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Computational Analysis of Natural Convection Heat Transfer From Pin Finned Plate

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ABSTRACT: *Purpose: The objective of this study is to quantify and compare the natural convection heat transfer enhancement of fin array with different fin aspect ratio and at different angles of inclination. Design/methodology/approach: A finite volume numerical scheme is used to predict the flow field and the Nusselt number near the plate with the simultaneous computation of temperature field near the plate. Heat transfer behavior with both the conductive and non-conductive fins have been analyzed by examining variations of the local and average Nusselt numbers in two-dimensional flow. Findings: Significant heat transfer augmentation is obtained for both conductive and non-conductive fins. For conductive fins 20% higher augmentation factor is obtained when the fin aspect ratio is 6 angle of inclination is 60° and the pitch-to-length ratio is 0.2. For non-conductive fins 10% higher augmentation factor is obtained when fin aspect ratio is 8 angle of inclination is 45° and pitch-to-length ratio at 0.5. Research limitations/implications: A general correlation has been developed to predict the average Nusselt number and heat transfer augmentation factor for conductive and non-conductive fin arrays as a function of different fin configurations. Practical implications: The correlations developed could be of use to industry producing plate heat exchangers for paint shops. Originality/value: The effects of fin geometry and the angle of inclination of the fins on the natural convection heat transfer from a vertical isothermal flat plate with fins have been investigated numerically and correlations are developed to predict the average Nusselt number as a function of fin geometry, aspect ratio & angle of inclination.*

KEYWORDS: *Augmentation factor, heat transfer, natural convection, plate-fin heat exchanger, square pin fin, aspect ratio*

I. INTRODUCTION

A pin fin is a cylindrical or other shaped protrusion attached to a wall, with the transfer fluid passing in a cross flow manner over the protrusion. There are various parameters that characterize the pin fins, such as shape, height, thickness, height to thickness ratio etc. In addition, the pin fins may be positioned in arrays that are either in-line or staggered with respect to the flow direction. The heat transfer in a conductive-finned-plate heat exchanger is a conjugate problem. Conjugate heat transfer means computing more than one mode of the heat transfer simultaneously (conduction + convection) and it can be established efficiently by the way of numerical analysis. For a finned-plate heat exchanger, when the convection effect is intended to be used to enhance the heat transfer from the plate, the conduction in the fins has to be considered as well. For a better understanding of the most important mechanisms of heat transfer in flow passing through the finned-plate exchangers; numerical simulations emerge normally to be a very helpful tool. Especially, one can compute the heat transfer coefficient from pin finned surfaces by means of the numerical simulation and propose general purpose correlations for the heat exchangers which can be used by the industries.

Bassam (2003, investigated numerically the natural convection heat transfer through permeable fins and found that the heat transfer through permeable fins resulted in significant enhancement over solid fins. They stated that increase of number of permeable fins always resulted in increase in Nusselt number unlike in solid fins. They used certain assumptions to make the analysis simple that the fins are made up of highly conducting material. They did not validate their results with experimental work. Ridouane and Campo [3] in their study showed the enhancement in heat transfer using grooved channels. They found that the grooves enhance the local heat transfer as compared to flat passages.

Jamin and Mohamad [4] quantified and compared the steady state heat transfer from a heated vertical pipe with and without porous medium. They found that the largest increase in Nusselt number was achieved by high thermal conductivity solid carbon foam sleeve, which was about 2.5 times greater than a bare copper pipe. Ahn et al. [5] in their experiment compared the heat transfer rates with rounded and elongated holes in rectangular plate. They showed that elongated holes enhance heat transfer rate more than rounded holes but at the cost of pressure drop. Layeghi [6] in his numerical analysis also showed that heat transfer can be enhanced using porous media, but with an increase in pressure drop. Ben-Nakhi et al. [7] studied the natural convection in open cavity. They found that the heat transfer rate increases with the thin fins attached to the hot surfaces.

Starner and McManus [8] presented one of the earliest experimental investigations for heat transfer coefficients of natural convection heat transfer from fin arrays. They found that for short fins, horizontal orientation is more suitable whereas the vertical orientation is better in the case of long fins. Sparrow and Vemuri [9] studied the fin orientation and the effects of fin population on natural convection/radiation heat transfer from pin fin arrays. They concluded that the total heat transfer initially increases with fin population and then decreases beyond an optimum value of fin population. Zografos and Sunderland [10] investigated experimentally the natural convection heat transfer performance of in-line and staggered pin fin arrays. They found that the in-line arrays generally yield higher heat transfer rates than the staggered ones. Their investigation suggested that the most important geometric parameter is the ratio of the diameter to the centre-to-centre spacing of pin fin and an optimal value of 1/3 is reported. Guvenc and Yuncu [11] reported the experimental studies on natural convection heat transfer of rectangular fins attached to horizontal or vertical surfaces. They found that mounting rectangular fins on a vertical plate is a more preferable solution than that on a horizontal plate for enhancement of heat transfer. Sikka et al. [12] performed experiments on various kinds of heat sinks under natural and forced convection. Their results showed that pin fin heat sinks outperform those with plate fin, fluted fin, or wavy-plate fin heat sinks.

The objective of the present study 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. This numerical investigation has been carried out for the Rayleigh number of 108 (based on vertical plate length). A finite volume numerical scheme is used to predict the flow field near the plate and the Nusselt number with the simultaneous computation of temperature field near the plate. The influence of the geometric parameters of pin fins on local and average Nusselt number for vertical isothermal plate was studied and general correlations were developed so that the results could be of use to industry producing plate heat exchangers for paint shops.

II. MATHEMATICAL FORMULATION

In order to solve the flow and the temperature field around the finned plate it is required to use the Navier-Stokes equation with buoyancy driven terms in the y-momentum equation along with the energy equation which will solve for the temperature field in the domain as well as in the fin (provided it is conductive). If the fin is non-conductive (k = 0) we will simply take out the non-conductive portion of the fin from the computational domain and attach a boundary condition on the surface of the non-conductive fin.

The following assumptions were made in the mathematical formulation and the solution process.

1. The flow field is steady, laminar and incompressible with the fluid stresses being Newtonian.
2. The properties of the fluid (μ, k, c_p and ρ) are kept constant at free stream conditions in all the equations except for the density of air, ρ, in the buoyancy term of the momentum equation only where it is assumed to be a function of temperature so that the buoyancy force can be induced in the fluid. The Boussinesq approximation for the buoyancy is not assumed in the present study; rather density of air is taken to be a straight function of temperature through the ideal gas law and is provided as a table to the computing software. The temperature difference between the plate and the surroundings fluid is of the order of 25 °C, so Boussinesq approximation is not used. The computation of density from the ideal gas law is done by assuming the pressure to be atmospheric only since the equipments work in the atmospheric conditions.
3. Radiation from the plate and from the fin is also neglected because the plate temperature is hardly 25 °C more than the ambient temperature.

With the above assumptions the following are the governing equations for the flow field and heat transfer around the vertical plate with fins.

Continuity Equation:

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

***Momentum Equation*:**

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

p in Eqn. 2 is a modified pressure defined as $p = p_s + \rho_\infty g z$, 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 (elsewhere it is constant).

Energy equation in fluid:

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

Energy equation in fin:

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

Post processing Equations:

The local heat transfer coefficient for unfinned portion of plate:

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

The local Nusselt number for unfinned portion of plate:

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

The local heat transfer coefficient for finned portion:

$$h_y = -k_m \frac{1}{(T_w - T_\infty)}\left(\frac{\partial T}{\partial x}\right)_{x=0} \qquad (7)$$

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:

$$Nu_y = -\frac{k_m y}{k_f (T_w - T_\infty)}\left(\frac{\partial T}{\partial x}\right)_{x=0} \qquad (8)$$

The average Nusselt number of plate with fin:

$$N\overline{u} = \frac{\overline{h}L}{k_f} = \frac{q_{plate}}{k_f (T_w - T_\infty)} \qquad (9)$$

$$q_{plate} = q_{plate\_unfinned} + q_{plate\_finned} \qquad (10)$$

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.

$$q_{plate\_unfinned} = -k_f \int_{y_1}^{y_2}\left(\frac{\partial T}{\partial x}\right)_{x=0} dy \qquad (11)$$

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.

$$q_{plate\_finned} = -k_m \int_{y_0}^{y_1}\left(\frac{\partial T}{\partial x}\right)_{x=0} dy \quad \text{(for straight fins only)} \qquad (12)$$

$$q_{inclined\_fin} = -k_m \sin^2\theta \int_{y_0}^{y_1}\left(\frac{\partial T}{\partial x}\right)_{x=0} dy \quad \text{(for inclined fins)} \qquad (13)$$

Boundary conditions

The boundary conditions for the solutions of Equations (1), (2) and (3) are shown pictorially in Fig. 1

At the wall (Heated plate)

All velocity components are zero and, $T = T_w$ (14)

At the inlet

$$U = 0, \quad \mathrm{V} = 0, \quad \text{and } \mathrm{T} = \mathrm{T}_\infty \qquad (15)$$

At the pressure outlet boundary
Top pressure outlet

$p = 0$, V is determined from local pressure field and U is found out to satisfy continuity equation, $\frac{\partial T}{\partial y} = 0$, (16)

On the pressure outlet boundary the flow can come in to the domain or can go out of the domain depending on the local pressure field. The velocity perpendicular to the boundary can be computed from pressure conditions and a realistic flow field can be established through this boundary condition. Other boundary conditions cannot be applied to a boundary where the flow either goes out of the domain or comes in to the domain

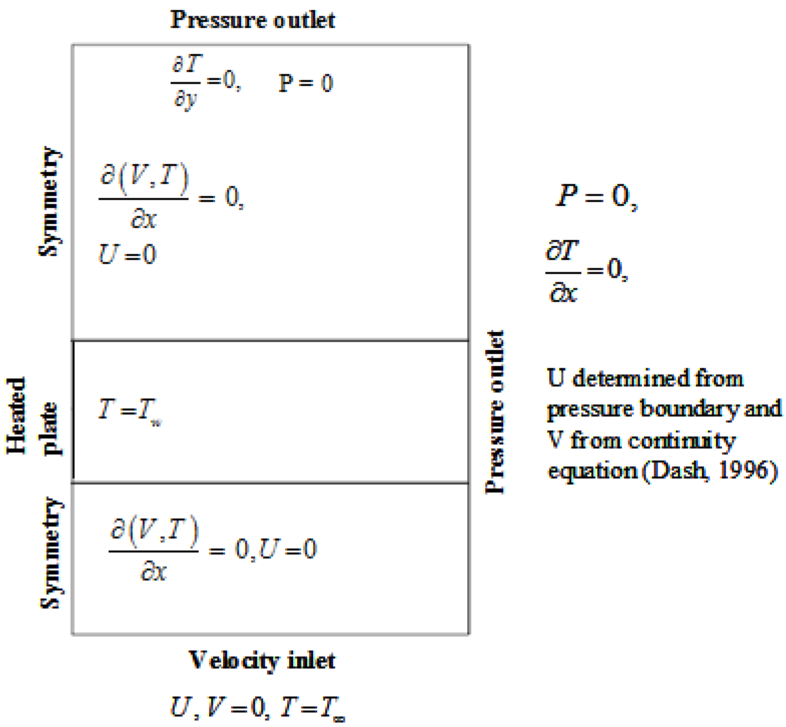


Fig. 1 Boundary conditions

Side pressure outlet

$P = 0$, U is determined from local pressure field and V is found out to satisfy continuity equation, either $T = T_\infty$ or $\frac{\partial T}{\partial x} = 0$ could be used. (17)

At the symmetry boundary

$$\frac{\partial (V,T)}{\partial x} = 0 \;\; and \;\; U = 0 \tag{18}$$

At the fin

The base of the fin is having a known temperature of T_w at x = 0. At x = H; $-kdT/dx$ = heat lost to surrounding fluid which is taken care of by the fluid flow near the adjacent cell at the tip of the fin. Near the surface $y = y_0$, and y_1 the heat lost by conduction from the fin is convected to the ambient fluid. So, the specification of heat transfer coefficient is not required at the fin surfaces which are taken care of by the fluid flow and energy equation through a conjugate heat transfer mechanism. For a non-conductive fin the surfaces of the fin are assumed to be adiabatic since no heat flow occurs from the fin to the ambient.

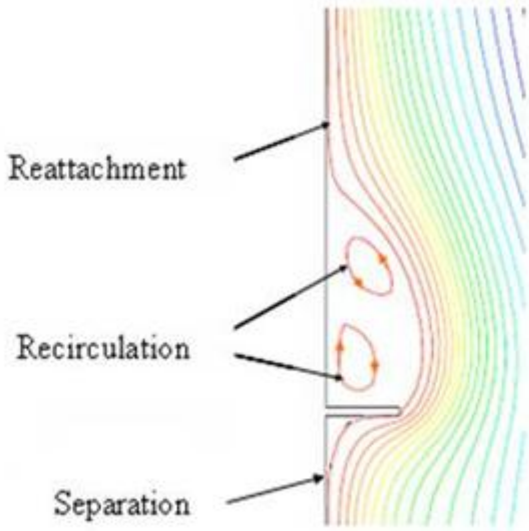


Fig 2 : separation & reattachment of the flow around a fin

Heat exchange device performance is often limited by the air side because the heat transfer coefficients are inherently lower. The air side temperature distribution is intimately coupled to the velocity field, often taking the form of a thermal boundary layer. This temperature distribution is a manifestation of the air-side heat resistance, and it can be modified through roughness elements (fins). The roughness elements (fins) attached horizontally to a vertical flat plate, stagnate the flow and a separation from the plate occurs at the rear of the fin (Fig. 2). The stagnation and the separation of the flow followed by the reattachment, affect heat transfer characteristics. This heat transfer characteristics are greatly influenced by surface geometry and orientation of fins.

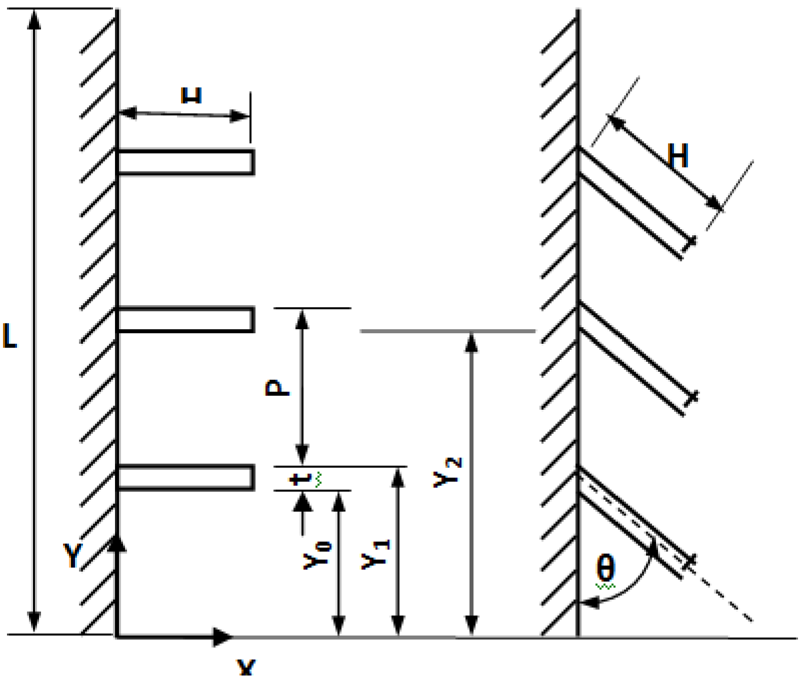


Fig. 3 Schematic diagram of the heated plate with straight and inclined fins

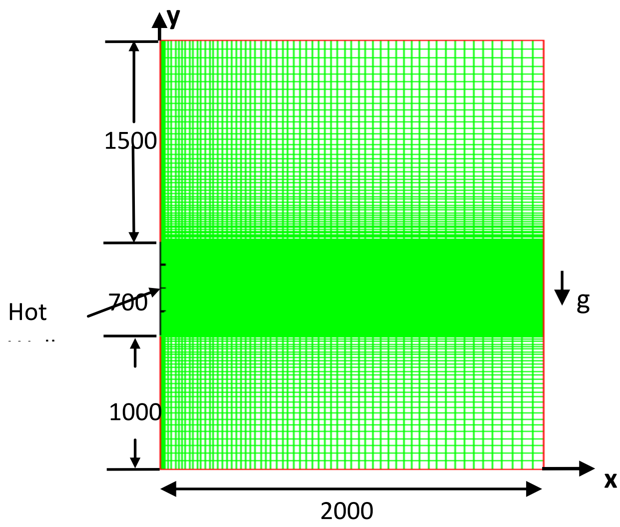


Fig. 4 Computational domain and cell arrangement in the domain with three fins

The schematic diagram of a heated vertical flat plate with fins is shown in Fig.3. The co-ordinate system in which the plate is fixed and the computations were carried out is also shown in the Fig 4. The plate is fixed with some fins which have a height H, thickness t and fin separation pitch between two consecutive fins is P.

The objective is to find out the net heat transfer from the plate in laminar natural convection and hence determine the average Nusselt number for the plate and finally predict the heat transfer augmentation factor as a function of fin configuration. Normally such type plate heat exchangers are used in paint shops of car manufacturing units and they are limited to a height of 0.7 m and at best a temperature difference of 25°C exists between the plate and the surroundings where it is expected that the flow around the plate would be laminar (Ra < 109) and the heat transfer would be governed by laminar natural convection. Even if the surface has fins and there are vortices behind the fin the average flow velocity is very small in natural convection, particularly for the case considered. So the vortices will not produce turbulence because the viscous effects will suppress the turbulence production keeping the flow essentially to laminar. If a heat transfer augmentation factor for different fin configuration is known, then fins can occasionally be used on the plates or even permanently without adding any extra cost to the device.

III. RESULTS AND DISCUSSIONS

In the numerical solution procedure, we have outlined the grid sensitivity of the solution process. In Fig. 5 the average Nusselt number of the plate with three fins have been shown as a function of grid refinement or number of cells. The three fins were having different heights of 2, 4, 6 and 8 times their thickness with a P/L = 0.25. The fins are at 90o to the plate and the average Nusselt number sharply increases with the change in grids (cells becoming smaller). After a cell of 25 000 the average Nusselt number increases very slowly and there is practically no change in the Nusselt number when the total cells increased from 30 000 to 60 000. For our computation we chose roughly a total number of cells around 36 000 which gives pretty accurate Nusselt number.

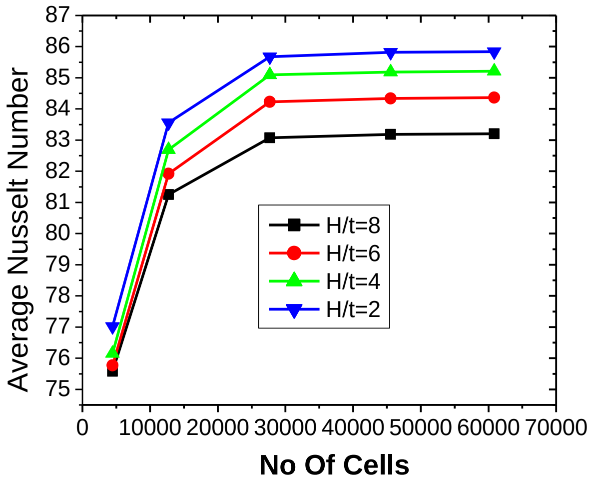
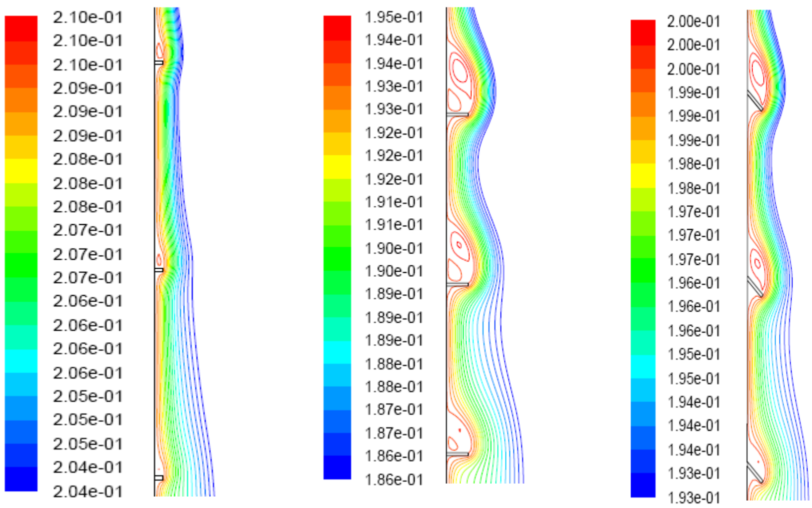


Fig. 5 AverageNusselt number as a function of the grid size with three fins of height = 6, 12, 18 and 24 mm with P/L = 0.25



(a) H / t = 2, (b) H / t = 8; ϴ = 90°, (c) H / t = 8; ϴ = 45°

Fig.6 Streamline patterns near heated vertical plate with different fin configuration

Fig.6 shows the streamline patterns near the heated vertical flat plate with different fins. Fig.6(a) shows a fin of 6 mm height, where the ambient fluid diverts a bit from its straight path and then again reattaches to the vertical plate at a certain distance from the fin. Due to the reattachment the local Nusselt number on the plate rises a bit. When the fin height is 24 mm (H/t = 8) the stream of fluid diverts much away from the plate and reattaches to the plate at a higher length compared to the case of the fin having a height of only 6 mm. Due to a large deflection of the stream in to the ambient, the stream gets cold air from the ambient which reattaches to the plate thus increasing the local Nusselt number on the plate in the un-finned portion. Fig.6(c) shows the streamline pattern of the fluid near the plate having fins of 24 mm height but inclined at an angle of 45o. Here, the stream does not get deflected much in to the ambient but the reattachment length of the diverted stream is longer compared to the case of a straight fin shown in Fig.6(b).

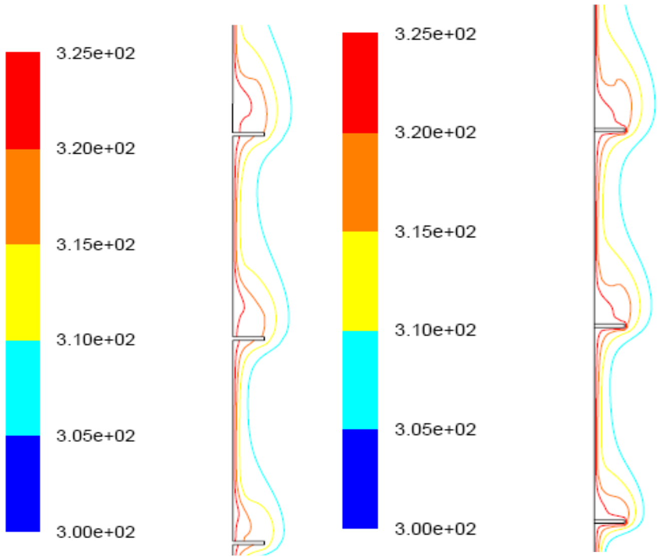


Fig. 7 Isotherm lines(K): (a) nonconductive fins and (b) conductive fins

Fig.7 (a) shows the isotherm lines near the vertical plate which has three non-conductive fins on it (H/t = 8 and P/L = 0.25) and Fig.7 (b) shows the isotherms for three conductive fins. It can be seen from the figure that the isotherms start from one fin and ends at the other fin and they are discontinuous around the non-conductive fin where as for the conductive fin the isotherms are continuous around the fins. When the fin is non-conductive the temperature inside it is always ambient. The temperature of air very near the plate would be normally less than that of the plate. So when a contour of temperature line is made between two successive fins, there will be a discontinuity at the surface of the non-conductive fin so the isotherms look discontinuous (because the entire non-conductive fin is at a temperature of T_{∞}). But for a conductive fin the temperature inside the fin is computed and that is slightly lower than the plate temperature, so there exists continuity to some extent in the temperature between the fin and the surrounding fluid (although strictly there is a discontinuity in temperature from the air to the fin which cannot be seen here due to the fin being too small compared to the size of the plate and the surroundings). So the isotherm lines look continuous for the case of conductive fins.

Velocity field near the plate with fins having height to thickness ratio = 8 and different angles of inclination (i) $\theta = 90^0$, (ii) $\theta = 75^0$, (iii) $\theta = 60^0$ and (iv) $\theta = 45^0$ are illustrated in Fig. 8

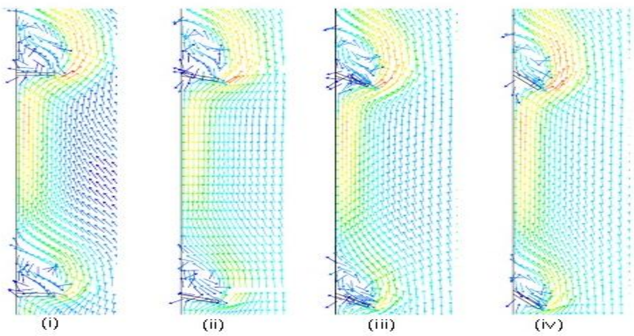


Fig. 8 Velocity field near the plate with fins (H/t = 8) having different inclination (i) θ = 90°, (ii) θ = 75°, (iii) θ =60° and (iv) θ = 45°

Fig. 9 shows the variation of the average Nusselt number for the plate with different fin spacing and angle of inclination of the fins. The average Nusselt number increases with increasing fin spacing, attaining a maximum value of 97.64 at P/L = 0.2 at ϴ = 60°. It then decreases slowly as the fin spacing continues to increase. As the fin spacing rises there will be less number of fins on the plate and the flow will develop in between two successive fins and the heat transfer coefficient will fall causing the local Nusselt number to be less and hence the average Nusselt number will fall. For a particular fin spacing the average Nusselt number can attain a maximum value which can be considered to be the optimum fin spacing.

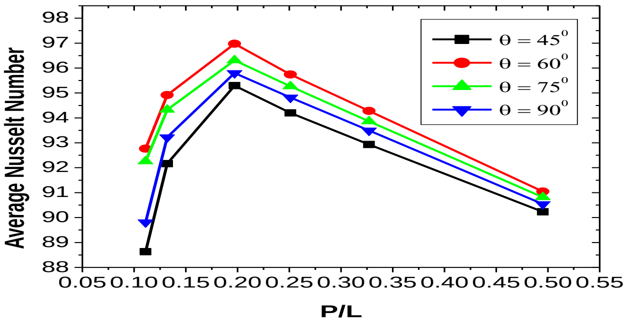


Fig. 9 Effect of fin spacing on average Nusselt number for aspect ratio of H / t = 6

It can be seen that there exists an optimum inclination of the fins to get highest average Nusselt number. When the inclination of the fins changes, the flow around the fin gets deflected and tends to glide along the vertical plate so the reattachment length of the stream changes on the plate. The reattached fluid stream can withdraw different amount of heat from the plate, depending on its length, thus causing the average Nusselt number for the plate to change with the inclination of the fin and hence highest average Nusselt number can be obtained for a particular inclination of the fin. It was observed that for an inclination angle of 60^0, highest average Nusselt number was obtained.

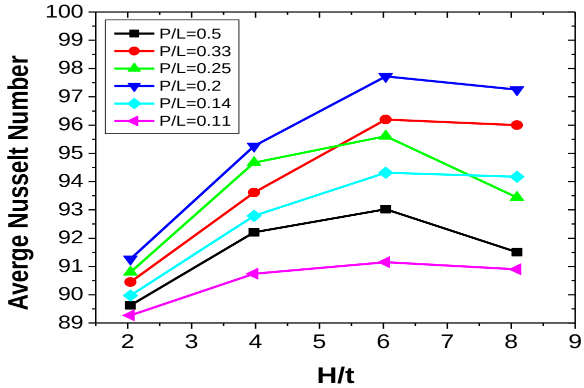


Fig. 10(a)Average Nusselt number as a function of fin aspect ratio for (a) ϴ = 60°

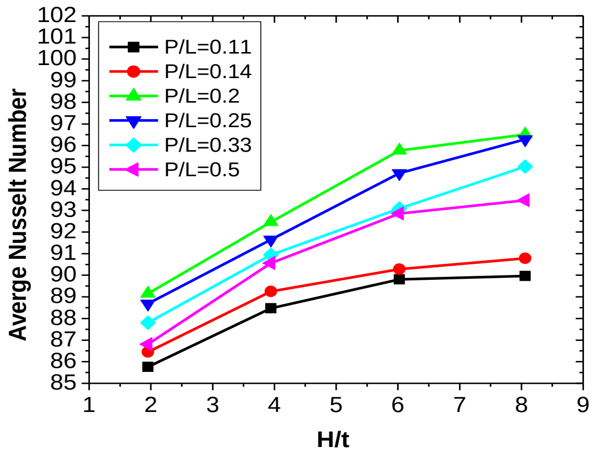


Fig. 10(b) Average Nusselt number as a function of fin aspect ratio for (b) ϴ = 45°

Fig.10(a) and (b) show the variation of average Nusselt number on the plate as a function of H/t when P/L varies from 0.5 to 0.11. When P/L = 0.5 there is only one fin and for P/L = 0.25 there are 3 fins and for P/L = 0.2 there are 4 fins. This means when P/L decreases number of fins increases. From the figure it is clear that for fin inclination of 60^o, 18 mm fins produce highest average Nusselt number compared to any other fin heights and for fin inclination of 45°, 24 mm fins produce higher average Nusselt number. For both the cases there is optimum fin spacing to get the highest average Nusselt number. For fin inclination of 60° the optimum fin spacing, P/L= 0.2 and for 45° inclination the optimum fin spacing, P/L = 0.25. When the number of fins is small, the flow will develop in between two successive fins and the heat transfer coefficient will fall with the height of the plate. The main idea of putting fin is to break this developing flow and bring the ambient fluid again closer to the plate which can take away more heat from the plate as the temperature gradient is more.

Development of a new correlation for average Nusselt number

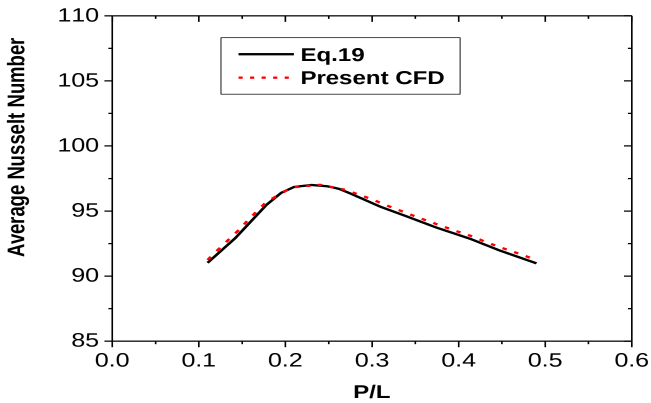
For the two dimensional simulation of the present numerical study, an empirical correlation for the average Nusselt number is developed for the vertical isothermal plate with conductive and nonconductive fins based on the present computation. For the case of conductive fins there are a total of 96 computed data points (average Nusselt number) and similarly for the case of non-conductive fins there are also 96 computed data points. The objective is to bring out a correlation for the average Nusselt number which is a function of P/L, H/t and θ of the fin. The Prandtl number is constant at 0.7 and the Grashof number can be up to a maximum of 10^8 so that the value of $\overline{Nu}_{pp}$ can be determined from some other correlation for a plain plate without fins. Such a correlation will be useful for the actual industry. Eq. 19 shows the predicted Nusselt number for a plate with fins as a function different fin configurations. $\overline{Nu}_{pp}$ is the average Nusselt number for the plain plate without any fins. The right hand side term next to $\overline{Nu}_{pp}$ is the augmentation factor which is determined from this study.

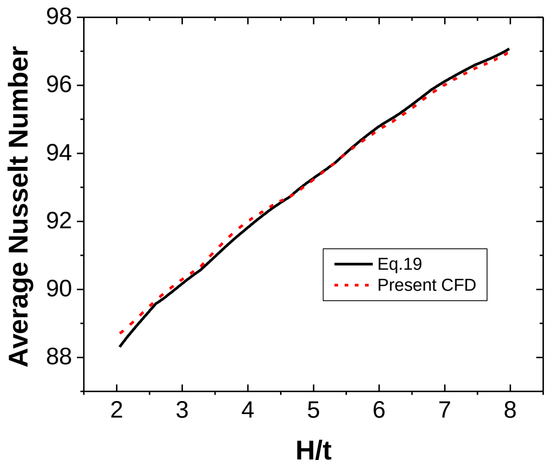
$$\overline{Nu}_{Pred} = \overline{Nu}_{PP}\left[\frac{\left(a+b\theta+c\theta^2\right)\left\{d+e\left(\frac{P}{L}\right)+f\left(\frac{P}{L}\right)^2\right\}}{g+\left(\frac{H}{t}\right)^h}\right] \quad (19)$$

It can be seen from Eq. 19 that the dependence of the average Nusselt number is quadratic on θ and P/L and logarithmic on H/t. The average Nusselt number for the conductive fins has a maxima against the variable θ and P/L so a quadratic dependence on these variables have been used in the correlation. The constants a, b, c, d, e, f, g, and h are tabulated in the table 1. The present correlation gives a very good numerical match with the present computed CFD result. For the conductive fins all the predicted data points fall within an error limit of 5% (CFD $\overline{Nu}$ is the base). For the non-conductive fins Eq. 19 predicts the $\overline{Nu}$ to an error limit of 5% for 92 data points, whereas for only 4 data points the error goes to a maximum of 9%. Eq. 19 has been developed from the Engineering Equation Solvers (EES) software with our present CFD result.

Comparison of numerical results with the developed correlation Eq. 19 (conductive fins)

Fig.15 shows the average Nusselt number for the plate as a function of fin inclination θ with other parameters of H/t and P/L. The computed value from the present correlation Eq. 19 is also shown for comparison on the same plot. It can be seen that the comparison between the correlation Eq. 19 and the direct CFD data are in very good agreement with each other.





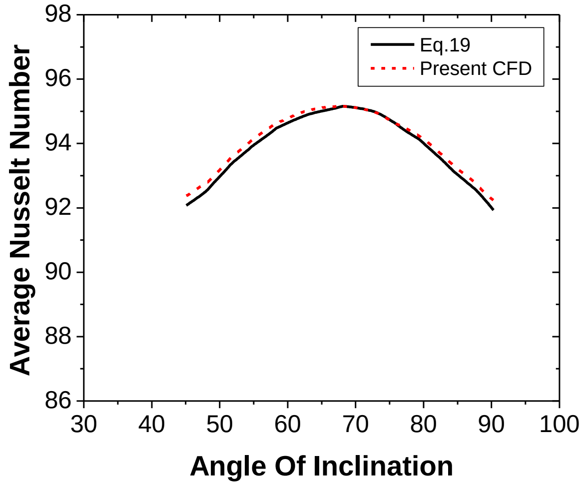


Fig. 15 Numerical and predicted Nusselt numbers for (a) H/t = 8, ϴ = 90°, (b) P/L = 0.25, ϴ = 45°, and (c) H/t = 4, P/L = 0.2; conductive fins

IV. CONCLUSIONS

The effects of fin geometry and the angle of inclination of the fins on the natural convection heat transfer from a vertical flat plate with fins have been investigated numerically. The most important results of the simulation are as follows.

(1) The maximum increase in the average Nusselt number for a finned plate is around 20% as compared to a plain plate for the same operating conditions. This is obtained with pitch- to- length ratio (P/L) of 0.2, fin height of 18 mm, and angle of inclination of 60°.
(2) The average Nusselt number increases with increase in fin spacing. It attains a maximum value for particular fin spacing and then decreases with increase in fin spacing.
(3) With increase in the inclination angle, the average Nusselt number initially increases up to a certain value of inclination angle and subsequently decreases with increase in inclination angle.
(4) A correlation is developed to predict the average Nusselt number of the plate as a function of fin spacing, aspect ratio & angle of inclination.
(5) The positions of minimum Nusselt number and the reattachment point upstream of a rib

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