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NATURAL CONVECTION HEAT TRANSFER FROM FIN ARRAYS-EXPERIMENTAL AND THEORETICALAL STUDY ON EFFECT OF INCLINATION OF BASE ON HEAT TRANSFER

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ABSTRACT

The problem of natural convection heat transfer from fin arrays with inclination is studied experimentally and theoretically to find the effect of inclination of the base of the fin array on heat transfer rate. A numerical model is developed by taking an enclosure, which is formed by two adjacent vertical fins and horizontal base. Results obtained from this enclosure are used to predict heat transfer rate from the fin array. All the governing equations related to fluid in the enclosure, together with the heat conduction equation in both the fins are solved by using Alternate Direct Implicit method. Numerical results are obtained for temperature along the length of the fin and in the fluid in the enclosure. The experimental studies have been also carried out on two geometric orientations viz., (a) vertical base with vertical fins (vertical fin array) and (b) horizontal base with vertical fins (horizontal fin array), with the five different inclinations like 0^0, 30^0, 45^0, 60^0, and 90^0. The experimental results are compared with the numerical results computed by the theoretical analysis shows the good agreement.

Keywords: natural convection, heat transfer, vertical fin array, inclined fin array.

1. INTRODUCTION

Fins are extensively used in air cooled automobile engines, air craft engines, cooling of generators, motors, transformers, refrigerators, cooling of computer processors and other electronic devices etc. Previously, a great number of experimental and numerical works has been carried out to study the effect of fin parameters like fin height, fin spacing etc. on heat transfer rate from fin array by the investigators. Elenbaas [5] conducted an experimental study on heat dissipation of parallel plates by free convection for wide range of Rayliegh numbers, viz., $0.2 < Ra < 10^5$. He determined that in the limit of small gap width, Nusselt number varies proportional to the channel Rayliegh number (Ra).

Experimental work on horizontal fin arrays was studied by [2, 10, 13-15, 17, 18]. Numerical work on horizontal fin arrays was studied by [4, 19]. Experimental work on vertical fin arrays was studied by [6, 8]. Numerical work on vertical fin arrays was studied by [9, 12]. Experimental work on downward facing fin arrays was studied by Dyan *et al.*, [3]. Experimental work on vertical base and width-wise horizontal fins was studied by Bilitzky [1]. Starner and McManus [16] did an experimental investigation of free convection heat transfer from rectangular fin arrays. Average heat transfer coefficients were presented for four fin arrays positioned with the base vertical, 45 degrees, and horizontal while dissipating heat to surrounding fluid. Sparrow and Vemuri [15] did an experimental study on Natural Convection/ Radiation Heat Transfer from highly populated pin fin Arrays. Experiments were performed to determine the combined mode of natural convection/ radiation heat transfer characteristics of highly populated arrays of rod like cylindrical fins i.e., pin fins. They found that, if the number of fins were increased for fixed values of the other parameters, the heat transfer increased at first, attained a maximum and then decreased.

A survey of literature indicates that very few experimental and theoretical studies are available on inclined fin arrays. Hence, an experimental and theoretical investigation of the natural convection heat transfer from two geometric orientations viz., (a) vertical base with vertical fin arrays and (b) horizontal base with vertical fin array, with the five different inclinations are presented in this paper.

2. PHYSICAL MODEL AND FORMULATION

A horizontal fin array contains fins arranged on a horizontal plate, which is called the 'base of the fin array'. The considered fin arrays are consisting of seventeen, eight and four fins. In four fin array first and fourth fins are the end fins and the remaining are inner fins. However the end fins are exposed to an infinite ambient medium on one side.

The two adjacent inner fins having a common base with the fin spacing S, which is shown in Figure-1. The base is maintained at a constant temperature $T_{w,0}$ and $T_{w,0} > T_{\infty}$, where T_{∞} is ambient air temperature. The length and width of each fin are L and W respectively and the ratio (W/L) is by far greater than unity. t_f is the half thickness of the fin. Heat is transferred from the base to the fins by conduction and from the fins to the ambient air by convection.

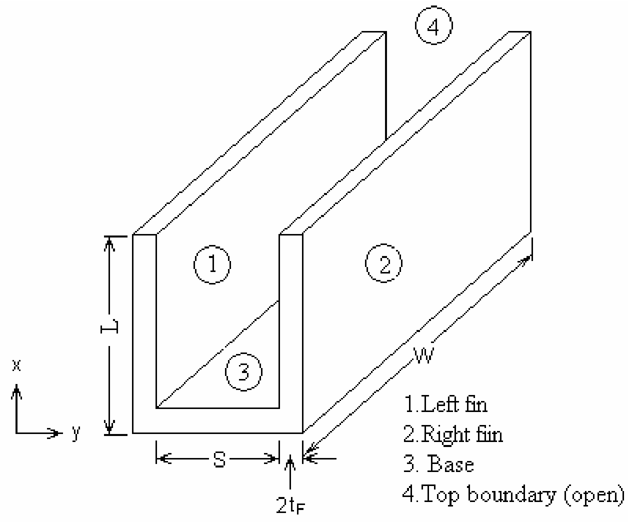

Figure-1. Physical model of two fin enclosure.

The problem is formulated considering the two vertical fins and the horizontal base together as a two-dimensional enclosure [4]. The two fins and the base are the left, right and bottom boundaries of the enclosure respectively. The top, which is open, is considered to be the fourth boundary. The velocity and temperature fields in the two-fin enclosure are governed by the mass, momentum and energy balance equations for the fluid in conjugation with the one-dimensional heat conduction equation for each fin, which are given below. Thus the equations considered are as follows.

Fluid medium:

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} = 0 \tag{1}$$

$$\rho_f \left(\frac{\partial u}{\partial t} + u\frac{\partial u}{\partial x} + v\frac{\partial u}{\partial y} \right) = - \frac{\partial p}{\partial x} - \rho_f \left[1-\beta(T-T_\infty)\right] g \cos\phi + \mu_f \left(\frac{\partial^2 u}{\partial x^2} + \frac{\partial^2 u}{\partial y^2} \right) \tag{2}$$

$$\rho_f \left(\frac{\partial v}{\partial t} + u\frac{\partial v}{\partial x} + v\frac{\partial v}{\partial y} \right) = - \frac{\partial p}{\partial y} + \mu_f \left(\frac{\partial^2 v}{\partial x^2} + \frac{\partial^2 v}{\partial y^2} \right) - \rho_f \left[1-\beta(T-T_\infty)\right] g \sin\phi \tag{3}$$

$$\rho_f C_{pf} \left(\frac{\partial T}{\partial t} + u\frac{\partial T}{\partial x} + v\frac{\partial T}{\partial y} \right) = k_f \left(\frac{\partial^2 T}{\partial x^2} + \frac{\partial^2 T}{\partial y^2} \right) \tag{4}$$

Heat conduction in left fin:

$$\rho_w C_{pw} \frac{\partial T_w}{\partial t} = k_w \frac{\partial^2 T_w}{\partial x^2} + \frac{P}{A_w} k_f \left.\frac{\partial T}{\partial y}\right|_{y=0} \tag{5}$$

Heat conduction in right fin:

$$\rho_w C_{pw} \frac{\partial T_w}{\partial t} = k_w \frac{\partial^2 T_w}{\partial x^2} - \frac{P}{A_w} k_f \left.\frac{\partial T}{\partial y}\right|_{y=S} \tag{6}$$

$$P = 2\,t_F + W \text{ and } A_w = t_F\, W \tag{7}$$

The u- and v- momentum balance equations, i.e., Eqs. (2) and (3) are coupled making use of vorticity ζ, which is defined as follows.

$$\zeta = \frac{\partial u}{\partial y} - \frac{\partial v}{\partial x} \tag{8}$$

The terms in Eq. (2) are differentiated with respect to y and those in Eq. (3) are differentiated with respect to x. The resulting equations are subtracted one from the other to yield the following vorticity equation.

$$\rho_f \left(\frac{\partial \zeta}{\partial t} + u\frac{\partial \zeta}{\partial x} + v\frac{\partial \zeta}{\partial y} \right) = \rho_f g \beta \left(\frac{\partial T}{\partial y} Cos\phi - \frac{\partial T}{\partial x} Sin\phi \right) + \mu_f \left(\frac{\partial^2 \zeta}{\partial x^2} + \frac{\partial^2 \zeta}{\partial y^2} \right) \tag{9}$$

The stream function ψ is defined as

$$u = \frac{\partial \psi}{\partial y}, \; v = - \frac{\partial \psi}{\partial x} \tag{10}$$

The vorticity equation in terms of stream function is as follows:

$$\zeta = \frac{\partial^2 \psi}{\partial x^2} + \frac{\partial^2 \psi}{\partial y^2} \tag{11}$$

The stream functions ψ can be evaluated using Eq. (11) if the vorticities ζ are known. The velocity components u and v are to be computed from the values of ψ through equation (10).

3. BOUNDARY CONDITIONS

The boundary conditions at the left, right, bottom and top boundaries of the enclosure are given below.

Left boundary (y = 0):

$T = T_{w,0}$ = constant at x = 0 (fin base);

$u = v = \psi = 0$ for $0 \le x \le L$

$\frac{\partial T}{\partial x} = 0$ at x = L (fin tip) ;

$$\left.\zeta\right|_{y=0} = \frac{2\,\psi_{y=\Delta y}}{(\Delta y)^2} \quad \text{for } 0 \le x \le L \text{ (left fin)} \tag{12}$$

Right boundary (y = S):

$T = T_{w,0}$ = constant at x = 0 (fin base)

$\frac{\partial T}{\partial x} = 0$ at x =L (fin tip); $u = v = \psi = 0$ for $0 \le x \le L$

$$\zeta\big|_{y=S} = \frac{2\psi_{y=S-\Delta y}}{(\Delta y)^2} \text{ for } 0 \le x \le L \text{ (right fin)} \tag{13}$$

Base or bottom boundary (x = 0):

$u = v = \psi = 0$, and $T = T_{w,0}$ =constant for $0 \le y \le S$

$$\zeta\big|_{x=0} = \frac{2\psi_{x=\Delta x}}{(\Delta x)^2} \text{ for } 0 \le y \le S \tag{14}$$

Top boundary at x = L:

The top boundary is open, or no boundary. Roache [11] described it as an "open flight case" and suggested the following boundary conditions.

$$v = \frac{\partial u}{\partial x} = \frac{\partial \psi}{\partial x} = 0 \text{ and } \frac{\partial T}{\partial x} = 0;$$

$$\frac{\partial \zeta}{\partial x} = 0 \quad \text{at } x = L \text{ for } 0 \le y \le S \text{ (top)} \tag{15}$$

All the equations are normalized by making use of the following variables.

$$x^+ = \frac{x}{S}, y^+ = \frac{y}{S}Gr^{1/4}, \; t^+ = \frac{\nu_f t}{S^2}Gr^{1/2},$$

$$u^+ = \frac{Su}{\nu_f}\frac{1}{Gr^{1/2}}, v^+ = \frac{Sv}{\nu_f}\frac{1}{Gr^{1/4}}, \; T^+ = \frac{T - T_\infty}{T_{w,0} - T_\infty},$$

$$\psi^+ = \frac{\psi}{\nu_f}\frac{1}{Gr^{1/4}}, \zeta^+ = \frac{S^2\zeta}{\nu_f}\frac{1}{Gr^{3/4}}, \; \alpha^+ = \frac{\alpha_w}{\alpha_f},$$

$$T_w^+ = \frac{T_w - T_\infty}{T_{w,0} - T_\infty}, q_{ri}^+ = \frac{q_{ri}\, S}{k_f\left(T_{w,0} - T_\infty\right)}\frac{1}{Gr^{1/4}},$$

The dimensionless system parameters are namely M, the conduction-convection ratio parameter, A_R, aspect ratio and γ, temperature ratio parameter.

$$M = \frac{k_f PS}{k_w A_w} Gr^{1/4} \quad A_R = \frac{L}{S}, \; \gamma = \frac{T_{w,0} - T_\infty}{T_\infty}$$

The energy balance and vorticity equations for the fluid, heat conduction equations for the left and right fins and the boundary conditions for the inner as well as end fins are written in normalized form, but are not shown here to conserve space.

The method of solution and accuracy and convergence criteria are explained in Dharma Rao *et al*., [4]

4. HEAT TRANSFER RATE FROM A FIN ARRAY

The heat fluxes in dimensionless notation from the left fin and base are given by the following equations.

$$q_1 = \int_0^L \left[-\frac{\partial T}{\partial y}\bigg|_{y=0} \right] dx \,;\; Q1 = (L\,W)\, q_1 \tag{16}$$

$$q_3 = -\frac{1}{Gr^{1/4}}\frac{\partial T}{\partial x}\bigg|_{x=0} \;;\quad Q_3 = (S\,W)\, q_3 \tag{17}$$

$$q_5 = \int_0 \left[-\frac{\partial T}{\partial y}\bigg|_{y=0} \right] dx \,;\; Q5 = (L\,W)\, q_5 \tag{18}$$

q_1 and q_3 are heat fluxes from the fin and the base respectively. Q_1 and Q_3 are the heat flow rates from the fin and the base taking into consideration the respective surface areas. Q_5 refers to the heat flow rate from the outer face of the end fin (either first or the last in the array) of half-thickness t_F. It may be noted that Q_1 refers to the heat flow rate from one face of the fin only. Further the heat transfer rates from the left and right fins are assumed to be equal.

The heat transfer rate from the fin array (Q_T) is given by

$$Q_T = (N-1)\,(2\,Q_1 + Q_3) + 2\,Q_5 \tag{19}$$

4.1 AVERAGE NUSSELT FROM THE FIN ARRAY

Total heat transfer area for the N-fin array is given by

$$A_T = (B - 2\,N\,t_F)\,W + 2\,N\,L\,W$$

$$B = (N - 1)\,S + 2\,N\,t_F \tag{20}$$

Where B is the breadth of the plate and N is the number of fins. The average heat transfer coefficient, h_m for the N-fin array for combined convection is defined by the following equation

$$h_m = \frac{Q_T}{A_T\left(T_{w,0} - T_\infty\right)} \tag{21}$$

The average Nusselt number (Nu_m) for the N-fin array is defined as

$$Nu_m = h_m\, S / k_f \tag{22}$$

5. EXPERIMENT PROCEDURE

A photographic view of the experimental setup is shown in Figure-2.

Figure-2. Photographic view of an experimental setup.

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An experimental investigation of the natural convection heat transfer from fin arrays is carried out for two different orientations a) Horizontal base with vertical fin arrays and b) Vertical base with vertical fin arrays with the five different inclinations like 0^0, 30^0, 45^0, 60^0, and 90^0. Temperatures are measured at different positions on the inner fin of the array during the experiments. The heat transfer coefficients for the fin array are calculated by making use of the measured temperatures by using the calculation procedure of Sujatha and Sobhan [17]. Experimental heat transfer rates are obtained for different number of fins present in the fin array. Four different fin arrays are fabricated containing 4, 8 and 17 fins respectively. Two of the fin arrays are having 20 mm long fins and another two fin arrays are having 40 mm long fins with the breadth of 153.5 mm base plate and the width is 100mm for all the four fin arrays.
The range of parameters is listed in Table-1.

Table-1. Range of parameters.

Fin length (L), mm	20 and 40
Fin thickness ($2t_F$), mm	3
Fin array length (B), mm	153.5
Fin spacing (S), mm	47,19 and 7
Number of fins (N)	4,8, and 17
Fin width (W), mm	100
Ambient temperature T_∞	28 ^{0}C
Base temperature ranges($T_{W,0}$), ^{0}C	40 -130
Inclinations	0^0, 30^0, 45^0, 60^0,90^0

6. RESULTS AND DISCUSIONS

The numerical results are obtained for vertical fin array of having 14 fins with the parameters L=25.4mm, W =254mm, S =7.95mm, t_F =1.016mm of 45^0 inclination by varying base temperature from 90 to 200^0C. The obtained numerical results are compared with the experimental data of Starner and McManus [16] in Figure-3.

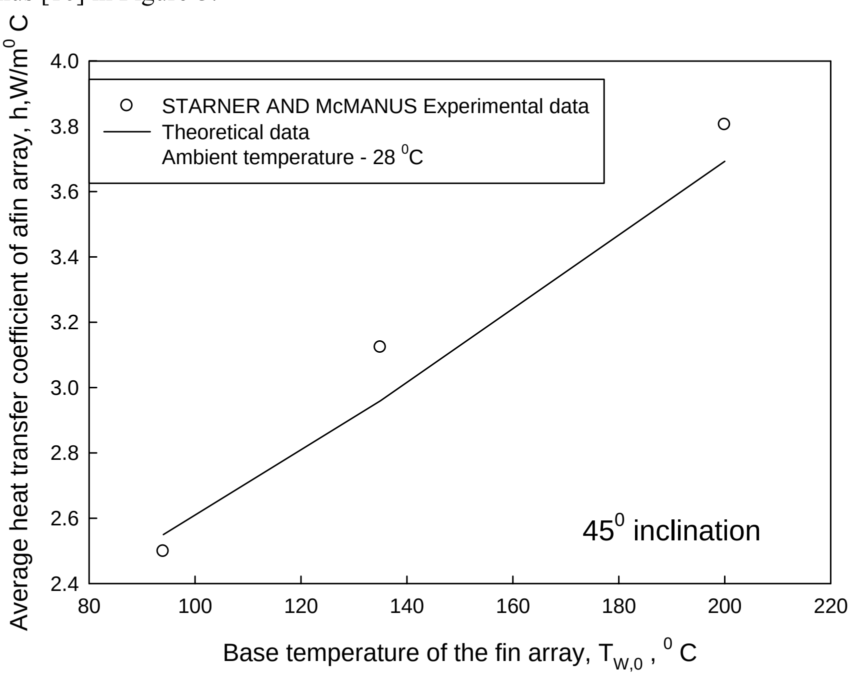

Figure-3. Comparison of present theoretical data with the experimental data available in literature (Starner and Mcmanus, 1963).

The numerical results show good agreement with the experimental data of Starner and McManus [16].

Numerical results are obtained for horizontal fin array for 20mm and 40mm lengths, fin spacing 7,10.7,19 and 47mm having B = 153.5 mm and W = 100mm. Nusselt numbers are calculated with respect to Rayleigh number and the variation of average Nusselt number with Rayleigh number shown in Figure-4.

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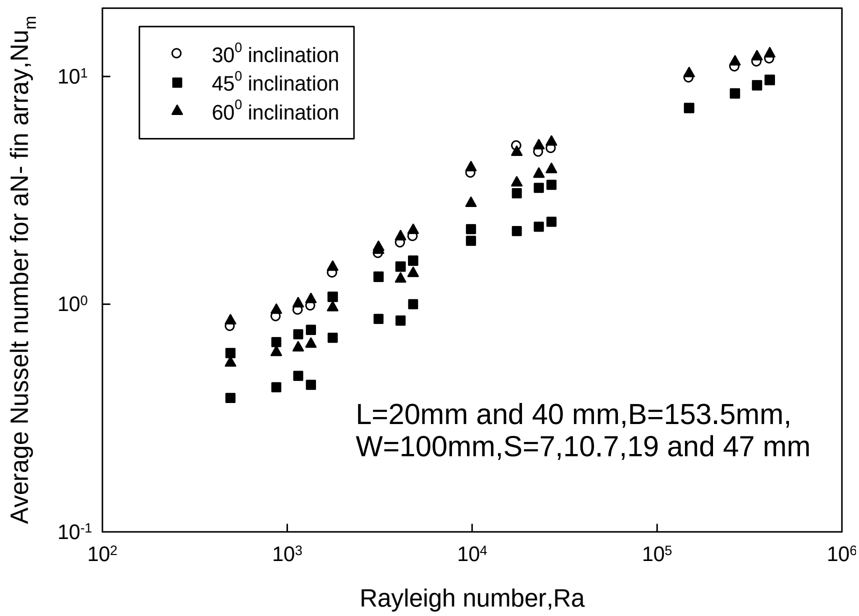

Figure-4. Variation of average Nusselt number with Rayleigh number.

The experimental results are obtained for a fin array of horizontal base with vertical fins with the parameters N=17, W=100mm, L=20mm, S=7 mm, B=153.5 mm and N=8, W=100mm, L=40mm, S=19 mm, B= 153.5 mm. The convection heat transfer rates are calculated from the experimental data and shown the effect of inclination in Figures 5 and 6, respectively.

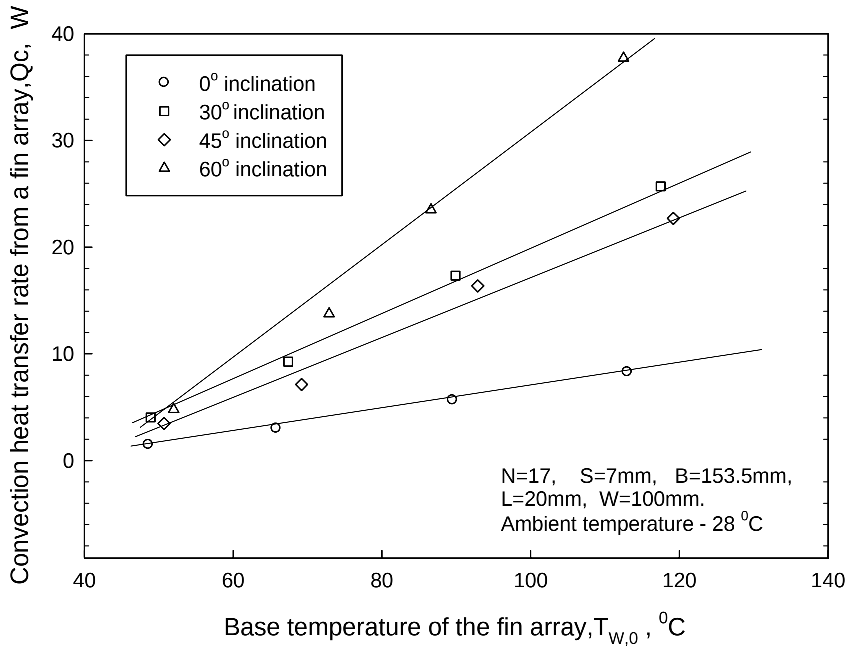

Figure-5. Effect of inclination on convection heat transfer rate from a fin array.

It is observed from Figures 5 and 6 that the convection heat transfer rates are increasing from 0^0 to30^0, decreasing from 30^0 to $45^0$and again increasing from 45^0 to 60^0 and 60^0 to 90^0 inclination for orientation of horizontal base with vertical fins.

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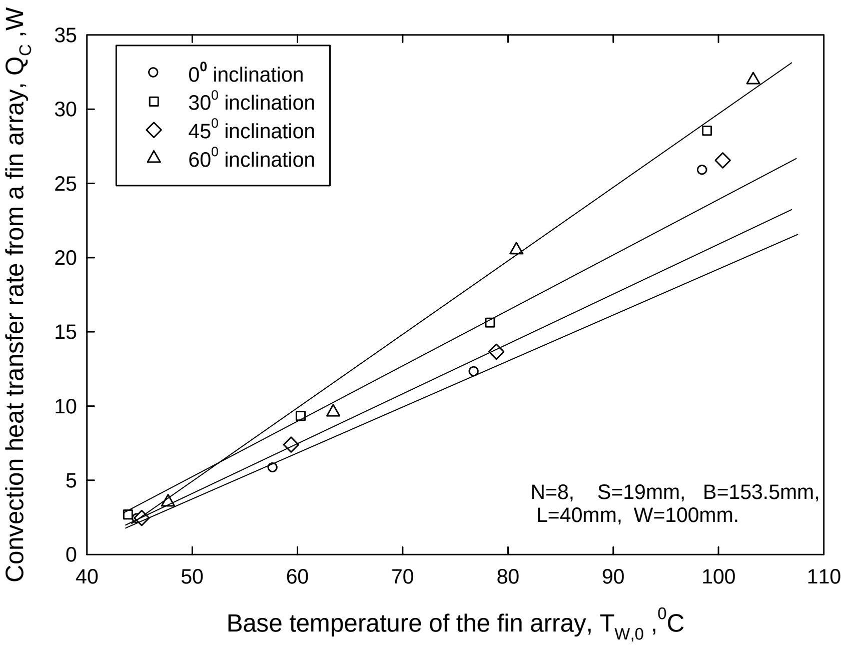

Figure-6. Effect of inclination on convection heat transfer rate from a fin array.

The experimental results are also obtained for fin array of vertical base and vertical fins with the parameters N=17, W=100mm, L=20mm, S=7 mm, B= 153.5 mm and N=8, W=100mm, L=20mm, S=19 mm, B= 153.5 mm. The convection heat transfer rates are calculated from the experimental data and shown the effect of inclination in Figure-7 and in Figure-8 respectively.

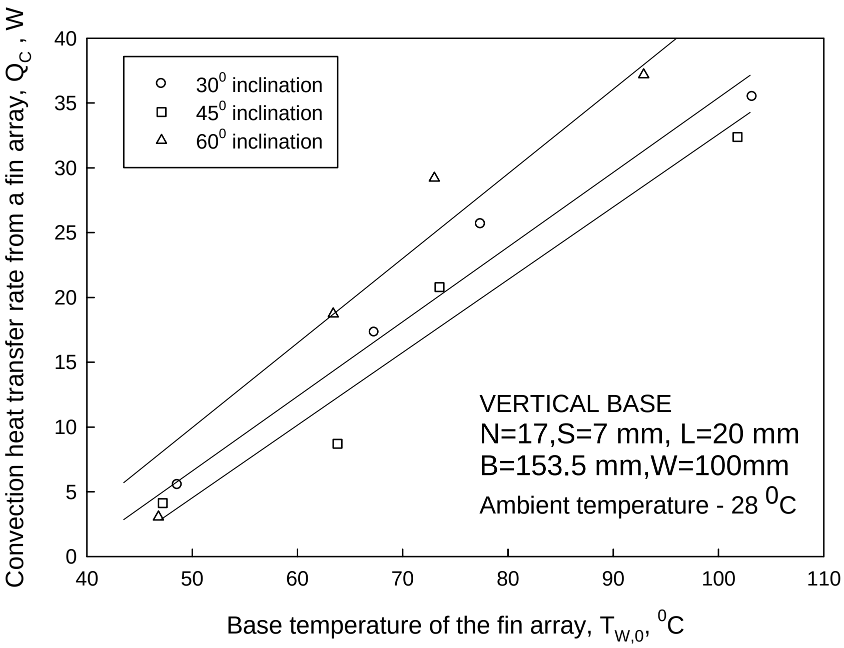

Figure-7. Effect of inclination on convection heat transfer rate from a fin array.

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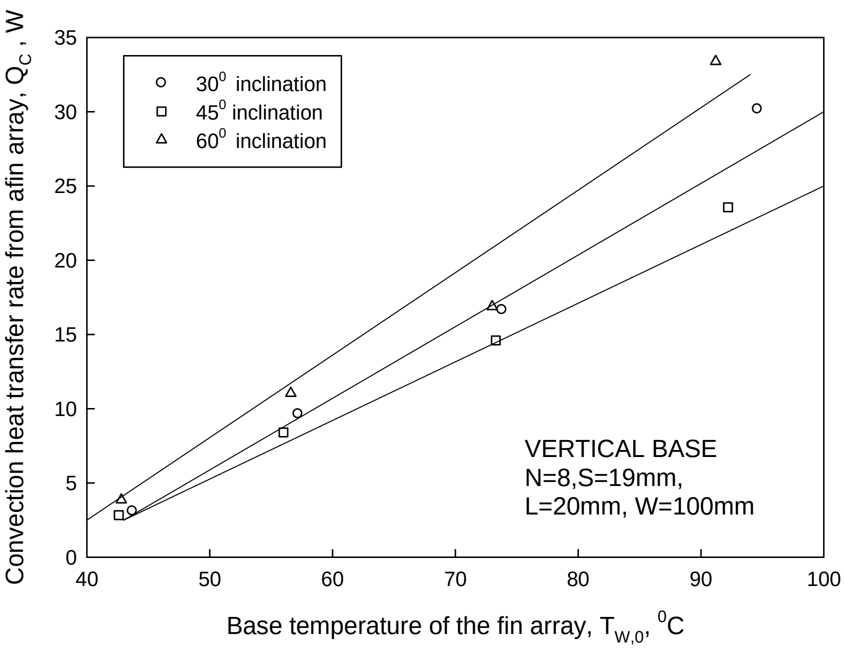

Figure-8. Effect of inclination on convection heat transfer rate from a fin array.

It is observed from Figures 7 and 8 that the convection heat transfer rates are increasing from 0^0 to 30^0, decreasing from 30^0 to $45^0$and again increasing from 45^0 to 60^0 and 60^0 to 90^0 inclination for orientation of vertical base with vertical fins.

Experimental results are compared with our numerical results for the fin array having 17 fins shown in Figure-9.

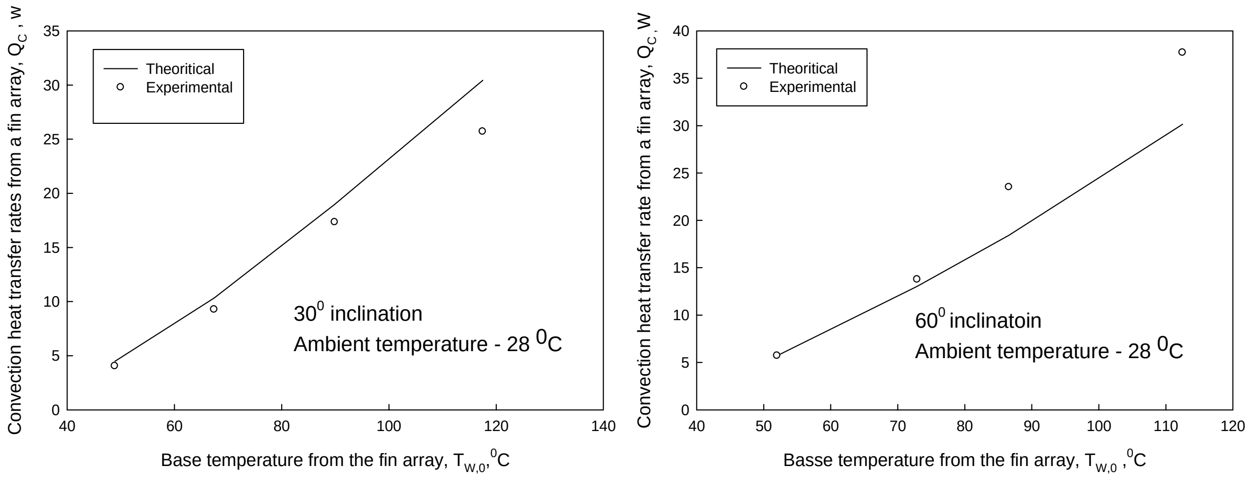

Figure-9. Comparison of experimental data with theoretical results for L=20 mm, S=7 mm, N =17, B =153.5 mm.

It shows good agreement between experimental and numerical results. The heat transfer rates are calculated from the experimental data for two different orientation of fin array is compared in Figure-10 for the inclinations of 30^0 and 60^0.

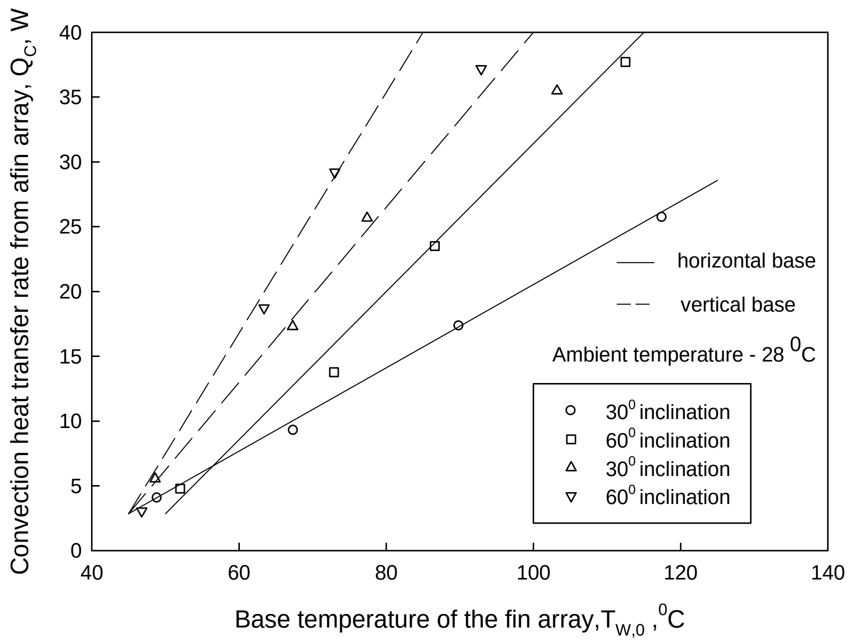

Figure-10. Comparison of experimental data for two different positions of the fin array.

It shows that the convection heat transfer rates are less for the position of horizontal base and vertical fins with length basis than horizontal base and vertical fins with width basis for the same inclination of the fin array.

7. CONCLUSIONS

A theoretical model is formulated to tackle the problem of heat transfer from a fin array with inclination. According to the model, the fin array is assumed to be formed by joining successive two fin enclosures. The problem for the case of a two fin enclosure is theoretically formulated and solved considering heat transfer by natural convection.

Theoretical studies have been carried out on natural convection heat transfer for Horizontal base with vertical fins with five different inclinations like 0^0, 30^0, 45^0, 60^0, and 90^0 Experimental data are obtained for natural convection heat transfer from a fin array by placing it in two different orientations a) Horizontal base with vertical fins and b) Vertical base with vertical fins with the five different inclinations like 0^0, 30^0, 45^0,60^0. A comparison with the experimental data existing in literature indicates satisfactory agreement.

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Nomenclature

A_B	area of the horizontal base plate, (B W),
A_R	aspect ratio for two-fin enclosure, (L/S)
A_S	Surface area of the fin, 2(LW), m^2
A_w	half the cross-sectional area of the fin, (t_F W), m^2
B	breadth of the horizontal base plate, m
C_p	specific heat, J kg^{-1} K^{-1}
g	acceleration due to gravity, m s^{-2}
Gr	Grashoff number, $g\beta\left(T_{w,0}-T_\infty\right)S^3/\nu_f^2$
h	heat transfer coefficient, W m^{-2} K^{-1}
k	thermal conductivity, W m^{-1} K^{-1}
L	length of the fin, m
N	number of fins in the fin array
Nu	Nusselt number
p	pressure, Pascals
P	half-perimeter of the fin, (2 t_F + W), m
Pr	Prandtl number, $c_{pf}\,\mu_f$ / k_f
q	heat flux, W m^{-2}
q^+	normalized heat flux, q S / [k_f ($T_{w,0}$ – T_∞) $Gr^{1/4}$
Q	heat transfer rate, W
Ra	Rayleigh number, g β ($T_{w,0}$ - T_∞) S^3 / ($\nu_f\,\alpha_w$)
S	spacing between adjacent fins, m
t_F	half-thickness of the fin, m
t	time, s
T	temperature, K
u	velocity component in x-direction, m s^{-1}
v	velocity component in y-direction, m s^{-1}
W	width of the fin (also that of base plate),
x	position coordinate along the fin measured from the base of the fin, m
y	position coordinate normal to the fin measured from the left fin

Greek symbols

ϕ	Inclination of the fin array
α	thermal diffusivity of fluid, $(k/\rho c)_f$,m^2 /s
α^+	α_w/α_f
β	isobaric coefficient or thermal expansion coefficient of fluid, K^{-1}.
ΔT	$T_{w,0}$ - T_∞
γ	temperature ratio parameter, $(T_{w,0}-T_\infty)/T_\infty$
μ	dynamic viscosity, kg m^{-1} s^{-1}
ν	kinematic viscosity, m^2 s^{-1}
ρ	density, kg m^{-3}.
ψ	stream function.
σ	Stefan-Boltzman constant, (5.67 x 10^{-8} W m^{-2} K^{-4})
θ	T - T_∞
θ_w	T_w/T_∞
ζ	vorticity function, s^{-1}

Subscripts

1	fin in a two-fin enclosure
3	base in a two-fin enclosure
5	end fins
B	base of the fin array
B,0	base plate in the absence of fins
E	end fins of the fin array
I	internal fins of the fin array
c	convection
f	fluid.
L	fin tip
m	average
M	middle of the fin
T	total
w	fin surface
w,0	base of the fin
∞	ambient medium