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# VERTICAL RECTANGULAR FINS ARRAY DESIGN EXPERIMENTAL AND THEORITICAL COMPARISONS

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## ABSTRACT

*Experimental and theoretical comparison have been performed for natural convection heat transfer over rectangular fins array at different fin parameters. This investigation includes the effect of fin length, fin spacing, fin height, orientation angle, and temperature difference between the heat sink and the surrounding environment. To understand the general flow patterns dominating flows from the heat sink, the three dimensionless elliptic governing equations were solved using finite volume computational fluid dynamics (CFD) code, and the experimental work was carried for the system at different orientations. A new empirical correlation (modified of McAdam's correlation) was derived to correlate the mean Nusselt number as a function of the Rayleigh number.*

*The average heat transfer coefficient has a maximum value at an orientation angle equal to zero degrees, and it decreases with an increasing orientation angle. The heat transfer rate per unit base area increases as fin spacing increase until it reaches a maximum value (6.5 mm), then it decreases with a further increase of fin spacing.*

*The results of these investigations between the experimental and theoretical study were showing good agreements with similar international works.*

**Keywords: Free** convection Heat transfer; Fins; Laminar Flow; Natural convection; CFD.

## 1. INTRODUCTION

Heat removal in an efficient way is necessary to maintain reliable operation of electronic devices, solar energy applications automobile cooling systems, refrigeration, and air conditioning systems, and cooling of nuclear reactor fuel elements in the natural convection heat transfer processes. For an efficient application of natural convection to cooling processes it is necessary to fully understand the mechanisms involved in the free-convection cooling of any heat transfer system [1].

The enhancement of heat transfer by using the natural convection air-cooling extended surfaces provides a reliable, cheap, and widely used method of cooling for dissipating unwanted or excess heat from the system. Besides, their design is simple, economic and without any acoustic noise, the convective heat transfer from an extended surface can be increased either by increasing the heat transfer coefficient or the surface area or both of these quantities. Increasing the heat transfer area is preferred as the simplest method to enhance the heat dissipation rate because the use of better fluid to increase the heat transfer coefficient is not an economical and practical solution. The only controllable variable to enhance the convective heat transfer rate from an extended surface is the geometry of the fins (rectangular, cylindrical, and other shapes) [2].

Some previous works were done including all types of extended surfaces to enhance heat transfer by natural convection as:

Burak, et. al., were investigated the optimum fin spacing of vertically-based rectangular fin arrays subjected to natural convection heat transfer [3]. Dharma, et. al., were studied the laminar natural convection heat transfer problem from a fin array containing a vertical base and horizontal fins is theoretically formulated by treating the adjacent internal fins as two fin enclosures [4]. Edlabadkar et al. were reported a theoretical work on the enhancement of natural convection heat transfer from a vertical plate with a horizontal partition plate and V-plates in water as ambiance [5]**.** Arularasan et. al. were reported an analysis using CFD techniques in a heat sink for cooling electronic devices [6]. Denpong et. al. were investigated the effect of different fin geometries for heat sinks on laminar-forced heat transfer by convection [7]. Abdel-Latif A., et. al., were investigated the orientation effect on the natural convection performance of longitudinal fin arrays heat sink [8]. Bsebsu, et al, were reported an experimental work about the characteristics of a vertical rectangular fin. [9] Bsebsu, et al, were reported a theoretical work using CFD methods for modeling and simulation of vertical rectangular fins array [10]. Although many previous types of research [11–23] were made towards the understanding of natural convection from fin arrays, nearly no experimental and numerical data are available for the longitudinal fins which are also a very popular fin configuration used for electronic cooling and other applications. In particular, few kinds of research have explicitly demonstrated the subtle difference of orientation effect on the fin performance, which is of practical importance for relevant applications. For this reason, the first object of this study is to provide experimental and numerical data for longitudinal fin heat sink under natural convection. Secondly, the dependence of longitudinal fin performance on the orientation effect is presented in a more detailed manner.

## 2. EXPERIMENTAL WORK

The experimental work was carried out to investigate and study the heat sink thermal behavior [9], which consists of a square base plate, tested fins, and a heating unit. All the parts were assembled inside a wooden container with dimensions of 417 mm wide, 390 mm long, and 205 mm height as shown in Figure (1). The base plate is made from pure aluminum metal, 187 mm long. The tested fins are rectangular shape, and it was made from the same metal of the base plate material (pure Aluminum), and it has the following parameters:

1. Fin height (H = 15, 35 and 45mm),
2. Fixed fin thickness (t = 6.5mm),
3. The inter-fin space (s = 3, 6.5 and 16 mm),
4. Fixed fin numbers (n),
5. Variable rotating angle ($\beta = 0°$, 30°, and 60°).
6. Eight copper-constantan (Type T) and two nickel-chrome (Type K) thermocouples were distributed uniformly along each side to measure the temperature distribution through the fins.
7. The fins are mounted on the base plate, with inter-fin spacing (s) the distance between the two adjacent fins as showing in Figure 1.

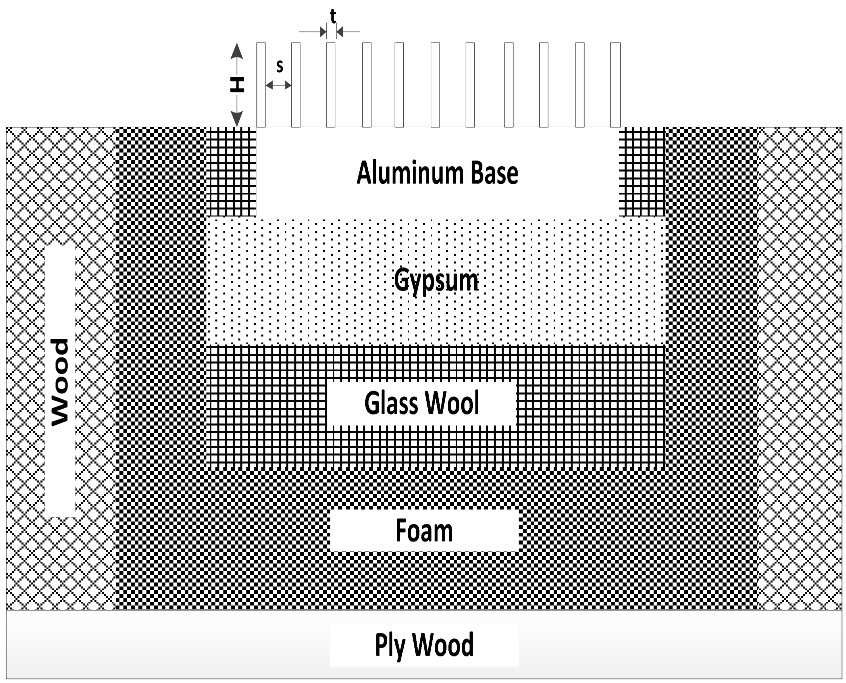



**Figure 1**. Schematic Diagram of the Testing Device.

**3.**
**THEORETICAL AND NUMERICAL METHODS**

A numerical solution was obtained with the CFD code. In principle, the code solves the governing set of elliptic partial differential equations, where the natural convection flow under investigation was modeled by a set of elliptic partial differential equations describing the conservation of mass, momentum, and energy in three rectangular cartesian coordinate directions for air and fin [10].

**1. Governing Equations For Air:**

• Continuity: $\frac{\partial(\rho u)}{\partial x}+\frac{\partial(\rho v)}{\partial y}+\frac{\partial(\rho w)}{\partial z}=0$ (1)

• Momentum in x, y, z directions are:

$$\frac{\partial(\rho u^2)}{\partial x}+\frac{\partial(\rho uv)}{\partial y}+\frac{\partial(\rho uw)}{\partial z}=-\frac{\partial(P_m)}{\partial x}+\rho v\left[\frac{\partial^2 u}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right] \tag{2}$$

$$\frac{\partial(\rho vu)}{\partial x}+\frac{\partial(\rho v^2)}{\partial y}+\frac{\partial(\rho vw)}{\partial z}=-\frac{\partial(P_m)}{\partial y}+\rho \upsilon\left(\frac{\partial^2 v}{\partial x^2}+\frac{\partial^2 v}{\partial y^2}+\frac{\partial^2 v}{\partial z^2}\right)+g(\rho-\rho_\infty) \tag{3}$$

$$\frac{\partial(\rho wu)}{\partial x}+\frac{\partial(\rho wv)}{\partial y}+\frac{\partial(\rho w^2)}{\partial z}=-\frac{\partial(P_m)}{\partial z}+\rho \upsilon\left(\frac{\partial^2 w}{\partial x^2}+\frac{\partial^2 w}{\partial y^2}+\frac{\partial^2 w}{\partial z^2}\right) \tag{4}$$

• Energy:

$$\frac{\partial(\rho uT)}{\partial x}+\frac{\partial(\rho vT)}{\partial y}+\frac{\partial(\rho wT)}{\partial z}=\frac{\upsilon}{Pr}\left(\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial w^2}\right) \tag{5}$$

2. **Governing Equations for Fin**

$$\frac{\partial^2 T}{\partial x^2}+\frac{\partial^2 T}{\partial y^2}+\frac{\partial^2 T}{\partial z^2}=0 \tag{6}$$

The assumptions employed in the governing equations are in agreement with:

1. Natural convection is the mechanism for heat transfer from the fin array.
2. Radiation heat loss is neglected.
3. The buoyancy forces representation is based on the Boussinesq approximation. The Boussinesq model treats density as a constant value in all solved equations, except for the buoyancy term in the momentum equation as: $(\rho-\rho_0)g \approx -\rho_0\beta(T-T_0)g$ , Where $\rho_O$ is the constant density of the flow, $T_O$ the operating temperature, and $\beta$ the thermal expansion coefficient. By applying the Boussinesq approximation to eliminate $\rho$ from the buoyancy term in the equation (3). This term is reduced to: $\rho=\rho_o(1-\beta\Delta T)$ .
4. The flow is, therefore considered as essentially incompressible. the density will be calculated as: $\rho=\left(P_o/(R/M)T\right)$
5. The fluid properties are assumed constant and are evaluated at the average of hot surface and the ambient fluid temperature.
6. Steady state natural convection.

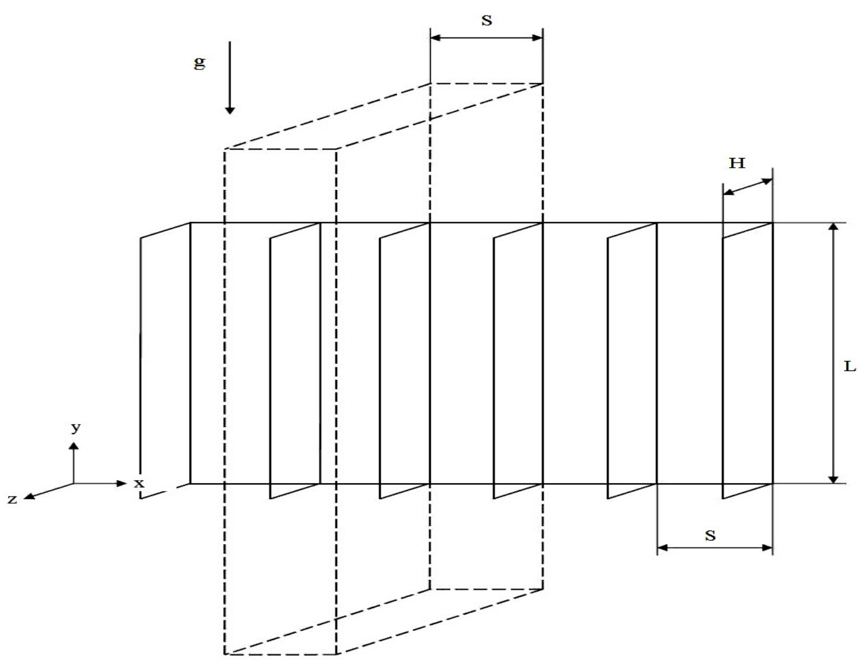



**Figure 2:** Schematic drawing of the fin array under investigation.

The boundary conditions were implemented by the inclusion of the additional source and/or sink terms in the finite volume equations for computational cells at the boundaries. In natural convection flows there is no information regarding the velocity temperature fields before the start of calculations. Since governing, equations are invariably coupled, the temperature field causes the velocity field to develop, and in turn, the velocity field affects the temperature field with the promotion of convective heat transfer. The imposed boundary conditions were as follows [9]:

1. The fin surface (ABCDA and EFGHE) and base surface (CDEHC) were held at constant temperature.
2. Symmetry boundary conditions were applied at surface IJCDKPI, PKLMP, MNOHELM and IJONI.
3. No-slip boundary conditions were applied to fin walls to simulate the effect of laminar friction and heat transfer.
4. All the remaining surfaces were open surfaces where air enters and leaves the surface at the ambient temperature T∞ and the corresponding density.

A schematic drawing of the fin array under investigation is shown in Figure (2), together with the actual simulated part of one fin. An infinite number of fins with negligible thickness was assumed. The fin surfaces and fin array base were assumed to be at a uniform temperature the computational domain for the present problem is shown in Figure (3) an infinite number of fins with negligible thickness was assumed. The computational domain has been beyond the actual dimensions of the fin array to account for effects due to the surrounding of the fin array. Computations were performed for the fin of L, H being the height of the fin, and S is (spacing) width of the domain in the x-direction. The domain length was 2L in the x-direction and the height of the domain 5H in the z-direction. The present study investigates the effects of a wide range of geometrical parameters on the heat transfer from fin arrays. Effects due to change in fin height, fin spacing, orientation angle, and temperature difference between fin and surrounding are investigated.

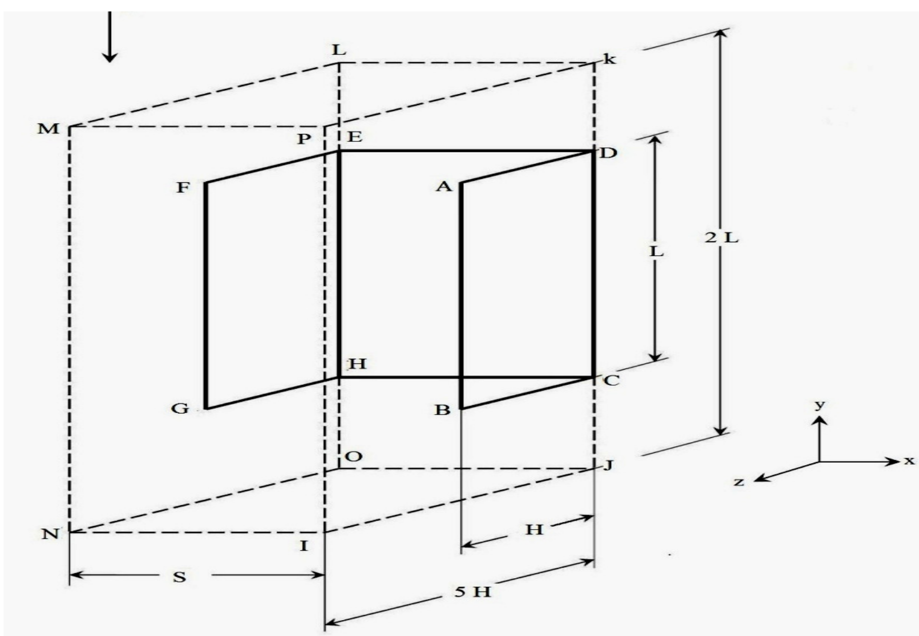



**Figure (3):** Schematic drawing of the computational domain with locations of the applied boundary conditions.

The system of Equation (1) through Equation (6) with the boundary conditions stated above is solved through a control volume to obtain a set of discretized linear algebraic equations. These equations were solved by the widely used commercial CFD package FLUENT 6 employing the SIMPLEST algorithm [24] for the pressure correlation process along with the solution procedure for the hydrodynamic equations. The central-difference-scheme leads to a second-order truncation error in the approximations, whereas the upwind-scheme gives only first-order accuracy. The discretized equations are solved by the TDMA (Tri Diagonal Matrix-Algorithm).

The solution is for conditions of steady stat laminar flow free convection heat transfer (Ra < $10^9$). Where Ra is Rayleigh number was defined as:

$$Ra = \frac{g\beta(T_F - T_\infty).S^3}{\nu.\alpha} \qquad (7)$$

where S, the spacing between fins. The physical properties necessary to evaluate Rayleigh and Nusselt numbers were taken at film temperature (Eq. 6). The heat transfer coefficient based on the area of the vertical plate and thus the Nusselt number were evaluated as:

$$Nu = \frac{h.S}{k} \qquad (8)$$

After the determination of the experimental Nusselt numbers, several correlations from literature were utilized to evaluate and compare the Nusselt numbers as given in Table (1).

## 4. RESULTS AND COMPARISONS

In this section, the comparison between numerical and experimental results of rectangular fins without shroud parameters was got from experimental [9] and theoretical [10] works, and we'll compare these results with the similar available international works and data.

### 4.1 Numerical Results

This analysis for the rectangular Fins without shroud, where the fin's height, orientation angel and temperature difference between fin and surroundings on the free convection heat transfer from horizontal fin arrays was carried out .The three-dimensional elliptic governing equations were solved using a finite volume based computational fluid dynamics (CFD) code.

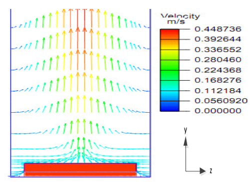
It is important to understand the general flow patterns dominating flows from fin arrays. In natural convection flow, there is no information regarding the velocity and temperature field, the temperature field causes the velocity field to develop and in turn, the velocity field affects the temperature field with the promotion of convection heat transfer. Figure (4 a and b) presents the velocity vector and thermal fields for fin arrays of fin spacing (17mm), and heat input to the base of the fin arrays is equal to 75W in the y–z plane at mid plane in x coordinates for fin heights equal to 20 mm and 60mm, respectively.

**Table (1)**:
Nusselt Number as a function of Rayleigh number

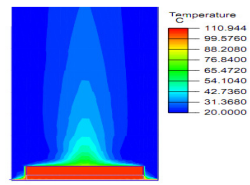
| Author | Nu | No, | Ref. |
|---|---|---|---|
| Churchill and Chu's<br>$10^{-1} < Ra < 10^{12}$<br>laminar and turbulent flows | $\left[0.825 + \frac{0.378 \times (Ra)^{\frac{1}{6}}}{\left[1+\left(\frac{0.492}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{8}{27}}}\right]^2$ | (9) | [13] |

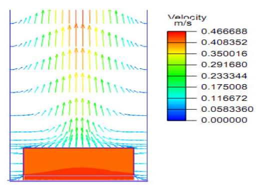
| | | | |
|---|---|---|---|
| Churchill and Chu's<br>$10^{-1} < Ra < 10^{9}$<br>laminar flow | $0.680+\frac{0.670\times(Ra)^{\frac{1}{4}}}{\left[1+\left(\frac{0.492}{Pr}\right)^{\frac{9}{16}}\right]^{\frac{4}{9}}}$ | (10) | [13] |
| McAdams' relation<br>$10^{4} < Ra < 10^{9}$ | $0.59\times(Ra)^{\frac{1}{4}}$ | (11) | [13] |
| Churchill and Usagi's<br>$10^{5} < Ra < 10^{9}$ | $\left(0.670\times(Ra)^{0.25}\right) \Big/ \left(\left[1+\left(\frac{0.492}{Pr}\right)^{0.5625}\right]^{0.4444}\right)$ | (12) | [13] |



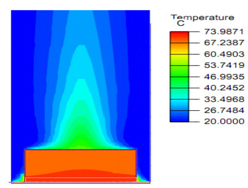

(a): Velocity Vectors




(a): isothermal contours




(b): Velocity Vectors




(b): Isothermal Contours

**Figure 5:** Velocity vectors and isothermal contours field in x-y plane at mid-plane in z coordinate.

Figure (5) indicates the variation of heat flux along the fin length. From this distribution, it is clear that the rate of heat dissipation from the central portion is less. This justifies where single chimney flow pattern is present, a stagnant zone is created at the central bottom portion of fin array channel and hence it does not contribute much in heat dissipation.

### 4.1.1. Effect of Temperature Difference

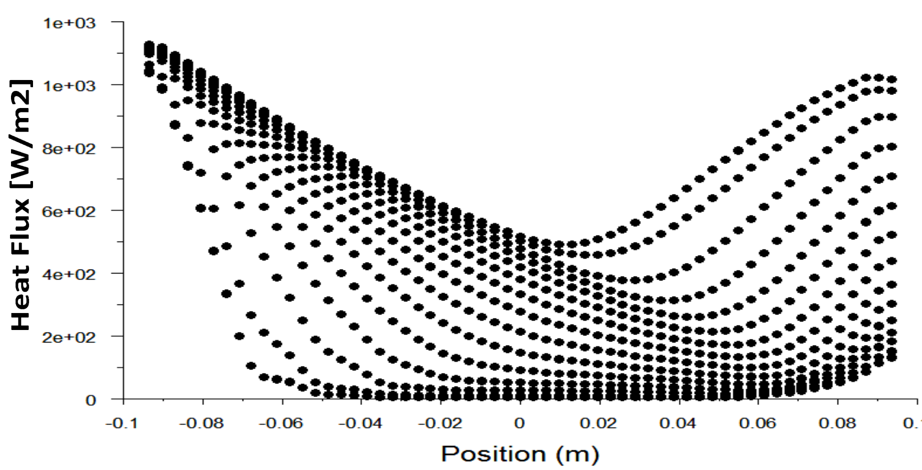



**Figure (5):** Variation of Heat Flux along the Fin Length.

Figure (6) shows the average heat transfer coefficient as a function of temperature differences ($\Delta T = T_F - T_\infty$), orientation angle, fin spacing, and fin height. From this figure, it is seen that the average heat transfer coefficient increases with increasing ΔT for all values of orientation angle. It is clear also; the heat transfer coefficient increases with decreasing orientation angle.

### 4.1.2. Effect of Fin Spacing

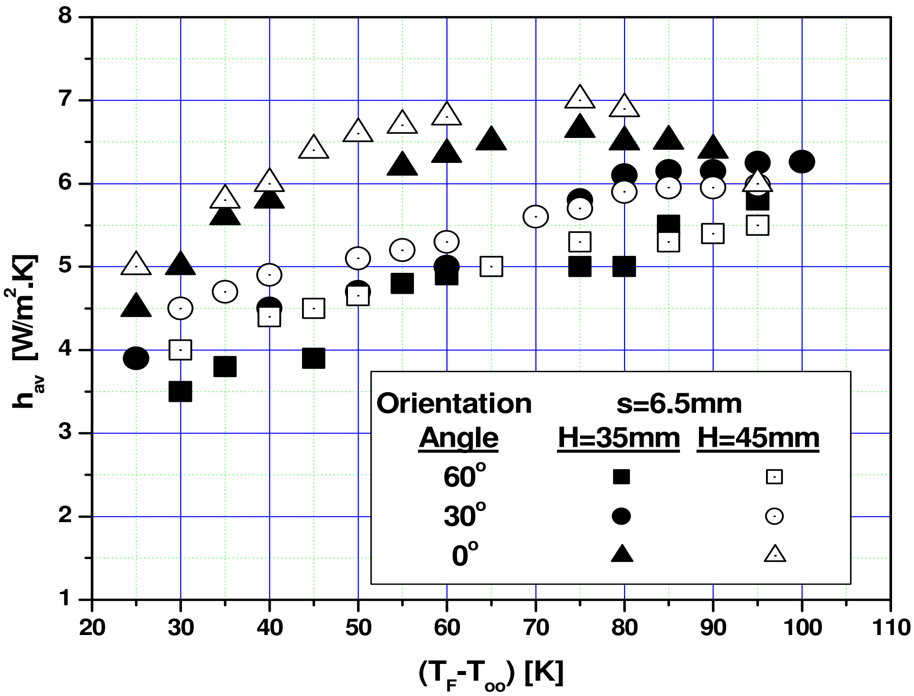



**Figure (6):** The Average Heat Transfer Coefficient as a Function of Temperature Differences, Orientation Angle, Fin Spacing, and Fin Height.

The variations of average heat transfer coefficient ($h_{av}$) as a function of fin spacing at different values of heat input to the base of the fin arrays for rectangular fin arrays without shroud at different values of fin heights (=20, 40, and 60mm) are shown in Figure (7). It is seen that the average heat transfer coefficient increases with increasing in fin spacing for all fin heights values and heat input to the base of the fin arrays (=25, 75 and 125W) until fin spacing reaches to 25mm, after that the average heat transfer coefficient value goes to constant value with slightly variation.

### 4.1.3. Effect of Heat Rate Input

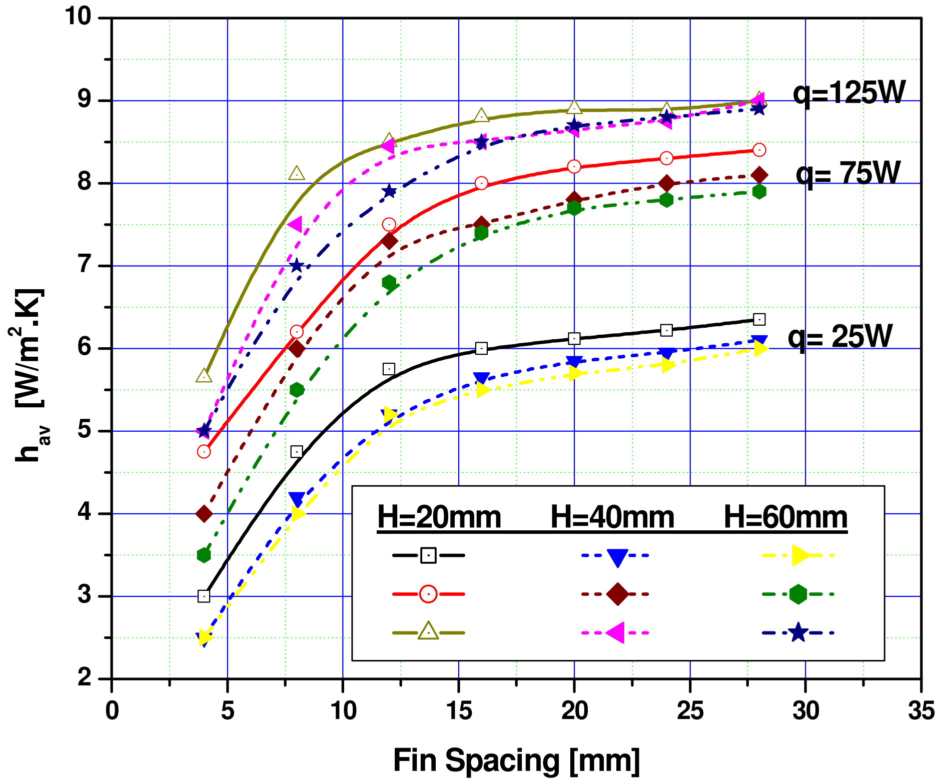



Figure (7): Variations of average heat transfer coefficient as a function of fin spacing, heat rate input and fin height.

The variation of the average heat transfer coefficient with heat rate input as a function of fin height and spacing is shown in Figure (8). This figure shows that the average heat transfer coefficient increases with increasing of heat input and decreases with increasing of fin height. The reasons for these variations are due to the change in the thermal boundary layers and the temperature distribution through the fin channel. The increase of the fin spacing significantly decreases the magnitude of air temperature in the channel and provides a more efficient heat transfer rate.

## 4.2 Experimental Results

Several experimental works were done to examine the performance and characteristics of rectangular fin-type without shroud parameters (fin height, fin spacing, fin orientation angle, fin temperature difference, heat input rate, and so on..) [9], but in this paper, we'll select some of them for to compare them with the similar of theoretical [10] and international results.

### 4.2.1. Effect of Fin Spacing

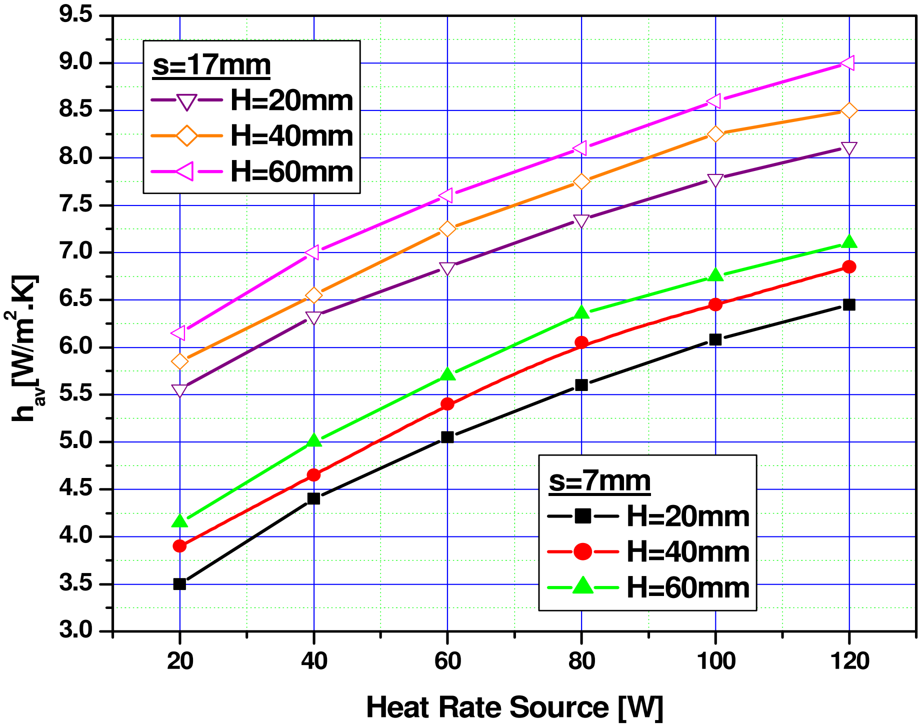



**Figure (8)** Variations of average heat transfer coefficient as a function of heat rate input, Fin height and fin spacing.

Figure (9) presents the convection heat transfer rates are plotted as a function of fin spacing and at various orientation angles. It is seen that the heat transfer rate per unit area of fin base increases with increasing the fin spacing until it reaches a maximum value and then decreases with a further increase in the fin spacing.

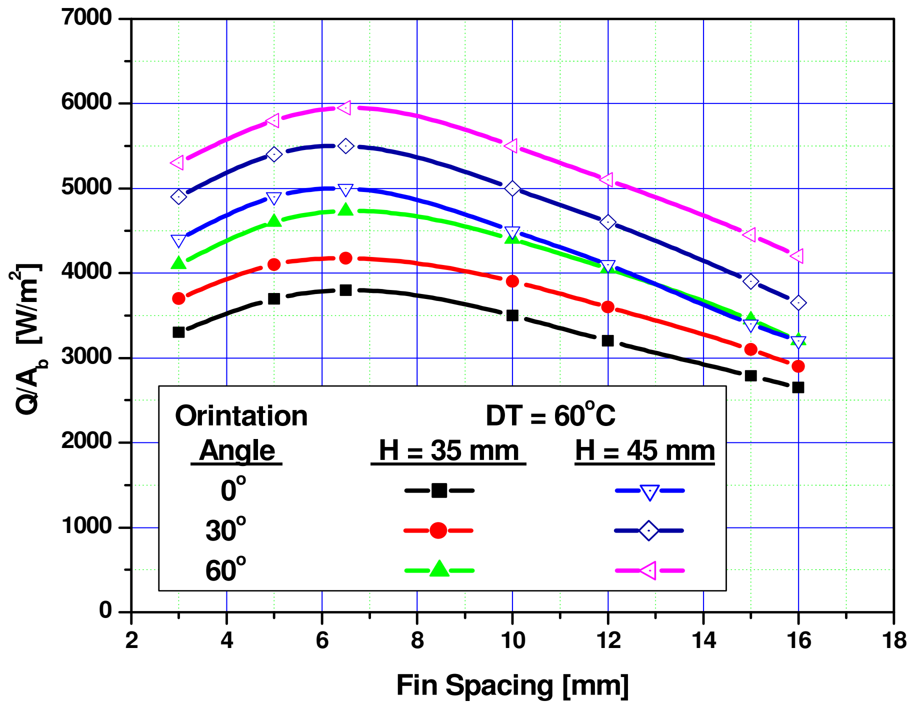



**Figure (9).** Convection Heat Transfer Rate as a Function of Fin Spacing, Orientation Angles and Fin Height at ΔT = 60°C.

When the convection heat transfer rate is maximized at certain space between fins. This space is called the optimum fin spacing, ($s_{opt}$) and from the Figure (10) the optimum space is equal to 6.5mm, and it depends on fin height and base to ambient temperature difference. As seen from this figure, the optimum fin spacing varies significantly with neither fin height nor base to ambient temperature difference. Therefore, for practical purposes, the optimum fin spacing to maximize the convection heat transfer rate may be taken as 7mm for arrays with fin heights from 15 to 45 mm and fin length equal to 187mm [9], for base to ambient temperature differences from 30 to 100 K.

### 4.2.2. Effect of Temperature Difference

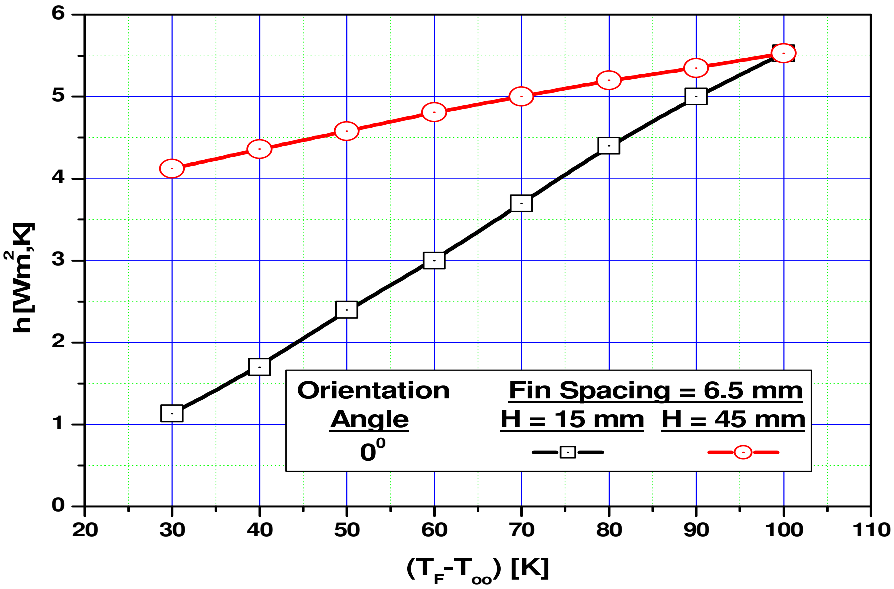



**Figure (10):** Heat Transfer Coefficient as a Function of Temperature Difference and Fin Height at S= 6.5mm and L=187mm.

The heat transfer rate per unit base area increases with the increase in the fin spacing to reach a maximum value and then decreases with farther increase in the fin spacing. the maximum heat dissipation occurs at optimal spacing ($S_{opt}$ = 6.5mm) [9].

As a point of departure for the presentation of convective heat transfer coefficient (h) values are plotted to ambient temperature difference ($T_F - T_\infty$) at different values of orientation angle ($\beta$) and fin height (H) as shown in Figure (10).

### 4.2.3. Effect of Heat Rate Input

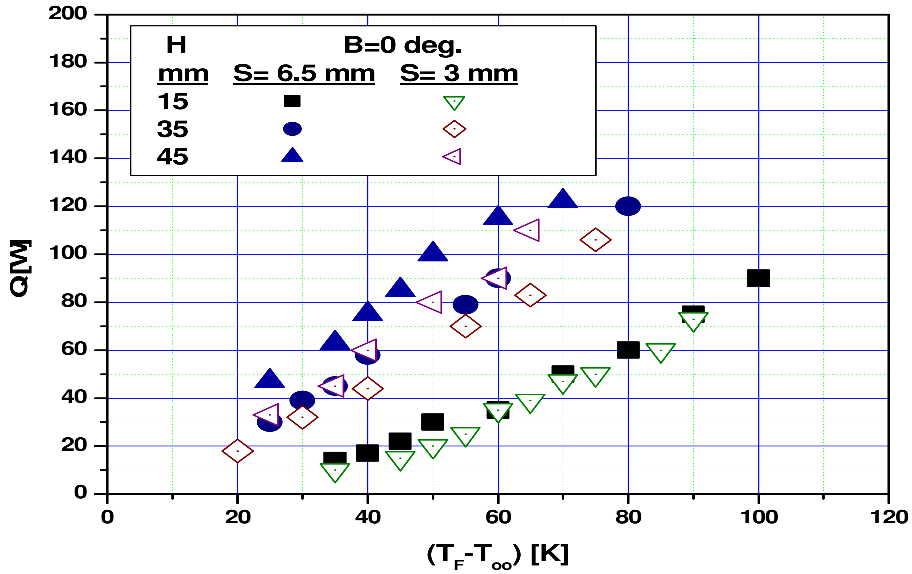



**Figure (11):** Convective Heat Transfer Rate as a Function of Fin Height and Spacing at L=187mm.

The convective heat transfer rates from fin arrays are plotted as a function of the base-to ambient temperature difference for fin spacing, s =3 and 6.5mm at fin length equal to 187mm. Figure (11) shows convective heat transfer rate as a function of fin height and spacing at L=187mm. For all fin arrays, this figure shows that the convective heat transfer rate depends on fin height, fin length, fin spacing, and base-to ambient temperature difference. The convective heat transfer rates from the fin arrays increase with fin height, fin spacing to ambient temperature difference. The curves in this figure show similar trends for identically spaced fin arrays.

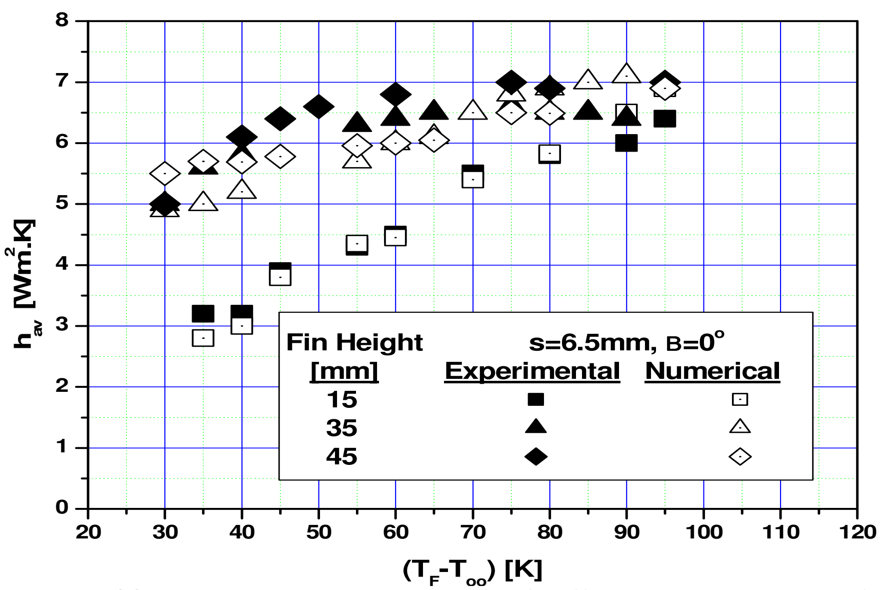



**Figure (12):** The Average Heat Transfer Coefficient as a Function of Temperature Difference, Fin Height, Fin Spacing and Orientation Angle.

Figure (12) shows the convective heat transfer rates measured from three different fin heights. The heat transfer rate increases monotonously with temperature difference between fin base and surroundings, ($T_F - T_\infty$). However, in fin arrays with small fin height the rate of increase of heat transfer rate with temperature difference is smaller than those ones with large fin height

**4.3 Comparison of Results**

The comparisons between experimental and theoretical (numerical) results are presented in the following figures.

Figure (13) presents the experimental and numerical computational fluid dynamics (CFD) for average heat transfer coefficient as a function of temperature difference, fin height, fin spacing at orientation angle equal to zero degree (horizontal). From this figure, it is seen that a good agreement between experimental and numerical results with a maximum deviation of 0.91% for fin has height of 15 mm and orientation angle of 60 degree. Table 2 shows a deviation of the experimental results from numerical results at s = 6.5mm, ΔT = 35°C to 95°C and at orientation angle equal to $0^o$ (horizontal).

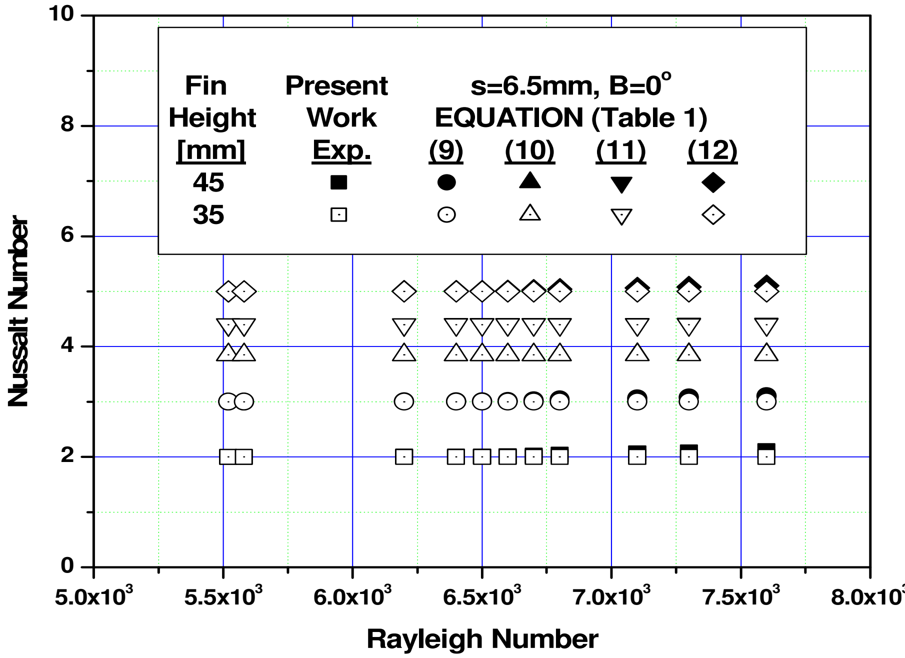



Figure (13): Comparison of Experimental Results with the Same Results of International Correlations from Literatures (Table 1).

The Nussalt number correlations in Table (1) were also compared with the experimental data as a function of Rayleigh number, as displayed in Figure (11). These results reveal that except for the McAdam's correlation, which is a very rough correlation, the experimental data are in excellent agreement with the results from literature. For McAdam's correlation, the average relative error is 3.5%, while for the others it is less than 3%. This agreement confirms the validity of the experimental set-up, the experimental procedure and the calibration procedure [13].

## 5. RESULTS AND DESCUSIONS

A correlation for average Nusselt number has been presented to relate the heat transfer from fin arrays in channel with dimensionless, Rayleigh number, modified experimental parameters such as fin spacing, fin height, fin length and orientation angle.

The new empirical correlation (modified of McAdam's correlation) was derived to correlate the mean Nusselt number as a function of Rayleigh number as follows:

$$Nu = 0.52 \times Ra^{0.16}$$

This equation is valid for free natural heat transfer convection in tested finned model with thickness t = 6.5mm, fin height (=15 to 45mm), fin spacing (=3 to 16mm) and Rayleigh number ranged from $5.0 \times 10^3$ to $1.0 \times 10^4$ at constant fin length and orientation angle equal to zero.

## 6. CONCLUSIONS

This work presents a comparison of experimental and numerical results of free convection heat transfer from rectangular fins array, and it is mounted on a vertical base at a different orientation angle, is equal to zero (horizontal). From these results, it can be concluded that the geometric parameters of fin array, fin height, fin length, and fin spacing, and base-to-ambient temperature difference affects the rate of convection heat transfer. Finally, and from these comparisons, we conclude the following:

1. The new empirical correlation (modified of McAdam's correlation) was derived to correlate the mean Nusselt number as a function of the Rayleigh number.
2. The average heat transfer coefficient has a maximum value at orientation angle equal to zero degrees and It decreases with increasing of orientation angle.
3. The heat transfer rate per unit base area increases with the increase in the fin spacing until it reaches a maximum value, and then decreases with further increasing of fin spacing. It reaches the maximum value at optimal spacing value (=6.5mm).
4. Good agreement was obtained among the analytical, numerical and experimental results.

## NOMENCLATURE

| | | | | |
|---|---|---|---|---|
| A | Total heat transfer area )$m^2$( | | t | fin thickness, m |
| CFD | Computational fluid dynamics | | T | Temperature, k |
| g | gravitational acceleration, $m/s^2$ | | $T_s$ | Temperature of the surface, k |
| h | convection heat transfer coefficient, $w/m^2.k$ | | x,y,z | rectangular coordinates, m |
| H | fin height, m | | **Greek Letters** | |
| k | thermal conductivity, w/m.k ; Boltzmann's constant | | $\alpha$ | thermal diffusivity, $m^2/s$ |
| L | based on characteristic length ; fin length, m | | $\varepsilon$ | Emissivity |
| N | Number of temperature increments in a flux plot | | $\varepsilon_f$ | fin effectiveness |
| Nu | Nusselt Number | | $\eta_f$ | fin efficiency |
| $L_c$ | characteristic length, m | | $\eta_o$ | fin temperature effectiveness |
| q | Heat transfer rate, w | | $\theta$ | Temperature deference, k |
| $\dot{q}$ | power input to the heater, w | | $\mu$ | viscosity, kg/m.s |
| q" | Heat flux, $w/m^2$ | | $\nu$ | Kinematic viscosity, $m^2/s$ |
| Ra | Rayleigh Number | | $\rho$ | mass density, $kg/m^3$ |

| | | | |
|---|---|---|---|
| $R_t$ | thermal resistance, k/w | σ | Stefan-Boltzmann constant, $w/m^2.k^4$ |
| $R_{t,f}$ | fin thermal resistance | Δ | Difference |
| $R_{t,o}$ | Thermal resistance of fin array, k/w | β | volumetric thermal expansion coefficient, $k^{-1}$ ; orientation angel, degree |
| S | shape factor of two-dimensional conduction, m ; fin spacing, m | | |

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