equation along with the energy equation in the fluid as well as in the fin. But the solution will be done in 3-D mode because the fins are no longer continuous in the width direction or the span wise direction of the plate.
The following are the governing equations for the flow field and heat transfer around the vertical plate with fins.

Continuity Equation

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

Momentum Equation

$$\frac{D}{Dt}(\rho u_i)=-\frac{\partial p}{\partial x_i}+\frac{\partial}{\partial x_j}\left[\mu\left(\frac{\partial u_i}{\partial x_j}+\frac{\partial u_j}{\partial x_i}\right)\right]+(\rho-\rho_\infty)g_i \tag{2}$$

p in Eqn. 2 is a modified pressure defined as $p = p_s + \rho_\infty gz$, where p_s is the static pressure in the fluid domain. p in the domain will vary since ρ is a function of temperature in the buoyancy term.

Energy equation in fluid

$$\frac{D}{Dt}(\rho c_p T)=\frac{\partial}{\partial x_i}\left(k\frac{\partial T}{\partial x_i}\right) \tag{3}$$

Energy equation in fin

$$\frac{\partial}{\partial x_i}\left(\frac{\partial T}{\partial x_i}\right)=0 \tag{4}$$

Post processing Equations

The local heat transfer coefficient for unfinned portion of plate

$$h_y=-k_f\,\frac{1}{(T_w-T_\infty)}\left(\frac{\partial T}{\partial x}\right)_{x=0} \tag{5a}$$

The local Nusselt number for unfinned portion of plate

$$Nu_y = \frac{hy}{k_f} = -\frac{y}{(T_w - T_\infty)}\left(\frac{\partial T}{\partial x}\right)_{x=0} \tag{5b}$$

The local heat transfer coefficient for finned portion:

$$h_y = -k_m \frac{1}{(T_w - T_\infty)}\left(\frac{\partial T}{\partial x}\right)_{x=0} \tag{6a}$$

Finned portion material is different compared to that of the unfinned portion. So definition of Nusselt number and heat transfer coefficient will be different.

The local Nusselt number for finned portion:

$$Nu_y = -\frac{k_m y}{k_f (T_w - T_\infty)}\left(\frac{\partial T}{\partial x}\right)_{x=0} \tag{6b}$$

The average Nusselt number of plate with fin:

$$N\overline{u} = \frac{\overline{h}L}{k_f} = \frac{q_{plate}}{k_f (T_w - T_\infty)} \tag{7}$$

$$q_{plate} = q_{plate\_unfinned} + q_{plate\_finned} \tag{8}$$

In order to find out the heat transfer from the unfinned portion of the plate, a summation of $q_{plate\_unfinned}$ over all the unfinned portions is to be done.

$$q_{plate\_unfinned} = -k_f \int_{y_1}^{y_2} \left(\frac{\partial T}{\partial x}\right)_{x=0} dy \tag{9}$$

In order to know the total heat flow from the finned portion of the plate a summation of q_{plate_finned} over the entire finned portion is to be done.

$$q_{plate\_finned} = -k_m \int_{y_0}^{y_1} \left(\frac{\partial T}{\partial x}\right)_{x=0} dy$$

(for straight fins only) (10)

$$q_{inclined\_fin} = -k_m \sin^2\theta \int_{y_0}^{y_1} \left(\frac{\partial T}{\partial x}\right)_{x=0} dy$$

(for inclined fins) (11)

The heat transfer from the plate will be summation of q_{fin}+ $q_{un\text{-}fin}$, where as $q_{fin} = \int_{y1}^{y1+t}\int_{z1}^{z1+t} -k\frac{\partial T}{\partial x}dzdy$ is the heat transfer from the fin. In order to know the total heat being transferred from all the fins this integral has to be computed over all the fin bases and added while keeping the value of k to be that of the fin material. $q_{un\text{-}fin}$ is the heat loss from all the areas of the plate where there is no fin. This is the same integral like q_{fin} but computed over the region where there is no fin (k is of air). So special purpose double integration is needed to compute this integral and we used the help of the Fluent heat flow calculator which easily evaluates this type of integrals. After knowing the net heat transfer from the plate it is easy to compute the average Nusselt number for the plate.

IV. RESULTS AND DISCUSSIONS

Fig. 2(a) and 2(b) show the variation of average Nusselt number as a function of S_v/L and H/t of the fin when the fin configuration is inline. The variation for average Nusselt number is shown for two inclinations of the fin, namely $60^o$and 45^o. From the figure it can be seen that at 45^o inclination of the fin the average Nusselt is highest at $S_v/L = 0.2$. The fin height of 24 mm or H/t = 8 produces the highest average Nusselt number. When the fin inclination is 60^o the highest average Nusselt number decreases a little compared to the case of 45^o. In both these cases the increase in highest average Nusselt number compared to a plane plate is about 27-28%.

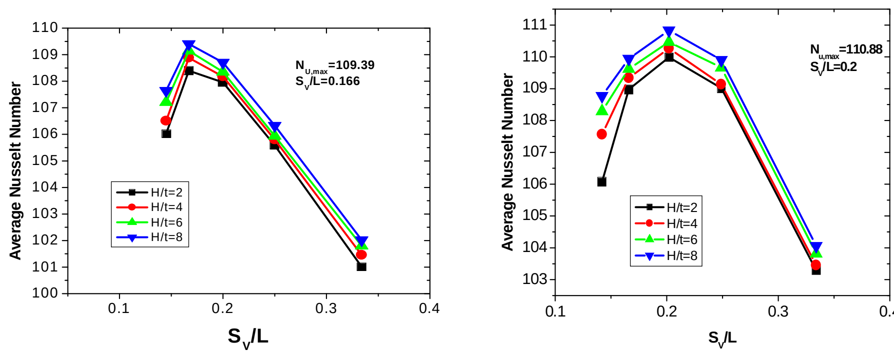

Fig. 2 Variation of the average Nusselt numbers with fin spacing in stream-wise direction for $S_h/W = 0.334$ at (a) $\Theta = 60°$, and (b) $\Theta = 45°$

Similarly Fig.3 (a) and 3(b) show the variation of average Nusselt number for the plate as a function of inclination height of the fins. At inclination of 45^o the average Nusselt number becomes highest. For the case shown in Fig. 3 (a) there is a minor effect of fin height on the average Nusselt number whereas for the case of 3(b) the effect of height of the fin is almost very marginal. The change in the highest average Nusselt number is very striking for the case of Fig.3 (b) where the change from a plane plate is about 36%.

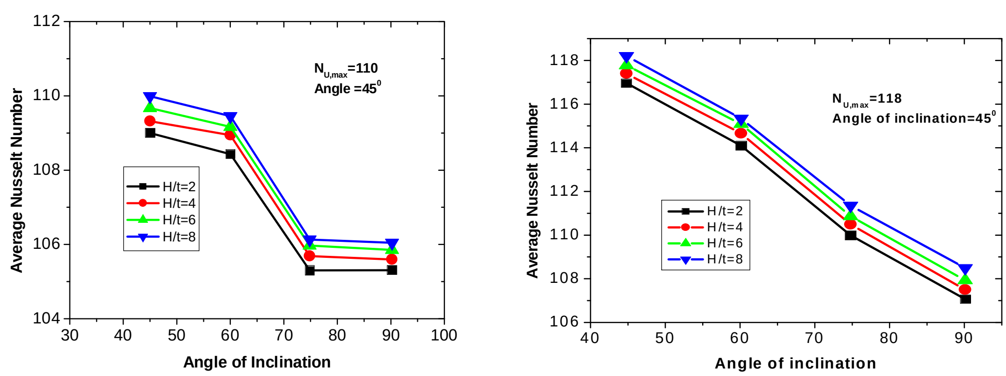

Fig. 3 Variation of the average Nusselt numbers with angle of inclination for (a) $S_v/L = 0.166$, $S_h/W = 0.334$, and (b) $S_v/L = 0.2$, $S_h/W = 0.25$

Fig.4 shows the variation of average Nusselt number for the plate as a function of H/t and inclination of the fin. It is clear from the figure that the average Nusselt number depends very much on the inclination of the fin as well as on its height. As the height of the fin increases the average Nusselt number increases always (in the range of parameters studied for the present case). At fin inclination of 45^o the average Nusselt number always attains the highest value. The highest value of average Nusselt number is about 29.50% higher than that for a plain plate.

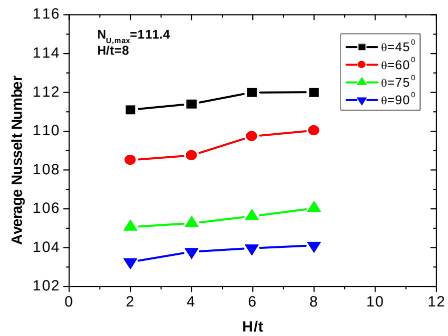

Fig. 4 Variation of the average Nusselt number with aspect ratio for S_v/L= 0.333 and S_h/W = 0.2

V. General Correlation developed for the average Nusselt number

The main objective of the present work is to predict the heat transfer augmentation factor with non-conductive fins and develop a correlation for different fin configurations so that the correlation can be useful to the paint industries in particularand all other industries in general. we could develop a correlation from the present CFD computation which can predict the average Nusseltnumber for a plate with many non-conductive pin fins either in inline or in staggered arrangement. Equation (12) shows the general correlation for the average Nusseltnumber where $\overline{Nu}_{pp}$ is the average Nusselt number for a plain plate with no fins.This correlation would be valid for Grashof number starting from 10^5 to $9.3\text{x}10^8$. If the fin configuration is known then one can get the predicted Nusselt number for the plate from equation (12). The errors that resulted from the correlation are shown in Table 2. The error is computed by taking the CFD result as the base. From the table it is clear that the errors are pretty less for any practical engineering application. The correlation has been developed from 240 CFD runs for staggered and 240 CFD runs for inline arrangements of fins.

$$\overline{Nu}_{pred} = \overline{Nu}_{pp}\left[\frac{\theta^a\left\{b + c\left(\frac{S_v}{L}\right) + d\left(\frac{S_v}{L}\right)^2\right\}\left(\frac{S_h}{W}\right)^e}{\left(\frac{H}{t}\right)^f}\right] \tag{12}$$

TABLE I. Coefficients of the correlation for different fin arrangements

Coefficients	Nonconductive inline	Nonconductive staggered
a	-0.103	-0.103
b	0.987	0.978
c	0.583	0.584
d	-1.700	-1.600
e	-0.164	-0.168
f	-0.006	-0.007

Fin arrangements	Min. Error (%)	Max. Error (%)
Nonconductive inline	-3.4	7.5
Nonconductive staggered	-3.9	4.4

VI. Comparison of numerical results with the developed correlation Eq.(12)

Fig.5 shows the comparison of the average Nusselt number with that of the correlation developed for many cases of the fin configurations. From the figures it is clear that the average Nusselt number predicted from the correlation is in pretty good agreement with the direct Nusselt numbers obtained from the CFD computations. Fig.6 shows the predicted versus the actual Nusselt number between the error limits of 5% and -5%. It can be seen that there is no data point which goes to the level of 5% error.

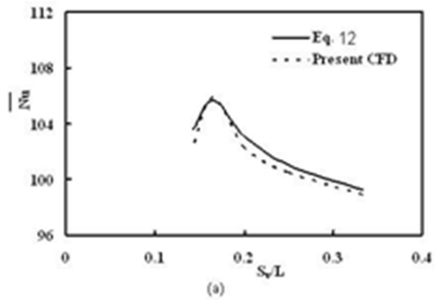

Fig. 5 Numerical and predicted Nusselt number for Sh/W =0.334, H / t =6, ϴ = 90°

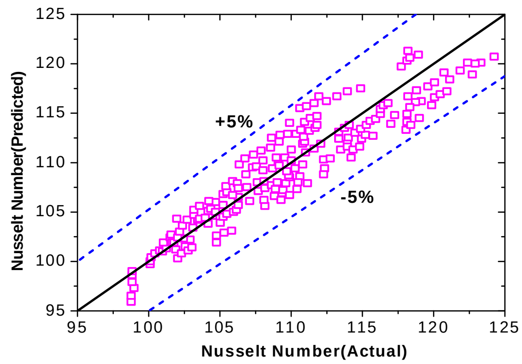

Fig.6 Comparison of numerical and predicted Nusselt numbers for staggered arrangements of fins

VII. CONCLUSIONS

Numerical studies on laminar natural convection near a plain plate and on a pin finned plate has been carried out to predict the average Nusselt number for the plate for many different configurations of the pin fins. Before the results could be taken for a pin finned plate numerical validations for a known pin finned plate was done to ascertain numerical accuracy of the method. Following conclusions from the present study can be arrived at:

1 The maximum increase of the average Nusselt number is around 36% for pin finned plate as compared to a plain plate for the same operating conditions. This is obtained with optimal fin spacing of $S_v/L = 0.2$ and $S_h/W = 0.25$, fin height of 24 mm (H/t=8), and angle of inclination of 45^o.
2 The average Nusselt number decreases with increasing in angle of inclination.
3 The average Nusselt number increases with increase in aspect ratio.
4 Present study reveals that in-line and staggered arrangements do not yield appreciably different results.
5 The maximum average Nusselt number difference between conductive and non-conductive fins is around 2.75% for $S_h/W= 0.33$, $S_v/L =0.2$ at $\theta = 45^o$, fin height of 6 mm (H/t=2). It means that fin material has not got much influence on the average Nusselt number for the case studied.
6 Based on all computations, one correlation is proposed to predict the average Nusselt number of an array of non-conductive fins based on parameters $\overline{Nu}_{pp}$, θ, S_v/L, S_h/W, H/t for two types of fin arrangements.

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