THE INFLUENCE OF ORIENTATION ON THE PERFORMANCE OF PIN FINNED HEATED PLATE SUBJECTED TO NATURAL CONVECTION +

تأثير اتجاه الزعانف على أداء صفيحة مسخنة ذات زعانف وتدية بالحمل الحر

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Abstract:

The objective of this paper is to perform an experimental study of free convection heat transfer from staggered pin fin-arrays on heated base plate. The experiments represented by studying three orientations of the fins (upward, downward and sideward). The results showed that the sideward orientation gives better results as compared with other orientations. Nusselt number determined as a function of Rayleigh number at Prandtl number approximately equal to (0.7) for all tests, Rayleigh number in this study ranges between ($1.877\times10^{3} \leq Ra \leq 8.086\times10^{3}$) . From the results the heat transfer coefficients of the heated plate in sideward orientation are about (8%) greater than those for the upward orientation and exceeds that of downward facing by (24%). Also the orientation effected in the distribution of temperature along the fins. The results shows that the temperatures along the fins in sideward state have an uniform distribution, while in upward orientation the temperature along the fins increased in location nears the base plate, this behavior becomes very clear in downward orientation because of the impeding in moving hot air.

Keywords : Natural Convection , Pin Finned , Orientation .

المستخلص:

الهدف من هذا البحث أجراء دراسة عملية لانتقال الحرارة بالحمل الحر من صفيحة مسخنة ذات صفوف من الزعانف الوتدية.تمثلت التجارب بدراسة ثلاث أوضاع للزعانف (للأعلى،للأسفل وللجانب).النتائج بينت أنه حالة الوضع الجانبي للزعانف تعطي نتائج أفضل بالمقارنة مع الأوضاع الأخرى.عدد نسلت (Nusselt number) تم تحديده كدالة لرقم رالي وعند عدد برانديتل(Prandtl number) يساوي (0.7) لجميع التجارب،وقد تراوح عدد

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رالي (Rayleigh Number) في هذه الدراسة بين ($1.877\times10^3 \le Ra \le 8.086\times10^3$).من النتائج معامل انتقال الحرارة بالحمل الحر في الحالة الجانبية أعلى بحدود(8%) منه في حالة الوضع الأفقي وفاق الحالة الجانبية بـــ (24%).كذلك وضع الزعانف اثر على توزيع درجات الحرارة على طول الزعانف. النتائج أظهرت أن درجات الحرارة على طول الزعانف في الحالة الجانبية لها توزيع منتظم،بينما في حالة الوضع للأعلى ازدادت درجات الحرارة قـــرب سطح الصحيفة وهذا السلوك أصبح واضحا جدا في حالة الوضع للأسفل بسبب الإعاقة في حركة الهواء الساخن .

Nomenclature:

A	Cross-sectional area of fin , (m^2)
A_b	Base area not occupied by fin, (m^2)
A_t	Total area, (m^2)
a	Width of heated base plate, (m)
b	Length of heated base plate, (m)
C_p	Specific heat, (J/kg.^{o}C)
d	Fin diameter, (m)
g	Acceleration due to gravity, (m^2/s)
Gr	Grashof number
h	Convective heat transfer coefficient, (W/m^2.^{o}C)
k	Fin thermal conductivity, (W/m.^{o}C)
k_f	Fluid thermal conductivity, (W/m.^{o}C)
L	Length of fin, (m)
m	Fin parameter, (m^{-1})
n	Number of fins
Nu	Nusselt number
Pr	Prandtl number
P	Periphery of fin, (m)
Q_b	Heat transfer from base unoccupied by fin, (W)
Q_c	Heat transfer by convection, (W)
Q_f	Heat transfer from single fin, (W)
Q_r	Heat transfer by radiation, (W)
Q_t	Total power supply, (W)
R_a	Rayleigh number
T_f	Film temperature, (^{o}C)
T_b	Base temperature, (^{o}C)
T_x	Temperature at position (x) , (^{o}C)
T_∞	Ambient temperature, (^{o}C)
x	Distance from fin base, (m)
η	Fin efficiency
β	Coefficient of thermal expansion (K^{-1})
υ	Kinematic viscosity, (m^2/s)
ε	Emissivity
σ	Stefan-Boltzmann constant (W/m^2.K^4)
θ	Non-dimensional temperature

Introduction:

Direct air-cooling is still the most popular means for cooling electronic products. Recognizing the poor heat transfer performance of air-cooling, secondary fin surface is usually exploited to solve the increased demand of heat dissipating. For further improvement of the overall heat transfer performance, augmentation via fin pattern is often employed. Among the fin configurations being implemented in practical applications, pin fin is regarded as one of the most effective heat transfer augmentation methods for providing boundary layer restarting and periodic vortex shedding. Notice that the pin fin is considered as the effective internal cooling means in the turbine blade, and the cooling of circuit boards. Extensive research for short pin arrays had been reported in the literature both numerically and experimentally. The effects of length to diameter, array geometry, entrance length, and fin shape had been reported.

Note that the most common pin fin shape takes the form as cylindrical for its superior heat transfer performance[1]. The aim of this investigation is to determine the heat transfer coefficient at constant heat flux for pin fins . The heat generated in the experiments is on the back of the base plate holding the fins . In all experiments , the area of base plate is the same . The pin fins are manufactured from aluminum materials as one piece of the base plates. Many experiments were performed in the literature in pin types of fins under investigation.

Saad M.J. Al-Azawi [2] performed experiments to study the effect of various orientations on natural convection heat transfer from longitudinal trapezoidal fins array heat sink. Test results indicated that the sideward horizontal fin orientation yield the lowest heat transfer coefficient . From results heat transfer coefficient of the sideward vertical fins was higher by 12% than the heat transfer coefficient of the upward while it was higher than the heat transfer coefficient of the downward by 26% and by 120% with sideward horizontal fins.
Ren-Tsung Huang et al [3] conducted an experimental study on natural convection heat transfer from square pin fin heat sinks subjected to the influence of orientation .A total of six pin fin heat sinks with various arrangements tested under controlled environment. Test results showed that the upward facing orientation yields the highest heat transfer coefficient, followed by the sideward facing and the downward facing ones. The heat transfer coefficients for upward facing were about (0% - 5%) greater than those for sideward facing and were (0% - 20%) greater than those for downward facing.

Wirts et al. [4] reported an experimental results on the thermal performance of model pin-fin fan-sink assemblies. They used cylindrical, square, and diamond-shaped cross-sectional pin-fins and found that cylindrical pin-fins gave the best overall fan-sink performance. Furthermore, the overall heat-sink thermal resistance decreases with an increase in either pressure rise or fan power and fin height.

Jonsson and Bjorn [5] performed experiments to compare the thermal performance of heat sinks with different fin designs including straight fins and pin fins with circular, quadratic, and elliptical cross sections. They evaluated the thermal performance by comparing the thermal resistance of the heat sinks at equal average velocity and equal pressure drop. They recommended elliptical pin-fin heat sinks at high velocities and circular pin-fin heat sinks at midrange velocities.

Kobus and Oshio [6] investigated the effect of thermal radiation on the heat transfer of pin fin heat sinks and presented an overall heat transfer coefficient that was the sum of an effective radiation and a convective heat transfer coefficient.Aihara et al. [7] performed a detailed experimental investigation of natural convection and radiation heat transfer from pin–fin arrays with a vertical base plate. Totally, 59 types of circular pin–fin dissipaters were used. They employed a symmetrical geometry about a common base to minimize the heat loss from the base plate. Temperature measurements and flow visualizations were conducted and an empirical expression for the average heat transfer coefficient was derived.

Fisher and Torrance [8] presented an analytical solution for a system consisting of a pin-fin heat sink and a chimney . The result is applied to problems in which the size of the overall system was constrained. For a given heat dissipation and total system size, optimal values of the pin-fin diameter and heat-sink porosity were observed. The optima occurred for systems with and without chimneys. The optimization was used to show that the minimum thermal resistance from a pin-fin heat sink was about two times larger than that of an idealized model based on inviscid flow.
Park et al. [9] performed numerically the optimization of the plate heat exchanger with staggered pin arrays for a fixed volume. The flow and thermal fields were assumed to be periodic fully developed flow and heat transfer with constant wall temperature and they were solved using the finite-volume method. The optimization is carried out by using the sequential linear programming (SLP) method and the weighting method is adopted for solving the multiobjective problem. The result showed that the optimal design variables for the weighting coefficient of 0.5 were represented in Diameters equal to (0.689mm & 2.396mm).
Babus'Haq et al. [10] investigated experimentally the thermal performance of a shrouded vertical Duralumin pin-fin assembly in in-line and staggered configuration. They found that under similar flow conditions and for an equal number of pin-fins, the staggered configuration yields a higher steady-state rate of heat transfer than the in-line configuration. They studied

the effect of changing the thermal conductivity of the pin-fin material and found that the optimal separation between the pin-fins in the stream wise direction increased with the thermal conductivity of the pin-fin material, whereas the optimal separations in the span wise direction remained invariant.
Many papers carried out to study the natural convection heat transfer from round pin fin arrays, but at least no one mainly specialized an experimental study of the performance of pin fins heated plated in the presence of the orientation effect. However investigations with arrays of pin fins on flat heated surface are rather limited. Hence the present study was undertaken. The prime objective of this study is to give an experimental data for pin finned heated plate under natural convection with different orientation and to present clearly its performance.

<u>Experimental apparatus:</u>

In order to experimentally measure the thermal performance of the pin finned heated plate, it is essential that the rate of heat transfer between the heated plate and the flowing air is accurately measured . In this study, electrical patch heaters were used to heat the base plate of pin fins . It would be highly desirable to be able to indirectly obtain an accurate measurement of the rate of heat transfer from the heated plate to the environment , by simply measuring the electrical power input to the heater . In order to verify the reliability of this measurement technique. Fig.(1) shows a schematic of the experimental apparatus. The experiments done on an aluminum pin finned heated plate with dimensions of (0.1m×0.11m).The fins are manufactured from polished aluminum as one part of the base plate.

Fig.(2) shows the dimensions of the pin finned heated plate and its orientation takes for study. The base plate has seventeen fins of (0.012 m) diameter and (0.066m) length. Tests carried out on natural convection heat transfer from pin finned heated plate placed in different orientations (Sideward face, Downward face, and Upward face). To fix the sample and move it in three orientations with flexible manner an iron frame was constructed for this purpose. The base plate is equipped with an electrical heating element placed at the back side of the base plate between two mica sheets providing electrical insulation. An insulation box with low thermal conductivity is placed beneath the heater to reduce the heat loss. In addition, a high thermal conductivity grease is used to connect the heat sink and the heater. The base plate temperature is measured by five thermocouples type (K), placed at a symmetric location in the base plate.

An electrical unit was used to control and indicate the power supplied to the heating heater. This unit contains (Ammeter, Voltmeter, and Regavolt transformer),all thermocouples of the base plate and those attached along the fin were connected to a selector switch type (K) and then to digital thermometer. All experiments were performed inside a chamber represent a nylon box of dimensions (0.5m × 0.5m) and (0.7m) height, used to observe the electrical apparatus and have a stable medium around the heat sink. More than two hours were required to reach the steady state condition. The heated plate is tested at ambient temperature (25^{o}C) with constant heat flux condition ranging from (5W- 40W).

<u>Calculation procedure:</u>

To calculate all needed parameters of study from the experimental data, the ambient temperature in all tests was controlled at 25^{o}C, and thermophysical properties in Nusselt and Rayleigh numbers are evaluated at the film temperature (T_f) as shown below [11]:

$$T_f = \frac{1}{2}(T_\infty + T_b) \qquad (1)$$

Also the temperature for each location along the fins will calculate as non-dimensional parameter as bellow,

$$\theta = \frac{T-T_\infty}{T_b-T_\infty} \qquad (2)$$

The well- known one-dimensional differential equation governing the temperature distribution for such a fin covered in most introductory heat transfer text-books, is given by,

$$\frac{d^2T}{dx^2} - \left(\frac{hP}{kA}\right)(T - T_\infty) = 0 \qquad (3)$$

Solving the resulting differential equation, subject to appropriate boundary conditions, [7] yields the axial temperature distribution in the representative fin. Using this temperature distribution, along with Fourier's law of conduction, the rate of heat transfer from a single pin fin, Q_f, can be modeled as [11]

$$Q_f = kmA(T_b - T_\infty)\tanh(mL) \qquad (4)$$

Where $, m=\left(\frac{hP}{kA}\right)^{1/2}$ and $P = \pi d$, $A=\frac{\pi d^2}{4}$

Also, in terms of the convective heat transfer coefficient, the rate of heat transfer from that part of the heat sink base not occupied by fins, Q_b can be expressed as

$$Q_b = hA_b(T_b - T_\infty) \qquad (5)$$

Where ; $A_b = (ab - nA)$, where n is a fins number and equal to (17) fins.

Therefore the total rate of heat transfer from the heated plate, Q_t ,which contain n-fins , can be expressed as

$$Q_t = nQ_f + Q_b \qquad (6)$$

and neglecting conduction downward losses then the energy balance for the heat sink is

$$Q_c = Q_t - Q_r \qquad (7)$$

$$Q_r = \varepsilon.\sigma.A_t(T_b^4 - T_\infty^4) \qquad (8)$$

The convective heat transfer coefficient can be expressed as[11]

$$h=\frac{Q_c}{(nPL\eta+(ab-nA))(T_b-T_\infty)} \qquad (9)$$

where η is the fin efficiency and defined as ,[11]

$$\eta=\frac{\tanh(mL)}{mL} \qquad (10)$$

It is clear that the heat transfer coefficient appears as a part of fin parameter (m) that is also found in equation (10),therefore the iteration method will be used to solve equation (9) for each reading .The Nusselt and Rayleigh numbers are calculated as

$$Nu=\frac{hd}{k_f} \qquad (11)$$

Where k_f is the conduction heat transfer coefficient for air at T_f, and the diameter of pins takes as the characteristic length[11].

$$Gr=\frac{g\beta(T_b-T_\infty)d^3}{\nu^2} \qquad (12)$$

$\beta=\frac{1}{T_f}$ Where

$$Ra = Gr.Pr \qquad (13)$$

The relation between Nusselt and Rayleigh numbers for each orientation will be calculated as following,

$$Nu = cRa^n \qquad (14)$$

Where (c & n) are constants.

Results and discussion:

The three orientations of pin finned heated plate are examined experimentally to perform the effects of orientation on natural convection heat transfer from the finned heated plate, also its effects on temperatures distribution along the fins. All experiments were conducted at steady state conditions. Moreover, the present work has also verified via a comparison with experimental work lead by Ren-Tsung Huang [3],as shown in fig.(3).The sideward facing for both studies are taken in comparison. From the figure, it can be seen nearly similarity in behavior, but the present study gives a heat transfer coefficient greater than that done in [3].

From experimental data Nu was determined and plotted as function of Ra in fig. (4) for all orientations.This figure shows the effect of orientation on heat transfer performance. Generally the sideward facing orientation yields the highest heat transfer coefficient, followed by the upward facing and the downward facing ones . However, the heat transfer coefficient for sideward and upward arrangements are of compared magnitude. Strictly, the heat transfer coefficients for sideward facing are about (7%) greater than those for sideward facing and exceeds the downward facing by 24% .Table (1) shows the values of the constants (c & n) those appears in equation (14) for all orientations.

Table (1) Values of (c & n) and Ra ranges

Orientation	c	n	Ranges of Ra
Downward	0.2356	0.5368	$1.910\times10^3 \leq Ra \leq 7.892\times10^3$
Upward	0.2333	0.5609	$1.912\times10^3 \leq Ra \leq 8.086\times10^3$
Sideward	0.3350	0.5216	$1.877\times10^3 \leq Ra \leq 7.311\times10^3$

Fig.(5) shows the influence of orientation on temperature difference between the base plate and surrounding with total heat input to the heated plate .From this figure, it can be seen that, at same orientation the temperature difference increase with increasing heat flux. For the sideward facing the temperature difference has the lowest value when comparing with others. It is clear that the downward facing has higher temperature difference at same value of heat flux. In upward facing the temperature difference increased by 16% relevant to sideward, while in the downward facing it is increased by 22%, as shown in table (2).These increases in temperature difference for downward facing because of the complicating moving of hot air and staying more time in touch with plate surface, leading to rise the base plate temperature.
To explain the effect of orientation on temperature distribution along the fins, the non-dimensional temperature for each measured pointes in x-direction is plotted as a function of total heat supplied. These plots are presented in figs.(6) through (8).Each figure contains

results for all orientations studies (upward, downward, and sideward) in the same fin position. From the figures, it can be seen that, the non-dimensional temperature (θ) increases with the heat flux and decreases along the fins at constant heat flux for all orientations. Table (2) gives a good indicate of the percentage of increasing or decreasing in non-dimensional temperatures for the upward and downward facing relevant to the sideward facing in different positions along the fins.

Table (2) Percentages of changes in temperatures

(↑ = Increases , ↓ = Decreases)

State	(T_b-T_α) $_{x=0mm}$	θ $_{x=8mm}$	θ $_{x=35mm}$	θ $_{x=60mm}$
Sideward/Upward	16.0%↑	5.6%↑	6.5% ↑	6.2% ↑
Sideward/Downward	22.0%↑	4.6%↑	7.8% ↓	15.4% ↓

From the results, the non-dimensional temperature for the upward facing in comparing with sideward increased by 5.6% ,while the downward increased by 4.6% relative to sideward facing at (x=8mm),this change appears clearly in fig.(6).That occurs as a result of orientation effects in temperature distribution along the fins. The last two figures (7&8) shows the variation of non-dimensional temperatures at (x=35mm & x=60mm),it is clear that the fins in downward facing at these locations became cooler than the sideward facing, while in upward facing the non-dimensional values stays higher than the sideward and downward. It (x=60mm) the non-dimensional temperature of downward facing decreased by 15.4% than sideward facing. The percentage of variation is explained in table (2).

As a result of cooling manner by natural convection this non-uniform in temperature distribution is occurred for the downward facing especially. Air as the working fluid became hotter and raised up because of the density decreases, hitting the plate surface causing increasing in base plate temperature and the positions of the fins near the surface with complicating and impeding in moving it. In sideward and upward orientations the circulation of the hot air with cold air (ambient air) will not meet this complication. Now from the results the sideward orientation gives the best behavior in decreasing the base plate temperatures with uniform temperatures distribution along the fins.

Conclusions:

In this study natural convection heat transfer from pin finned heated plate at constant heat flux in three orientations (upward, downward, and sideward) with Rayleigh number values from (1.877×10^3) to (8.086×10^3) has been investigated. From the experimental results, it can be concluded the following points:

1.The sideward orientation leads to the best enhancement in cooling the pin finned heated plate represented by decreasing the base plate temperature and gives a uniform temperatures distribution.

2.The heat transfer coefficients of the sideward orientation are about (7%) greater than those for the upward orientation and exceeds that of downward facing by (24%).

3. In upward facing the difference between the base plate temperature and the ambient temperature increased by 16% relevant to sideward, while in the downward facing it was increased by 22%.

4.The non-dimensional temperature for the upward facing in comparing with sideward increased by 5.6% ,while the downward increased by 4.6% relative to sideward facing at (x=8mm).

5. The fins in downward facing at (x=35mm & x=60mm) became cooler than the sideward facing, while in upward facing the non-dimensional values stays higher than the sideward and downward.

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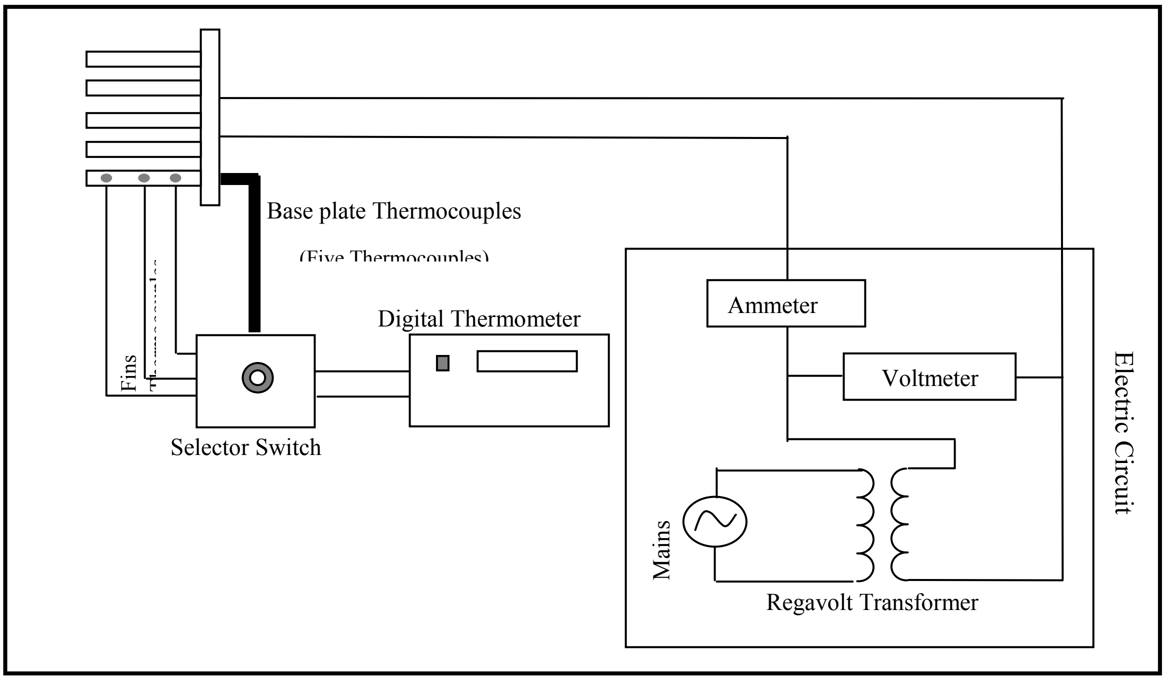

Fig. (1) Schematic of Experimental apparatus

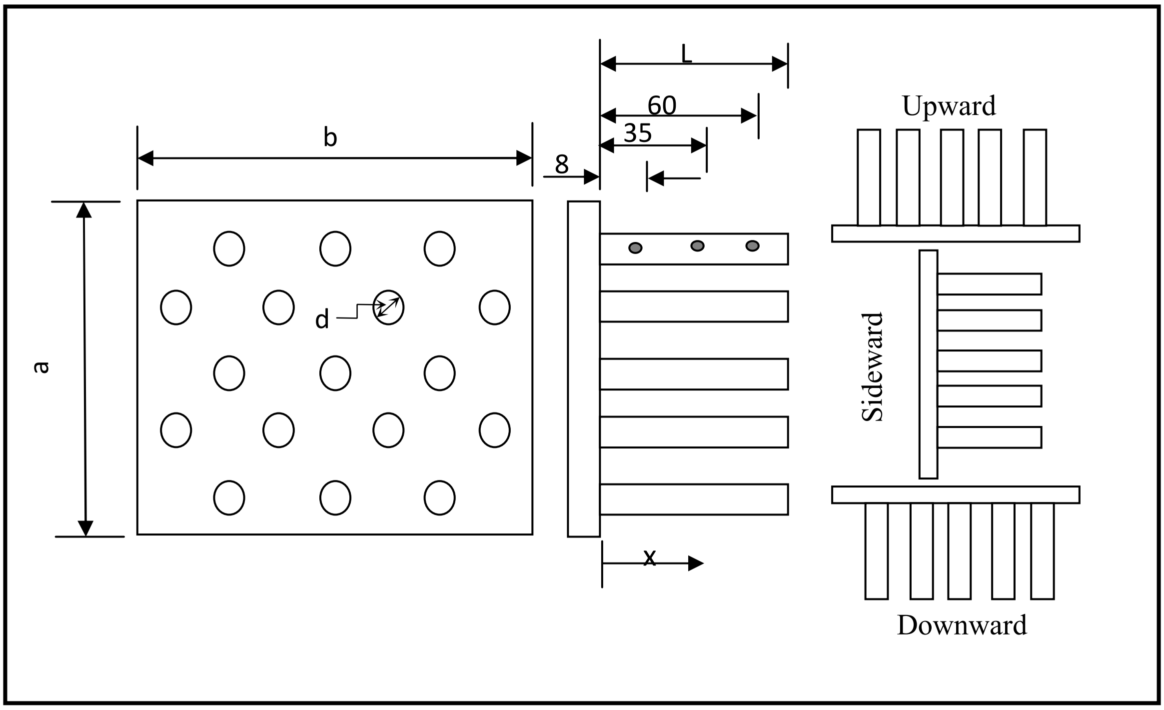

Fig. (2) Dimensions and Orientations of pin finned heated plate in (mm);
a=100mm, b=110mm, d=12mm, L=66mm

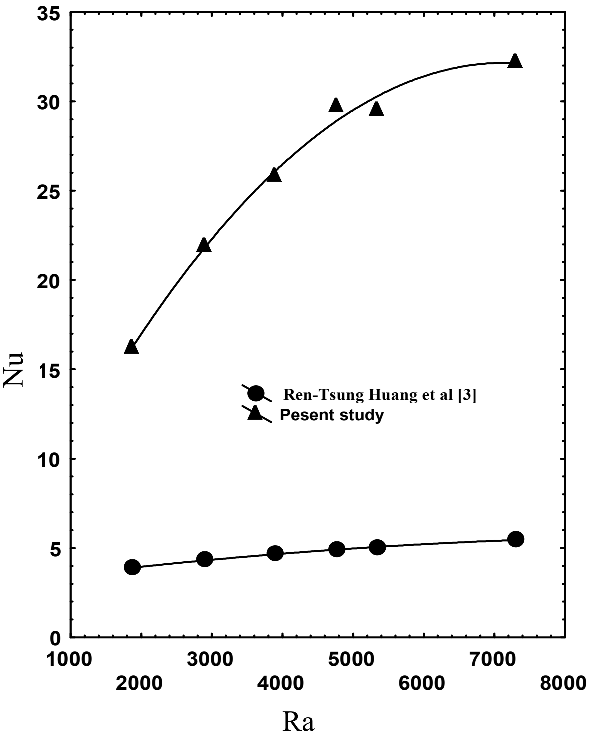

Fig. (3) Comparison of variation Nusselt number with Rayleigh number

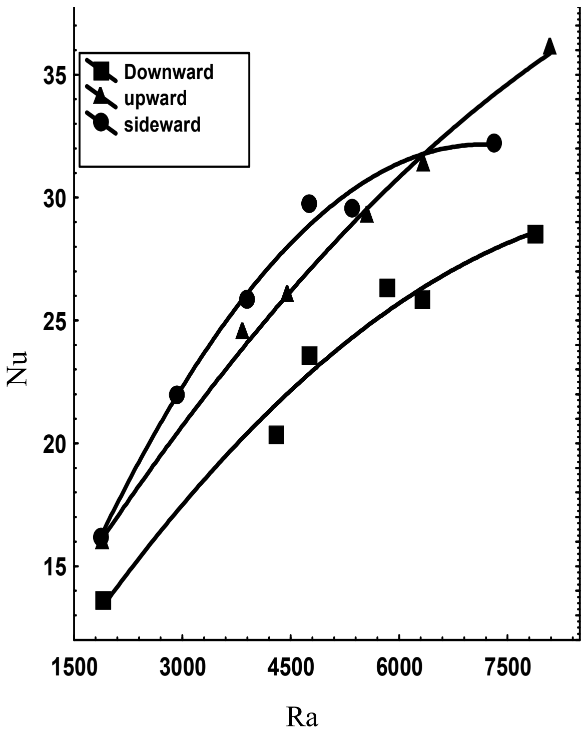

Fig.(4) Relation between Nu & Ra with orientation

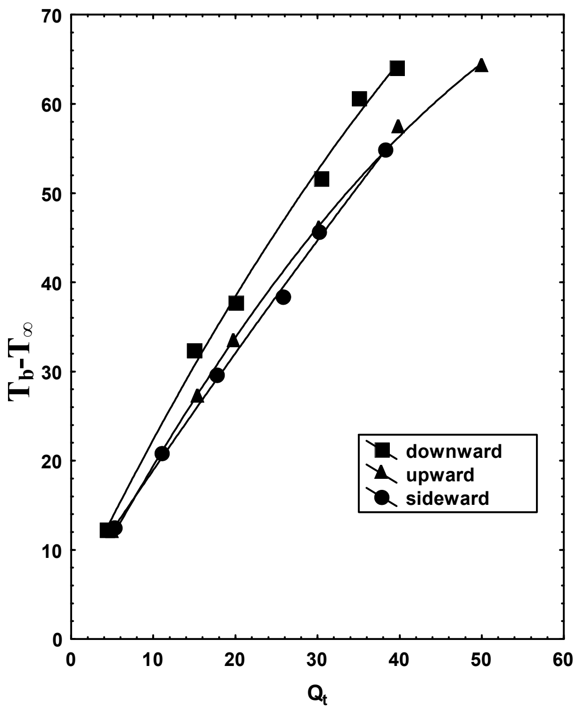

Fig.(5) Effect of orientation on base temperature deference

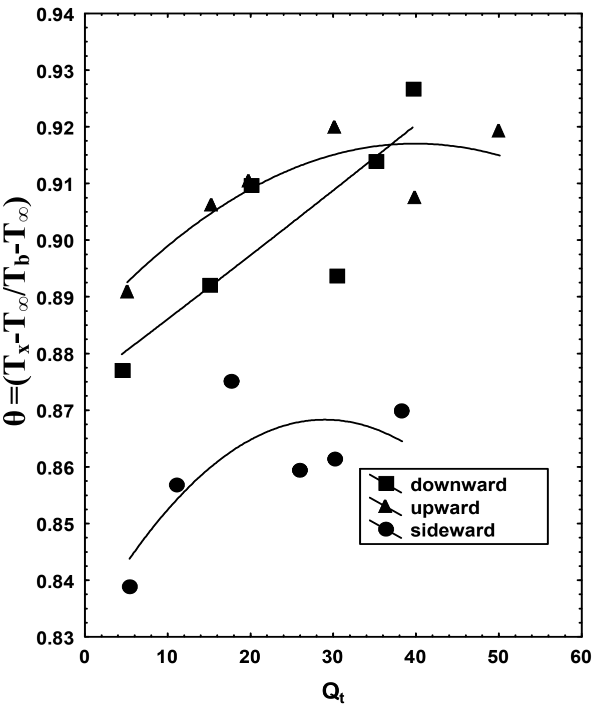

Fig.(6) Effect of orientation on a non-dimensional temperature at (x=8mm)

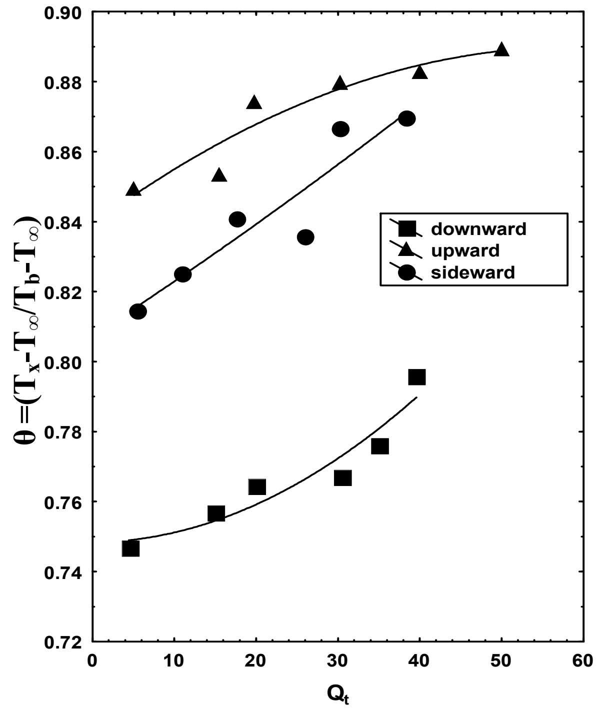

Fig.(7) Effect of orientation on a non-dimensional temperature at (x=35mm)

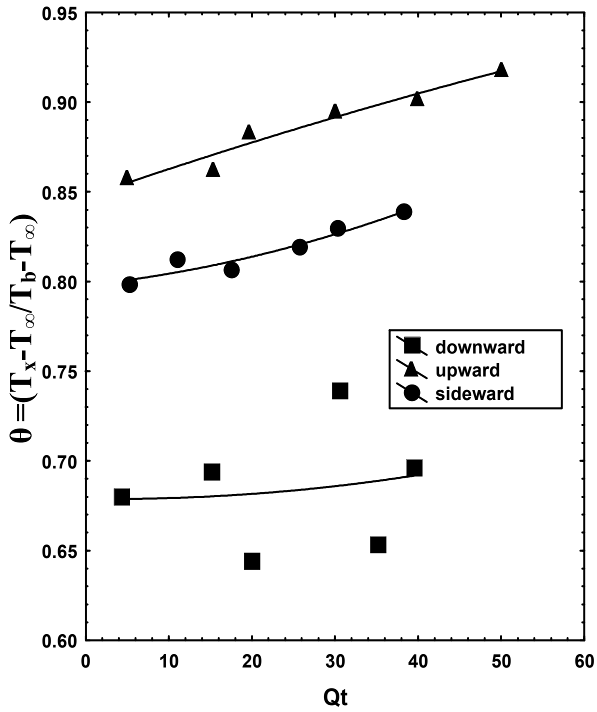

Fig.(8) Effect of orientation on a non-dimensional temperature at (x=60mm)