Effect Orientation on Performance of Longitudinal (Trapezoidal) Fins Heat Sink Subjected to Natural Convection

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Abstract:
Experiments were carried out on natural convection heat transfer from longitudinal trapezoidal fins array heat sink subjected to the influence of orientation. A trapezoidal fins heat sink with various orientations tested under a controlled environment. Test results indicate that the sideward horizontal fin orientation yield the lowest heat transfer coefficient. However the sideward vertical fin orientation gave the best performance on the natural cooling. From the experiments Nu is determined as a function of Ra at Pr=0.7 for each orientation with Ra ranging between (1400 and 3900).From the results; Heat transfer coefficient of the sideward vertical fins is higher by (12%) than the heat transfer coefficient of the upward while it is higher than the heat transfer coefficient of the downward by (26%) and by (120%) with the sideward horizontal fins. Orientation affected the temperature distribution along the fins, therefore the temperature along the sideward vertical fins have the best performance with uniform distribution, while in sideward and downward the temperature increased in the positions near the base plate surface because of the complication in moving the heated air.

Keywords:
Natural convection, Orientation effect, Longitudinal, Trapezoidal, fin.

1. INTRODUCTION

Natural convection heat transfer from finned surfaces has been the subject of a large number of experimental and theoretical investigations. Extended surfaces or fins are used to increase the heat dissipation in many engineering and industrial applications by increasing the heat transfer area. They are used in cooling of electronic and thermoelectronic devices, solar energy applications, cooling of nuclear reactor fuel elements, improving heat transfer in radiators for air conditioning and in air cooled heat exchangers. In most of these devices air is the heat transfer medium. At the air side of any heat exchanger, fins are used with different types.

The type of fins used depends on manufacturing process and the space available in heat generated devices to be involved in cooling process. Natural convection from heat sinks has long been utilized for the thermal management of low-power-density devices. With the features of high system reliability, free of maintenance, and zero power consumption, this cooling technique plays an important role in the electronic cooling industry and has attracted considerable researches for decades.

Natural convection from extended surface in different orientations can be found from experimental data [1-6] and theoretical works [7-12].Zografos and Sunderland [3] investigated the heat transfer performance of inline and staggered pin fin arrays in natural convection with deferent inclination angle, and concluded that the inline arrays generally yielded higher heat transfer rates than the staggered ones. In addition, their investigation showed little influence of inclination when the inclination angle was less than 30^{o} from the vertical.

Kobus and Oshio [4] carried out a comprehensive theoretical and experimental study on the thermal performance of a pin-fin heat sink. A theoretical model was formulated that has the capability of predicting the influence of various geometrical, thermal, and flow parameters on the effective thermal resistance of the heat sink. Experiments were carried out, and correlations obtained, for a wide range of parameters for pure natural convection and for combined forced and natural convection. The predictive capability of the theoretical model was verified by comparison with experimental data including the influence of various fin parameters and the existence of an optimum fin spacing. Sparrow and Vemuri [5] studied the fin orientation on natural convection/radiation heat transfer from pin fin arrays. Their results revealed that the upward facing orientation yielded the highest heat transfer rates, followed by the sideward facing and the downward facing ones. Dialameh,et al. [7] made a numerical study to predict natural convection from an array of aluminum horizontal rectangular thick fins of (3 mm< t < 7 mm) with short lengths (L< 50 mm) attached on a horizontal base plate. The results were natural convection heat transfer coefficient increases with increasing temperature differences, it is increases with fin spacing and decreases with fin length,and the fin thickness and fin height does not affect the value of average heat transfer coefficient considerably.

Baskaya Senol, et al. [11], carried out parametric study of natural convection heat transfer from the horizontal rectangular fin arrays. They investigated the effects of a wide range of geometrical parameters like fin spacing, fin height, fin length and temperature difference between fin and surroundings; to the heat transfer from horizontal fin arrays. However, no clear conclusions were drawn due to the various parameters involved. Finally they concluded that, it is not possible to obtain optimum performance in terms of overall heat transfer by only concentrating on one or two parameters. The interactions among all the design parameters must be considered. This study has shown that each of the variables produces an effect on the overall heat transfer. As a whole, it can be concluded that the overall heat transfer is enhanced with the increase in H, the height of the fin and with decrease L, the length of the fin.Karagiozis et al. [12] reported experiments with triangular fins in vertical and horizontal arrangement under natural convection heat transfer process.

Kang and Look [13] presented a comparison of the heat loss from trapezoidal fins of various slopes (triangular to rectangular).The method used produces accurate solutions to the general conduction equation and Fourier law for (Bi <0.1). The resulting relationship between the non-dimensional fin length and the slope of the fin's lateral surface for equal amounts of heat loss for given values of Biot numbers is also demonstrated. For that analysis, the same equation is used for all of the trapezoidal shapes by adjusting the value of the lateral surface slope. For simplicity, the fin root temperature and the fin's surrounding convection coefficients are assumed constant.Bejan and Almogbel [14] reported the geometric (constructal) optimization of T-shaped fin assemblies, where the objective is to maximize the global conductance of the assembly, subject to total volume and fin-material constraints. Hyung [15] analyzed and optimized a triangular fin with variable fin base thickness using a two-dimensional analytical method. For the fixed fin volumes, the maximum heat loss, the corresponding optimum fin effectiveness, fin base height and fin tip length as a function of the fin base thickness, convection characteristic number and dimensionless fin volume its represented. One of the results shows that the optimum heat loss increases whereas the corresponding optimum fin effectiveness decreases with the increase of fin volume.

Although many previous researches[1,3,5,7,8,12,18,21]were made towards the understanding of natural convection from a round pin fin, rectangular pin fin, longitudinal rectangular fin, triangular and trapezoidal fin, nearly no experimental data are available for the trapezoidal fin in different orientations. In particular, few researches have explicitly demonstrated the subtle difference of orientation effect on the fin performance, which is of practical importance for relevant application. For this reason, the objective of this study is to provide experimental data for trapezoidal longitudinal fin heat sink under natural convection with different orientation.

2. EEPERIMENTAL SET-UP PROCEDURE

The experimental facility developed for the investigation is shown in **Fig.(1)**.Experiments are carried out on natural convection heat transfer from longitudinal trapezoidal fins array heat sink placed in different orientations (Upward face, Downward face, Sideward face-vertical fins, and Sideward face-horizontal fins) by tilting mechanism. The tilting mechanism is designed to allow for the heat sink to move in angular motion from (0^o to 360^o). The base plate of the heat sink has the dimensions of (100mm x 110mm).The base plate is equipped with an electrical heating element with thermostatic protection against overheating was 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 a five calibrated thermocouples (copper-constantan), placed at a symmetric location in the base plate. The fins are manufactured from polished aluminum as one part of the base plate. **Fig.(2)**shows the dimensions of the trapezoidal fins array heat sink. During an experiment, the temperatures along the fins array are measured by thermo-couples made from copper-constantan. An electrical console in corporate a variable auto-transfer with a digital readout to control and indicate the power supplied to the fin heated heater as shown in **Fig.(3)**.

All experiments performed inside two chambers the first represent a Nylon box of (0.5mx0.5m) of 0.7m height, used to observe the electrical apparatus and have a stable medium around the heat sink. The second chamber of wood(2mx2m) of 2.5m height containing all apparatus and nylon box. All experimental reading take placed in cold environments(winter season),therefore the experimental set-up provided by a control heating system with thermostat to control the ambient temperature (Ta) at 25^oC.The heat sink is tested at ambient temperature with constant heat flux condition ranging from 10W to 55W.More than two hours are required to reach the steady state condition.

3. CALCULATION PROCEDURE

In the present study the ambient temperature is always controlled at 25^oC,and thermophysical properties in Nusselt and Rayliegh numbers are evaluated at the film temperature(T_f),

$$T_f = \frac{1}{2}(T_a + T_b) \tag{1}$$

If the base temperature of the heat sink T_b is the average temperature of the five thermocouples that placed in the base plate, then the non-dimensional temperature is given by

$$\theta = \frac{T - T_a}{T_b - T_a} \tag{2}$$

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

$$Q_C = Q_t - Q_R \tag{3}$$

$$Q_R = \varepsilon.\sigma.A_t\left(T_b^4 - T_a^4\right) \tag{4}$$

where Q_t ,represent the power input to the heater, the heat transfer coefficient can be calculated from the following

$$Q_C = n.Q_{fin} + Q_b$$
(5)

where:

$$Q_{fin} = A_{fin}.\eta_{fin}.h.\theta_b \tag{6a}$$
$$Q_b = A_b h\theta_b \tag{6b}$$
$$\theta_b = T_b - T_a \tag{6c}$$

The total heat transfer surface consisting from surface fins area(A_{fins}=nA_{fin}=0.1226m^2) and the base plate without fin area(A_b=0.0065m^2),therefore(A_t=0.1291m^2).By substitution eqs (6a&6b) in eq. (5),the convective heat transfer coefficient can be expressed as

$$h = \frac{Q_C}{\left(0.1226\eta_{fin} + 0.0065\right).\theta_b} \tag{7}$$

Where (η_{fin}) is the fin efficiency defined as

$$\eta_{fin} = \frac{\tanh(mH)}{mH}, m = \left(\frac{hp}{kA}\right)_{fin}^{\frac{1}{2}} \tag{8}$$

It is recognized that because the fin efficiency also involves the convective heat transfer coefficient,(h),an iteration scheme must be used to solve for (h) for each experimental data point.

The Nusselt and Rayleigh numbers are calculated as[10]

$$Nu = \frac{hb}{k_f} \tag{9}$$

Where k_f is the conduction heat transfer coefficient for air at $T_{f,}$ and b is the mean distance between two fins, which is equal to 0.01m.

$$Ra = Gr.\Pr = \frac{g\beta\left(T_b - T_a\right)b^3}{\alpha\upsilon} \tag{10}$$

Where $\beta = \frac{1}{T_f}$ .The relation between Nusselt and Rayleigh numbers for each orientation will be calculated as following

$$Nu = c(Ra)^n \tag{11}$$

4. RESULTS AND DISCUSSION

As mentioned in previous sections, the experiments were conducted at different orientations and steady state conditions. From measured parameters the heat transfer coefficient is determined and plotted against the base temperature difference. To verify the experimental data,the upward orientation of the present study was compared with the results presented by Dialameh and Yaghoubi[7],as shown in **Fig.(4)**.The present experimental results are in a good agreement with the numerical results of [7].

The effect of orientation on the heat transfer coefficient is also shown in **Fig.(5)**.In essence, the sideward horizontal fins (HF) orientation yields the lowest heat transfer coefficient and the heat transfer coefficient of the downward orientation has a value higher than it. However ,one can notice the heat transfer coefficient for the upward and sideward vertical fins (VF) orientations are nearly comparable magnitude.

Fig.(5) shows that the best results of heat transfer coefficient occurs for the sideward VF orientation with positive behavior between the heat transfer coefficient and the temperature difference, as a result of the bouncy force produced by the change of air density when the temperature difference is found, that is lead to the enters of air near the bottom of the fin channels, also some air will be inflow from the fin tips. Air is heated within the fin channels and exits at the top. With this simple air flow path ,the sideward VF orientation delivers the best performance for natural cooling. The heat transfer coefficient of the upward orientation is less than that is for sideward orientation by 12% because of the difference in the phenomena of cooling. For the upward there is an inflow from the ends of the heat sink, however, other flow patterns can coexist with the inflow from ends. In particular thermosiphoning can occur both longitudinally and between fins. In thermosiphoning, cooling air is drawn down into the fins and then rises in a cellular pattern, and it is important to note that the thermosiphoing effect does not occur in any significant manner for the sideward VF orientation.

Also **Fig.(5)** shows that the increase in temperature difference leads to decreasing in heat transfer coefficient for the sideward HF and downward orientation. However from the experimental results the heat transfer coefficient of the downward face is less than that of sideward VF by 26%,but it is higher than that of sideward HF by 71% because of the decreasing in natural cooling at these two orientations. In the downward orientation the air enters from down to the heat sink with the fin ends ,it will be heated by the fins surface causing to rise towards the heat sink surface (channel base area) before leaving the channels from side, this manner is the main reason that decreases the natural cooling. In sideward HV the problem will be increased and the cooling is more bad than other orientations because the heated air that inters to the fins will hit the surface of the fins before leaving slowly the sides, that leads to heated the heat sink as a result of the complicated manner of inters and exit the air.

Fig.(6) shows the variation of Nussel number with Rayliegh number for the four orientations. Rayliegh number ranges between 1400 to 3900 for the all experiments, and nusselt number ranges between 0.9 to 3 .From fig.6 we see that the value of Nusselt number increase with the increasing of Rayliegh number for the sideward VF and upward orientations, but it decreases when Ra>2700 for the downward and sideward HF.A linear logarithm regression between Nusselt and Rayliegh is performed for all experiments of the heat sink in each orientation. The following relations are obtained for Nu as a function of Ra:

Sideward VF , $(1663 \le Ra \le 3540)$

$$Nu = 1.911\, Ra^{0.31} \tag{12}$$

Upward, $(1400 \le Ra \le 3645)$

$$Nu = 1.80 Ra^{0.256} \tag{13}$$

Downward, $(1500 \le Ra \le 3900)$

$$Nu = 1.593 Ra^{0.252}$$
(14)

Sideward HF, $(2120 \le Ra \le 3.85)$

$$Nu = 0.771 Ra^{0.382} \tag{15}$$

The validity of these relations are with the ranges of Ra shown with each orientation and Pr=0.7,for a longitudinal trapezoidal fins array heat sink.

Fig.(7-9)shows the effect of orientation on dimensionless temperature in different locations along the fin. From the figures the best performance of temperature distribution along the fin occur in sideward VF arrangement, **Fig.(7)** represents the relation between the dimensionless temperature and the total power input at 0.12H, one can notice, that the lowest temperature for the same power input is represent in the sideward VF arrangement and decreases with the increase of the power input, when it is increased with increasing the input power for the sideward HF. From **Fig.(9)**,at 0.85H location that is near to the fin end the temperature of the sideward HF is decreased with increase in the input power that is true because the air rises and reaching the channel surface causing heated the heat sink and reducing the fin end temperature with increasing power input as a result of the manner of moving the heated air ,as it is discussed above.

5. CONCUSIONS

The heat transfer characteristics of longitudinal trapezoidal fins array heat sink subjected to the influence of orientation are examined experimentally under natural convection in the present study. Based on the foregoing discussion; the following conclusions were made;

1) Generally, the sideward HF orientation yields the lowest heat transfer coefficient, after the downward orientation comes. However, the heat transfer coefficient for sideward VF orientation is the best, and the upward comes after it.
2) The performance of sideward VF orientation superior to that of upward orientation by (12%),to downward by (26%),and by (120%) to the sideward HF orientation.
3) The regressions obtained for experiments for each orientation indicate that Nu for the sideward orientation is higher than the other three orientations.
4) The orientation effect in temperature distribution along the fins as a result of the manner of moving heated air and the rat of it is flow.

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NOMENCLATURE

A	Area,m^2
A_{fin}	Fin surface area,m^2
b	Mean distance between two fins,m
c	constant
Gr	Grashof number, dimensionless
g	Gravity, m s^{-2}
H	fin height, m
h	Convective heat transfer coefficient, W m^{-2} K^{-1}
HF	Horizontal fin
k	Thermal conductivity, W $m^{-1}K^{-1}$
L	Base plate length, m
n	constant
N	Number of fins, dimensionless
Nu	Nusselt number, dimensionless
p	Periphery of fin, m
Pr	Prandtl number, dimensionless
Q	Rate of heat transfer, W
Q_{fin}	Convection heat transfer from fin, W
Ra	Rayleigh number, dimensionless
T	Temperature along fins, ^oC
VF	Vertical fin
W	Fin width, m

Greek Symbols

α	Thermal diffusivity of air,m^2 s^{-1}
β	Coefficient of thermal expansion ,K^{-1}
ε	Radiative emissivity, dimensionless
η_{fin}	Fin efficiency,dimensionless
θ	Non-dimensional temperature
υ	Kinematic viscosity of air, m^2 s^{-1}

Subscripts

a	ambient
b	base plate
C	convection
f	film
R	Radiation

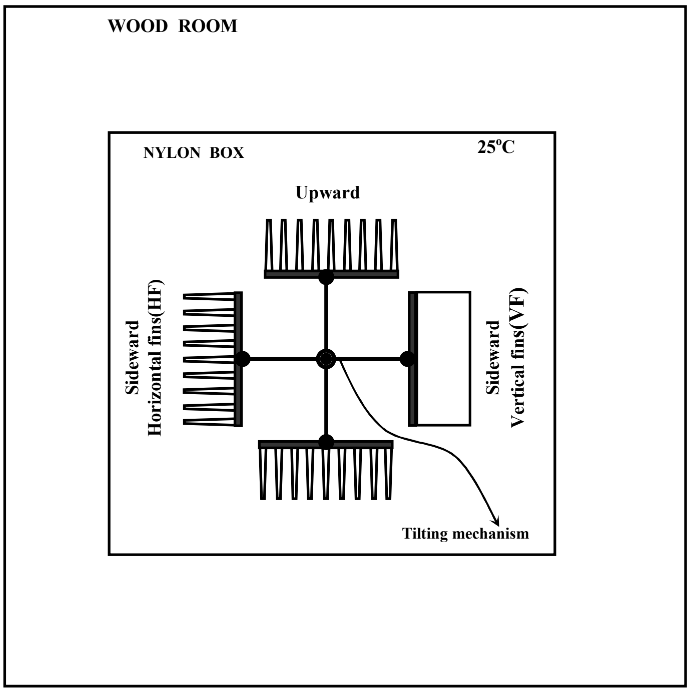

Fig. (1): Experimental set-up

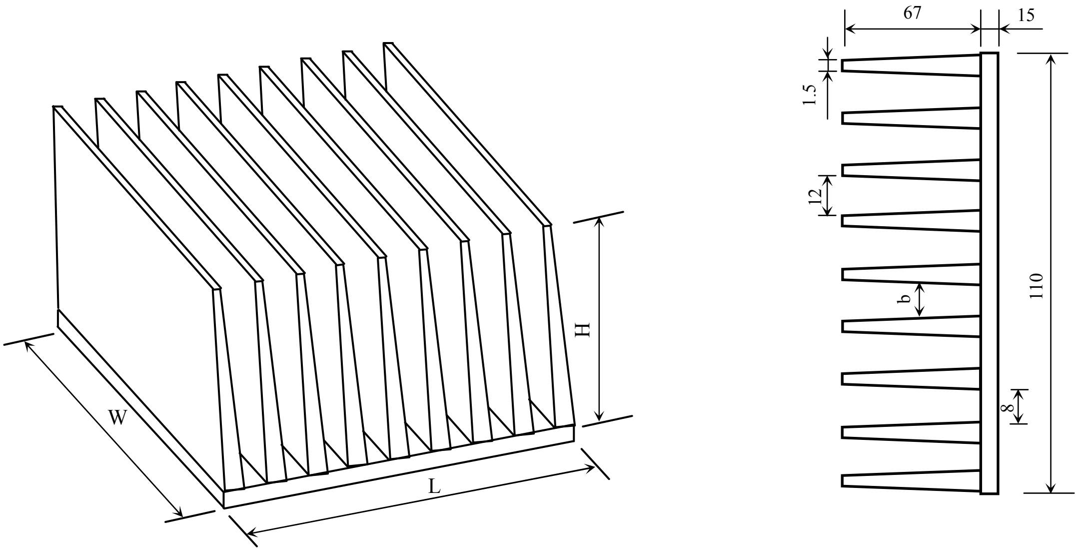

Fig. (2): Trapezoidal fins array dimensions (in mm);W=100,b=10

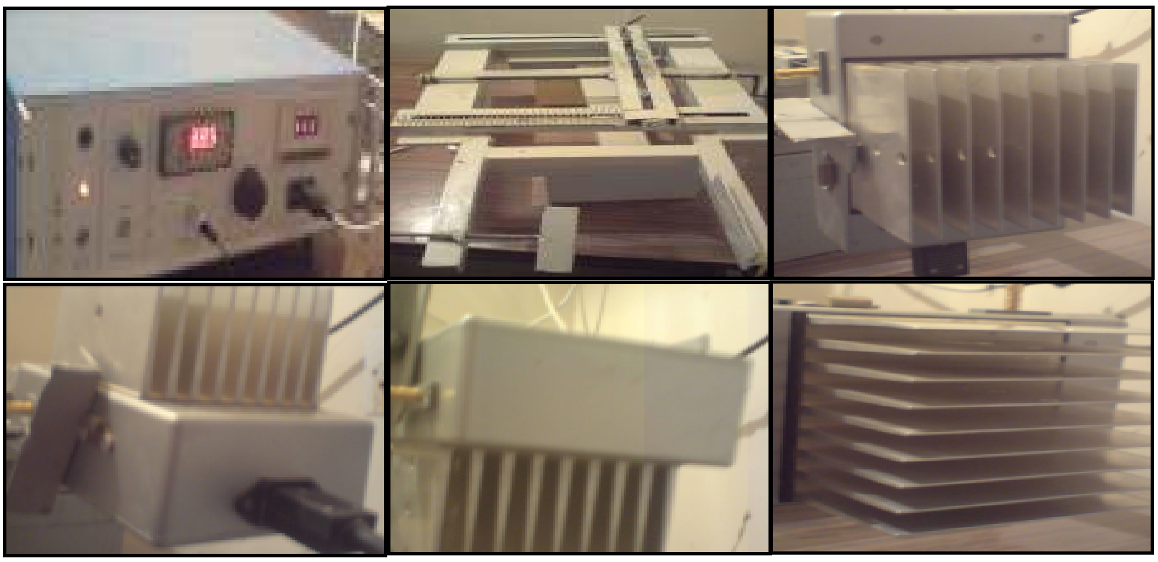

Fig. (3): Photographs of experimental set-up parts

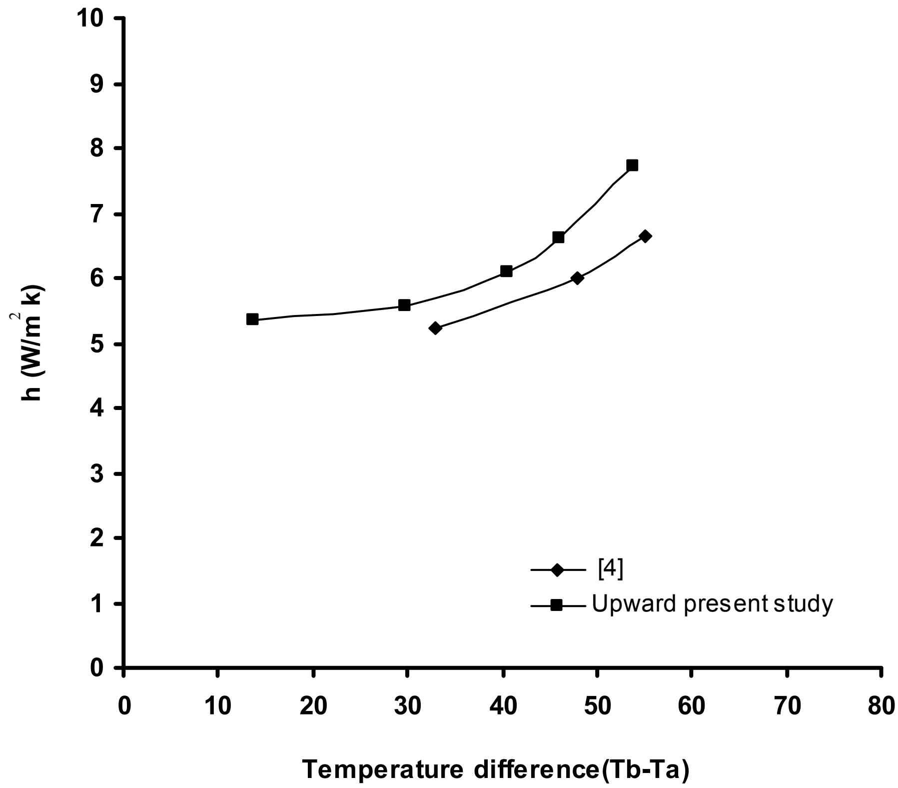

Fig. (4): Comparison of variation of heat transfer coefficient with temperature difference

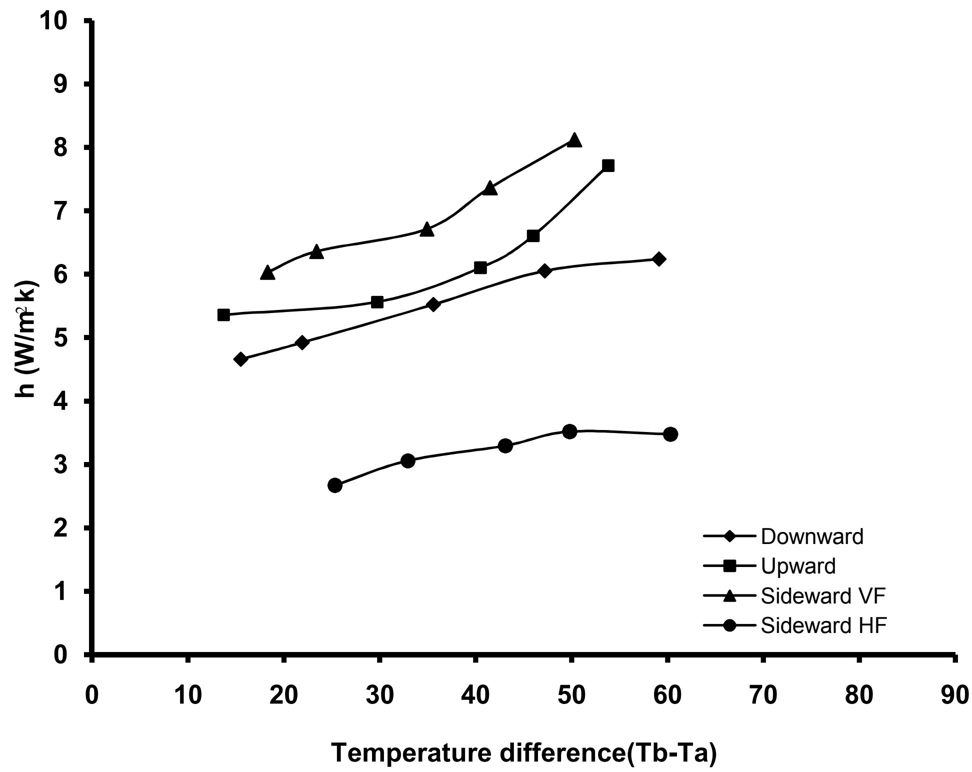

Fig. (٥): Heat transfer performance for all four orientations

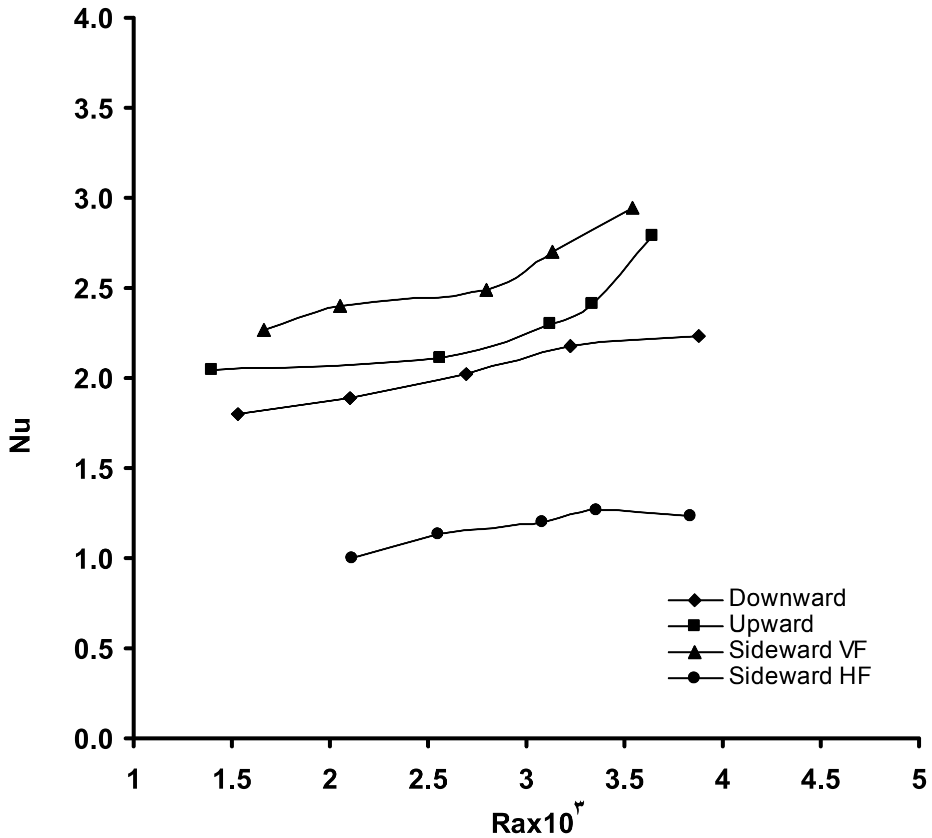

Fig. (٦): The relation between Nusselt and Rayleigh numbers

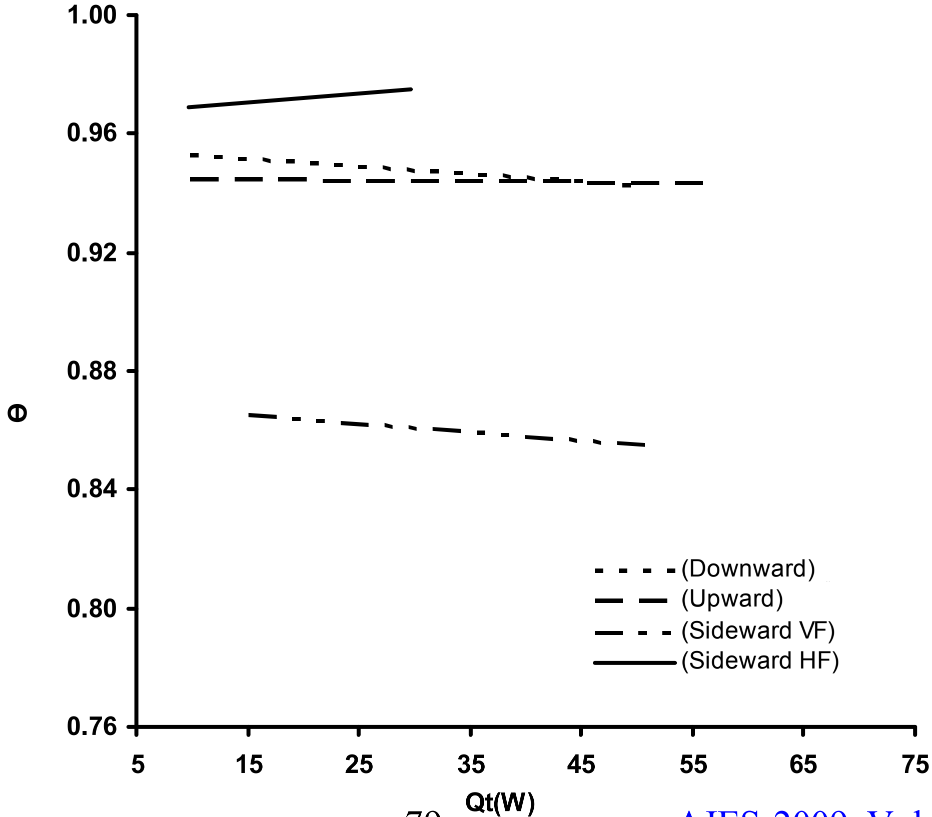

Fig. (7): Temperature distribution along the fins, at 0.12H

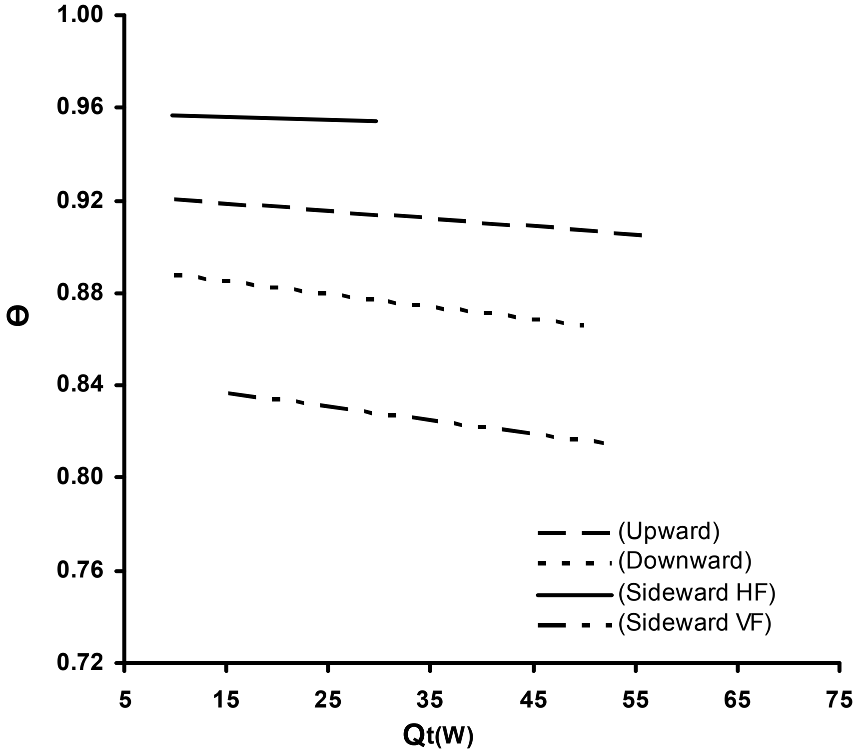

Fig. (٨): Temperature distribution along the fins, at 0.52H

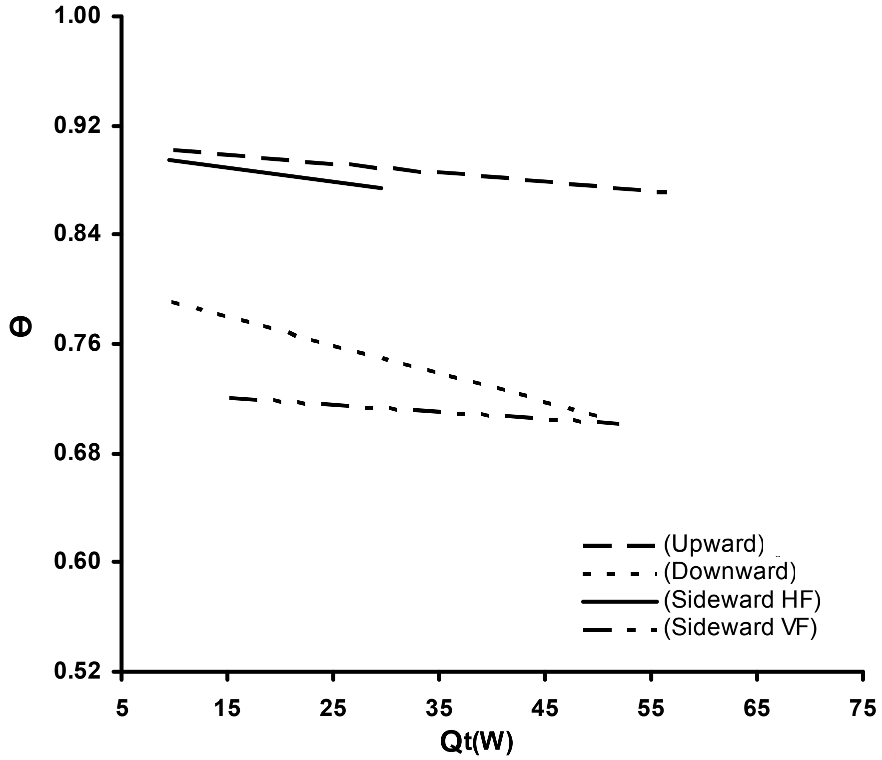

Fig. (9): Temperature distribution along the fins, at 0.85H

تأثير اتجاه الزعنفة على انتقال الحرارة بالحمل الحر من سطح ذو زعانف طولية

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الخلاصة:

أجريت التجارب المختبرية على انتقال الحرارة بالحمل الحر من سطح ذو صف من الزعانف الطولية وبأوضاع مختلفة.تم اختبار السطح تحت ظروف ثابتة لجميع الأوضاع.نتائج الاختبار بينت انه الحالة الجانبية للسطح والتي يكون فيها وضع الزعانف أفقي أدت إلى اقل معامل انتقال حرارة بالحمل الحر.أما الحالة الجانبية التي يكون فيها وضع الزعانف عامودي فقد أعطت أفضل أداء لعملية تبديد الحرارة.من النتائج العملية تم إيجاد عدد نسلت (Nu)كدالة لعدد رالي(Ra) عند عدد برانتل (0.7) لكل حالة ضمن مدى لعدد رالي(1400-3900).من النتائج تبين إن معامل انتقال الحرارة بالحمل الحر للحالة الجانبية بزعانف عاموديه أعلى بنسبة (12%) عن معامل انتقال الحرارة للحالة الأفقية بزعانف متجهة للأعلى في حين يكون أعلى من الحالة الأفقية ذات الزعانف المتجهة للأسفل بنسبة (26%) وبنسبة (120%) عن الحالة الجانبية ذات زعانف أفقية.وضع الزعنفة اثر كذلك على توزيع درجة الحرارة على امتداد سطح الزعنفة.من النتائج للحالات الأربعة تبين أن توزيع درجة الحرارة على امتداد سطح الزعنفة للحالة الجانبية عاموديه الوضع كان أفضل أداء مع توزيع منتظم ،في حين تبين انه في الحالة الأفقية للزعانف المتجهة إلى الأسفل والحالة الجانبية الأفقية الزعانف درجة الحرارة ترتفع بالقرب من القاعدة وذلك بسبب الإعاقة التي يواجهها الهواء الساخن.