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An Experimental Study of Natural Convection Heat Transfer from a Horizontal and Slightly Inclined Plate-Fin Heat Sink

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Abstract

The steady-state natural convection from heat sink fin arrays was studied on horizontal, vertical, and inclined heat sinks. Under natural convection, horizontal and vertical heat sinks with parallel fins were tested by considering radiation heat transfer. The experiments were conducted with power inputs ranging from 60 W to 455 W in order to obtain different temperatures. According to the result, when heat input increases, the heat transfer coefficient increases by 38 %, 40.78 % for horizontal and vertical respectively. For horizontal and vertical cases, new correlations have been presented to calculate the Nusselt number influenced by the Rayleigh number. For the incline case, the effect of buoyancy force was studied by changing the inclination angles at 0, 30, 45, and 60 degrees from vertical position. According to comparisons between vertical and incline cases, Nusselt number and heat transfer coefficient were most improved at 30 degrees by 6 %.

Keywords: Natural convection, Heat sink, Plate-fin, Electronics cooling.

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1. Introduction

The most common mechanisms in electronic cooling to dissipate heat generated are natural convection and radiation heat transfer. For this purpose, heat sinks with extended surfaces are used in several practices such as radio frequency power transistors, central processing units (CPUs), and light-emitting diodes (LEDs). These mechanisms are also used to reduce operating temperatures, maintain system stability, and prolong the life of many machines, including motors/generators and automobiles. A natural convection mode is used to reduce mechanical failure and noise.

1.1. Configuration of the heat sink

The most common heat sink configurations are horizontal base plate with vertical fin channels, vertical base plate with vertical fin channels, and inclination for each as shown in Fig. 1. A vertical base plate with vertical fin channels provides better natural cooling performance where air enters from the bottom of the fin channels to the fin tips [1].

1.2. Literature review

From the literature, many researchers have improved the thermal performance of heat sinks by improving the fin arrays with different geometry and orientations. Elenbaas [2] used parallel plates in his experiments to investigate the effect of natural convection on heat dissipation. Starner and McManus [3] conducted experiments to compute the average heat transfer coefficient for four different rectangular fin arrays and three different orientations, including horizontal, 45 degrees, and vertical. Harahap and McManus [4] are studied experimentally on the fin arrays with the base oriented horizontally. In their investigation, eight different fin heights and spacing were studied to elevate that the fin height is the most important geometric parameter. Another experimental study is carried out by Aihara [5] to investigate the temperature distribution in the boundary layer and evaluated the effect of geometric parameters on the thickness of the boundary layer. As it results, a correlation was suggested for the average Nusselt number of the parallel plate finned heat sink.

Jones and Smith [6] performed another experimental research on the natural convection of fins. Temperature gradients in the fins were observed in the experimental investigation using the interferometer technique. As a result, they propose two empirical correlations for optimizing fin spacing. Fujii and Imura [7] studied the heat transfer from a plate with arbitrary inclination under natural convection experimentally. The Nusselt number was found to be proportional to one-fifth of the *Ra* number, and a slightly inclined heated plate and a horizontal heated plate, both facing downwards, were proportional to one-fifth of the *Ra* number. It was also when the inclination of the angle of the fins was greater than the inclined heated plate faced upwards. Under natural convection conditions, Ko and Leung [8] studied experimentally the heat dissipation from an array of stainless-steel vertical rectangular fins to estimate the effect of fin length, fin spacing, and temperature difference between the heat sinks fin array and their surroundings. Additionally, they present the optimal fin spacing for a vertically based finned system when the excess temperature changes from 20 to 40 K.

Mittelman et al. [9] studied the behavior of laminar free convection under a hot isothermal and inclined fin array using an experimental and numerical investigation. The effect of incline on the place where the flow stagnates and splits was evaluated. In addition, the heat transfer rates were estimated for various fin array shapes and temperatures. The results indicate that for small inclination angles, the cooling rate is practically constant. According to their study, they found that the array's optimum fin spacing is independent of the inclination angle. Additionally, a semi-empirical model for the heat transfer coefficient of horizontal and slightly inclined arrays with high fin spacing was developed in the theoretical part.

Huang et al. [10] investigated the effect of inclination on natural convection heat transfer from square pin fin heat sinks. Seven square fin heat sinks and a flat base plate with different configurations are tested in a controlled environment. The results show that the downward-facing arrangement has the lowest heat transfer coefficient due to flow obstruction. The thermal performance of the upward and sideward arrangements is comparable. The fining factor is important in enhancing thermal performance. When the factor is less than 2.7, the sideward arrangement outperforms the upward arrangement, and when the factor is above 2.7, it indicates a revered trend. In the case of a sideward arrangement, a greater increase in the finning factor approaches its performance to match that of a downward arrangement.

Naserian et al. [11] investigated the natural convective heat transfer from V-type fin arrays on a vertical base both experimentally and numerically. In this study, fluent software is used to run the CFD simulations. Additionally, the experiment was carried out with varying fin shapes, fin spacing, and fin numbers. The investigation was extended to study the effect of row spacing on the fin. The results show that increasing the number of fins on a vertical surface enhances thermal performance as long as the boundary layer thickness on the fin surface is small.

Ismael et al. [12] investigated numerically the natural convection heat transfer in porous cavities with arc-shaped walls. For investigation, a thermal insulation is used to insulate the horizontal walls of the cavity, while a constant temperature is maintained at the right arc shape wall (T_h), and a constant temperature is maintained at the left arc shape wall (T_c). In addition, this investigation uses Flex PDE software to solve equations governing heat transfer and nanofluid flow, as well as temperature-independent models for nanofluid properties. There were three parameters investigated: nanoparticle volume fraction, Rayleigh number, and arc center. Detailed analysis of the results is shown through contours, isotherms, and the average Nusselt number. It has been found that the average Nusselt number decreases as the number of arc center increases, whereas the Ra number increases with the arc center.

Lee et al. [13] investigated the cooling performance of electronic devices using horizontal tubes with inclined rectangular fins. According to the researchers, inclined rectangular fin cooling performance with a 60° inclination angle is 6 % greater than solid rectangular fins. Additionally, the optimal cooling performance of the tilted finned tube is 9.2 times greater than that of the non-finned tube. As a result, inclined rectangular fins could be used to cool a variety of electrical devices.

Altun and Ziylan [14] carried out experimental research on natural convection heat transfer in horizontally oriented, vertical sinusoidal wavy fins involving radiation heat transfer. Experiments were carried out with wavy fins of three different amplitudes, namely H/30, H/15, and H/10, and with various heater power inputs. The results were compared to those of a rectangular finned plate and a horizontal reference base plate. Heat transfer enhancement is better with wavy fins than with rectangular fins, according to the results.

Hou et al. [15] conduct a numerical investigation of the steady-state natural convection of radiators with parallel configurations of rectangular cross-section fins on vertical trusses. In this study, the verified model is used for downward inclination by changing the direction of gravity acceleration. Additionally, it was found that for the slice layout structure, an inclination from the vertical does not reduce the convection heat transfer rate. However, a significant increase was obtained in the brick layout example, due mostly to flow separation inside the heat sink's fin channels.

Hadi et al. [16] investigated experimentally the effect of coating the heat sink with graphite and graphene on heat dissipation. This investigation involves three aluminum specimens, one coated with graphite, another with graphene, and the third left uncoated for comparison. For each specimen, an electric plate heater was used to simulate a heat sink's output in a homemade wind tunnel. The study used a heat sink with a high thermal conductivity between the heating plate and heat sink base plate to reduce heat flow resistance. As a result of the experiments, graphene-coated samples have the greatest thermal dissipation, whereas uncoated samples have the least.

1.3. The aims of the study

In this study, convection and radiation heat transfer from heat sink fin arrays was investigated for both horizontal and incline cases. In order to evaluate buoyancy force effects on the heat transfer rate from heat sink fin arrays, the change in the direction of gravitational acceleration was done.

2. Experimental Setup

2.1. General description

In these experiments, the physical causes of the heat sink fin arrays with constant fin geometry were used to evaluate the rate of heat transfer by natural convection, as shown in Fig. 1.

In the setup, the heat sink and the fin arrays were fabricated from aluminum marital due to their ease of machining, high thermal conductivity, and durability. The heat sink with fin arrays has the dimensions $L \times W \times H$, where L = 600 mm is the length, W = 100 mm is the width, and H = 60 mm is the height, as shown in Fig. 2. The experiments include a digital multimeter for measuring the baseplate temperature of the rectangular fin array, as well as a heater, thermocouples, selector switch, multi-meter, volt-meter, and adjustable stand to change the inclination and vertiac as shown in Fig. 3.

2.2. Experimental procedure

In the experiment, a variac (Voltage-controlled AC power supplies) was used to supply different values of heat fluxes by adjusting the power input voltage, as shown in Fig. 4. The surface temperatures of the base plate are measured using the average twenty-four thermocouples (type K NiCr-Ni) with an accuracy of ± 0.1 °C. These thermocouples are embedded in the bottom of the heat sink and in an aluminum plate of 6 mm thickness, as shown in Fig. 5. In addition, all thermocouples are wired and connected to a selector switch that is connected to a digital multimeter, as shown in Fig. 6. One more thermocouple for measuring ambient temperature was located near the rectangular fin array. For experimentation, heat inputs of 60, 115, 175, 250, and 455 W are investigated. For inclination experiments, an adjustable stand was used to indicate angles, 0°, 30°, 45°, 60°, and 90° from vertical as shown in Fig. 1. The steady-state conditions for all current work experiments are attained in about 90 to 120 minutes, depending on the environment and electrical power conditions. Whenever the steady-state condition is achieved, all-temperature thermocouple readings and power input rates are registered.

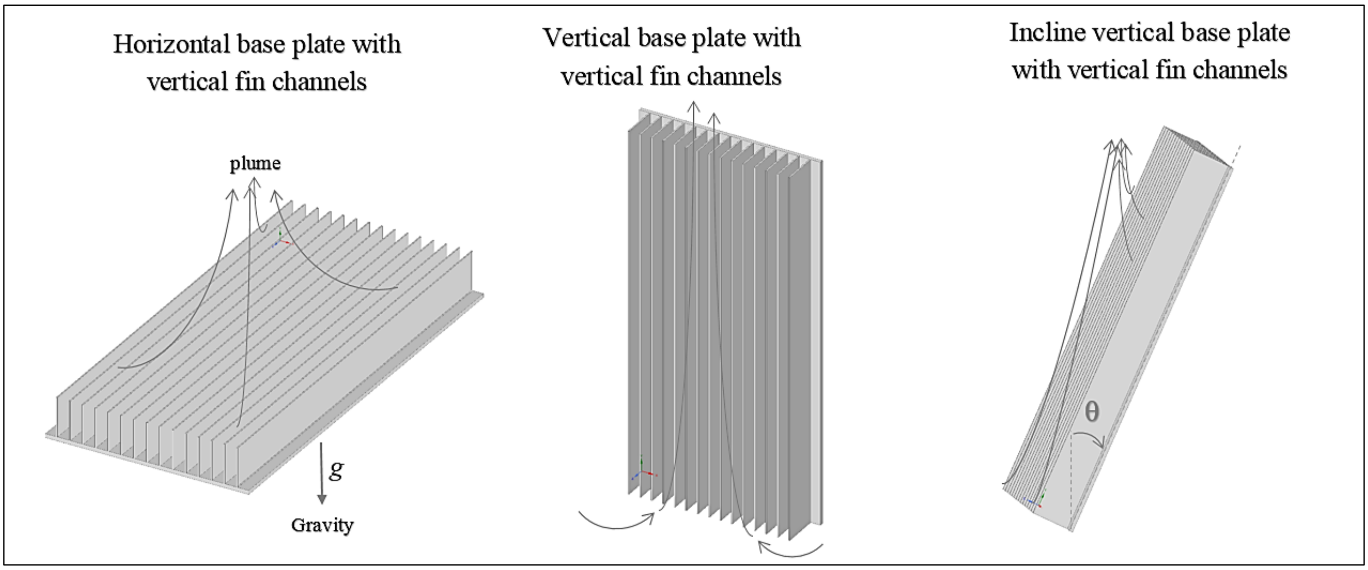

Fig. 1 Fin configurations.

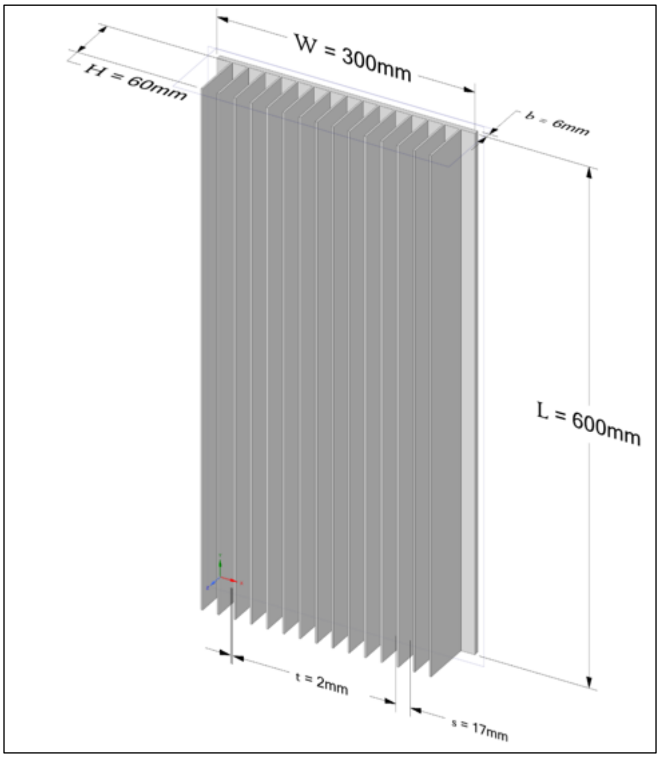

Fig. 2 Fin array geometry.

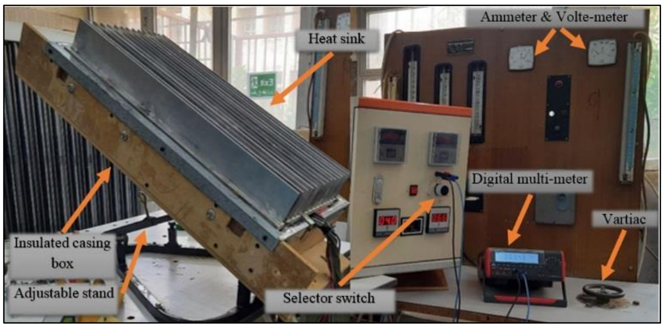

Fig. 3 Photographic view of the test section with several instruments.

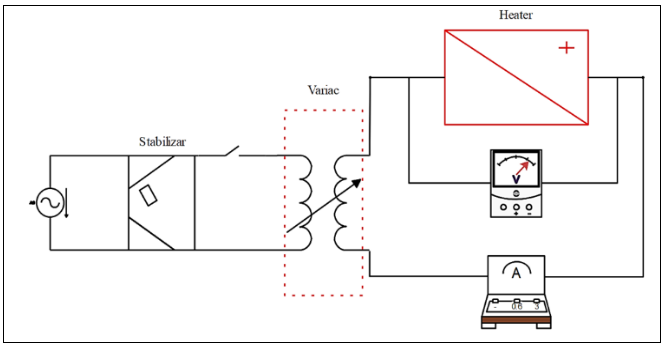

Fig. 4 Electrical circuit of the heater.

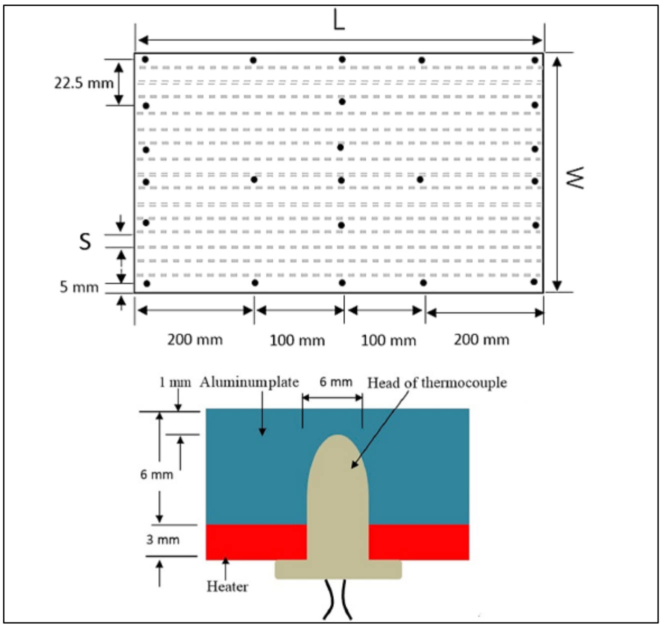

Fig. 5 Locations of the thermocouples.

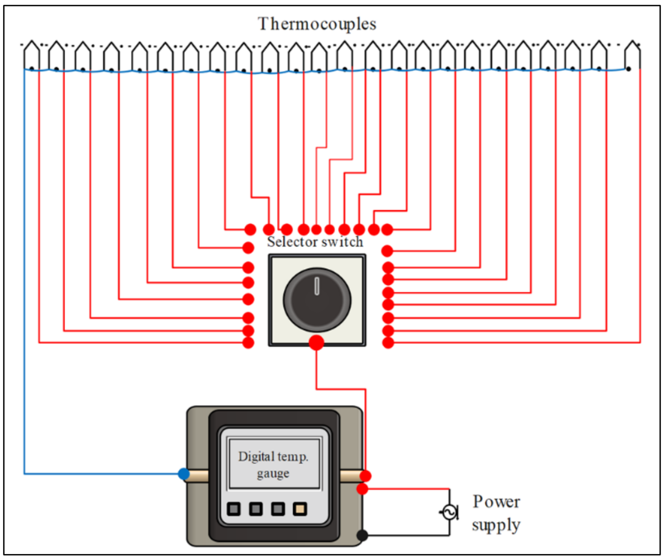

Fig. 6 Electrical circuit of thermocouples.

2.3. Heat transfer analysis

The analysis of these experiments was done for a base plate with straight fins at horizontal and vertical cases with various inclination angles. For these experiments the heat transfer from the rectangular fin array surface of area (A_t) to the surrounding is by convection and radiation, assuming that the conduction heat loss is negligible through the insulation sides for the base plates of the fin array [17]. Therefore, the total heat transfer from the surfaces may be written as:

$$Q_{tot} = Q_{conv} + Q_{rad} \tag{1}$$

Where Q_{tot} is equal electric power input and it can observe by multiplying voltage (V) by current (I) that measured of the heaters.

$$Q_{tot} = V \,.\, I \tag{1}$$

According to Newton's equation of cooling, the convection heat transfer from the fin array can be calculated as follows [18]:

$$Q_{conv} = h_{av}\, A_s\, (T_b - T_\infty) \tag{2}$$

Hence, average convection heat transfer coefficient h_{av} is:

$$h_{av} = \frac{Q_{conv}}{A_s (T_b - T_\infty)} \tag{4}$$

Hence, A_s represent to total surface area of the rectangular fin arrays:

$$A_s = N[2HL + SL + (2H + L)t] \tag{5}$$

In addition, T_b is the average temperature of the heat sink:

$$T_b = \sum_{i=1}^{p} \frac{T_S}{p} \tag{6}$$

T_∞ is the ambient temperature of the surrounding air.

The average Nusselt number can be calculated as:

$$Nu_s = \frac{Q_{conv}\, S}{A_s\, k(T_b - T_\infty)} \tag{7}$$

where S is the fin spacing, k is the thermal conductivity of the air.

The independent variable of the test is the temperature difference, T_b - T_∞ and is represented by the Rayleigh number.

$$Ra_s = \frac{g\, \beta (T_b - T_\infty) S^3\, Pr}{\nu^2} = Gr_s\, Pr \tag{8}$$

At incline, the gravitational acceleration (g) replaced by (g cos θ) for ($0 < \theta < 60°$), and β is the coefficient of thermal expansion and may be calculated as follows:

$$\beta = \frac{1}{T_f} \tag{9}$$

All thermal properties of air are evaluated at the film temperature, which is defined as:

$$T_f = \frac{T_b + T_\infty}{2} \tag{10}$$

For natural convection the radiation heat transfer effect must be taken into consideration for high accuracy. The radiation heat transfer for surfaces can be calculate as [17, 18].

$$Q_{rad} = F(\varepsilon)\, F_{c\text{-}sur}\, \sigma\, A_t \left(T_b^4 - T_\infty^4\right) \tag{11}$$

Where σ is the Stenfan-Boltzmann constant $\left(5.67 \times 10^{-8}\ \mathrm{W/m^2\ K^4}\right)$, ε is the emissivity of the surface, A_t is the total surface area for heat transfer, T_b and T_∞ are the temperature of fins and the ambient respectively, and $F_{c\text{-}sur}$ is the view factor.

Form literature heat sink with fin arrays, the fin can transport heat to the ambient and other fins; hence the view factor necessary to be determined using view factor methods [19]. The view factor equations of radiation can be used to determine a view factor for a composite of fins and un-finned areas, or it can be requested from the view factor catalog in [20]. The view factor between the fin side surfaces is $F_{S\text{-}s}$ the view factor between the fin side surfaces and their surroundings is $F_{s\text{-}sur}$, and the view factor between the fin tip and the surroundings is $F_{t\text{-}sur}$ will radiate heat at full view factor of (1).

Here the equations in [20] can be used to calculate $F_{S\text{-}s}$ and $F_{b\text{-}s}$ in this case:

$$F_{S\text{-}s} = \frac{2}{\pi\, m\, n} \left\{ \begin{array}{l} ln\left[\dfrac{(1+m^2)(1+n^2)}{1+m^2+n^2}\right]^{\frac{1}{2}} + m(1+n^2)^{\frac{1}{2}}\, arctan\dfrac{m}{(1+n^2)^{\frac{1}{2}}} \\ + n(1+m^2)^{\frac{1}{2}}\, arctan\dfrac{n}{(1+m^2)^{\frac{1}{2}}} - m\, arctan\, m - n\, arctan\, n \end{array} \right\} \tag{12}$$

Where, $m = L/Sn = H/S$

$$F_{b\text{-}s} = \frac{1}{\pi n}\left\{ \begin{array}{l} n \arctan\frac{1}{n} + m \arctan\frac{1}{m} - (m^2+n^2)^{1/2} \arctan\frac{1}{(m^2+n^2)^{1/2}} \\ + \frac{1}{4} \ln \left\{ \begin{array}{c} \frac{(1+n^2)(1+m^2)}{1+m^2+n^2} \left[\frac{n^2(1+m^2+n^2)}{(1+n^2)(1+m^2)}\right]^{n^2} \times \\ \left[\frac{m^2(1+m^2+n^2)}{(1+n^2)(1+m^2)}\right]^{m^2} \end{array} \right\} \end{array} \right\} \quad (13)$$

Where, $m = H/Ln = (S/2)/L$

Other view factors can be calculated using the summation rule and reciprocity.

$$F_{b\text{- }sur} = 1 - 2F_{b\text{-}s} \quad (14)$$

$$F_{s\text{-}sur} = 1 - \frac{A_{base}}{A_{side}} F_{b\text{-}s} - F_{s\text{-}s} \quad (15)$$

$$F_{t\text{-}sur} = 1 \quad (16)$$

The view factors for the overall channel and the surrounding $F_{c\text{-}sur}$ are as described in the following:

$$F_{c\text{-}sur} = \frac{2A_{side} F_{s\text{-}sur} + A_{base} F_{b\text{-}sur} + A_{tip} F_{t\text{-}sur}}{2A_{side} + A_{base} + A_{tip}} \quad (17)$$

Therefore, the heat sink overall, radiative heat transfer rate can be calculated.

$$Q_{rad} = N\left(2A_{side} + A_{base} + A_{tip}\right)\sigma\, \varepsilon\, F_{c\text{-}sur}\,(T_w - T_\infty) \quad (18)$$

Or

$$Q_{rad} = N[2HL + SL + (2H + L)S]\sigma\, \varepsilon\, F_{c\text{-}sur}\,(T_w - T_\infty) \quad (19)$$

The value of the view factor using the upper equations was (0.26668) and the radiation heat transfer rate was found to be less than 5 % of the total heat input for all experiments. This is due to the small emissivity of aluminum sheet commercial 0.09.

3. Experimental result and validation

3.1. Horizontal case

The experimental study was conducted on rectangular fin arrays with laminar natural convection on a horizontal base with vertical fin plates as shown in Fig. 1.

3.1.1. Validation the results

For the purpose of confirming the experimental procedure, the results were compared to empirical correlation from the literature [6] used for horizontal heat sinks with vertical fins to confirm that the experimental method was effective. As followers, this correlation relation is considered S as characteristic lengths.

Jones and Smith, 1970 [6],

$$Nu_S = 6.7 \times 10^{-4}\, Gr_S\, Pr \left[1 - e^{-\left(0.746 \times \frac{10^4}{Gr_S\, Pr}\right)^{0.44}}\right]^{1.7} \quad (20)$$

where, $Gr_S\, Pr = \frac{g\, \beta (T_s - T_a) S^3}{\nu^2} Pr = Ra_s$

Figure 7 shown the correlations also plotted along with experimental results. As a result, the experimental results agree with the Jones and Smith correlation [6], which used S as the characteristic length with a maximum error of 3 % for all experiment results. Additionally, the plot shows the variation Nusselt number as a function of the Rayleigh number. It can be seen that an increase in the Rayleigh number increases the Nusselt number increase. Due to the increasing temperature difference between the fin array and the surroundings, more heat is dissipated by natural convection.

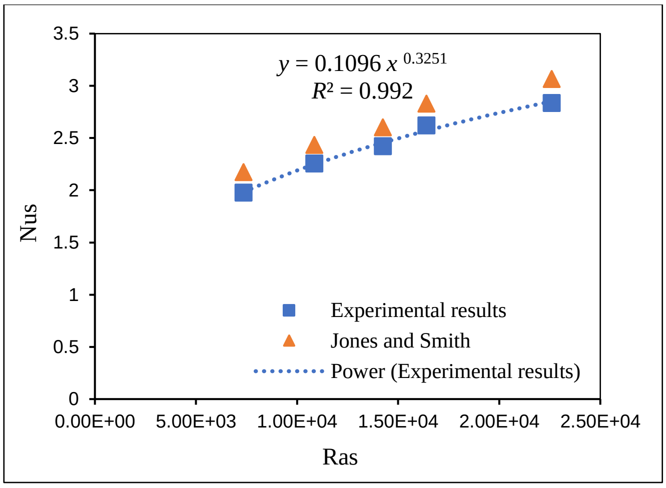

Fig. 7 Validation of the experimental results for horizontal heat sink with vertical fins.

3.1.2. Average Nusselt number correlation

Additionally, this experiment presented an empirical correlation to calculate the Nusselt number based on the Rayleigh number as a parameter and used S as the characteristic length. The following is the general equation for heat transfer by external natural convection:

$$Nu_s = C(Ra_s)^n \quad (21)$$

Where, C is the constant coefficient, and n is the constant exponent, which are affected by the surface geometry.

The constant-coefficient (C) is 0.1096 and the power (n) is 0.3251 and the squared correlation coefficient R^2 of the fit is 0.992. These values are valid in range $7.36 \times 10^3 < Ra_s < 2.26 \times 10^4$ and for fin spacing 17 mm, these values show a very well fits with the experimental data. Thus, a new empirical correlation relation is present.

$$Nu_s = 0.1096(Ra_s)^{0.3251} \quad (22)$$

for $\quad 7.36 \times 10^3 < Ra_s < 2.26 \times 10^4$

3.1.3. Effect heating input

Figures 8 and 9 show the Nusselt number and convective heat transfer coefficient as a function of the heating input. It shows clearly that heat input strongly effects on the Nusselt number and the convective heat transfer. As the heat input increases, the Rayleigh number increases as well as the natural convection. This is due to an increase in the differential temperature between the heat sink fin array and the surroundings. Additionally, these figures indicate that the maximum heat transfer coefficient and Nusselt number are (4.649 W/m^2 K) and (2.8337), respectively. The results are in good agreement with the experimental correlations with a maximum error of less than 3 % for all experiments.

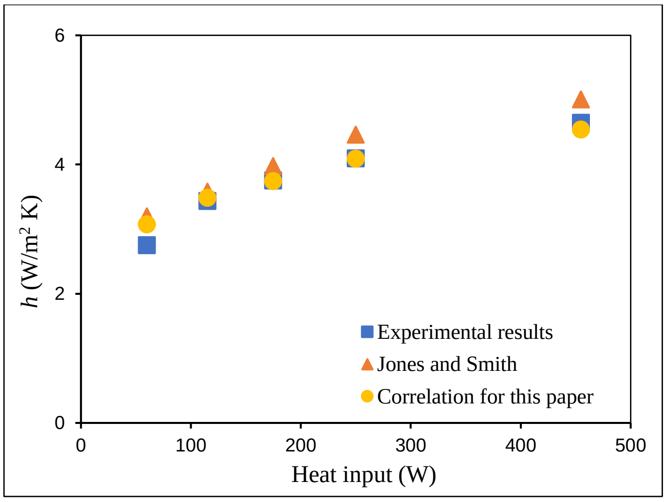

Fig. 8 Heat transfer coefficient as a function the heating input for the horizontal case.

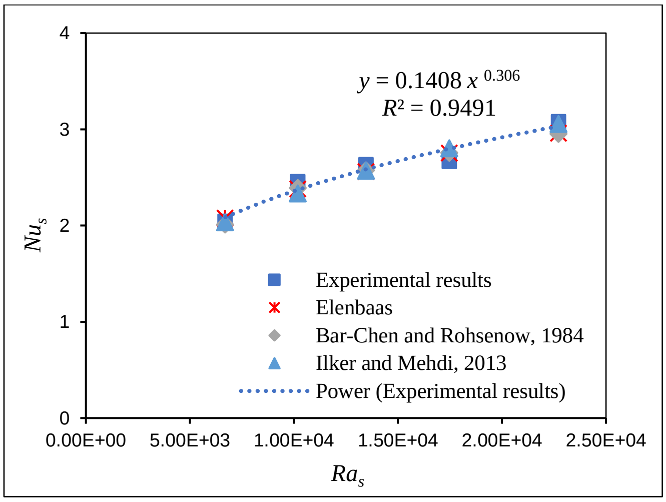

Fig. 9 Validation of the Experimental results for vertical heat sink with vertical fins.

3.2. Vertical model

The experimental study was conducted on the rectangular fin array with heat transfer of laminar natural convection in a vertical base with a vertical plate fin as shown in Fig. 1.

3.2.1. Validation the results

To confirm the experimental procedure, the results were compared with empirical correlations from the literatures [2, 19, 20] after replacing the finned heat sink with a vertical flat plate. As followers, these correlation relations are used as characteristic lengths.

Elenbaa's Correlation [2],

$$Nu_s = \frac{h\,S}{k} = \frac{1}{24}\frac{Ra_s\,S}{H}\left[1 - exp\left(\frac{-35}{\frac{Ra_s\,S}{H}}\right)\right]^{\frac{3}{4}} \tag{23}$$

where $Ra_s = \dfrac{g\,\beta(T_s - T_a)S^3}{V\,\alpha}$

Bar-Cohen and Rohsenow, 1984 [19],

$$Nu_s = \frac{h\,S}{k} = \left[\frac{576}{\left(\frac{Ra_s\,S}{H}\right)^2} + \frac{2.873}{\left(\frac{Ra_s\,S}{H}\right)^{0.5}}\right]^{-0.5} \tag{24}$$

Ilker and Mehdi, 2013 [20],

$$\overline{Nu_s} = 0.0929(Gr_s'\,Pr)^{\frac{1}{2}} \qquad Gr_s'\,Pr < 250$$

$$\overline{Nu_s} = 0.252(Gr_s'\,Pr)^{1/3} \qquad 250 < Gr_s'\,Pr < 10^4 (H \geq 15\text{ mm})$$

$$\overline{Nu_s} = 0.2413(Gr_s'\,Pr)^{1/3} \qquad 250 < Gr_s'\,Pr < 10^6$$

where $Gr_s' = \left(\dfrac{g\,\beta(T_w - T_a)S^3}{\nu^2}\right)\left(\dfrac{S}{H}\right)\left(\dfrac{H}{L}\right)^{1/2}$

Figure 10 shown, the above correlations were also plotted along with experimental results. Additionally, the plots show the variation Nusselt number as a function of the Rayleigh number. As a result, the Nusselt number increases as the Rayleigh number increased due to the difference in temperature between the fin array and the surroundings, more heat is dissipated by natural convection. The increased were founded the Nusselt number (3.079). This indicates a good arrangement with all correlations with a maximum error less than 5 %.

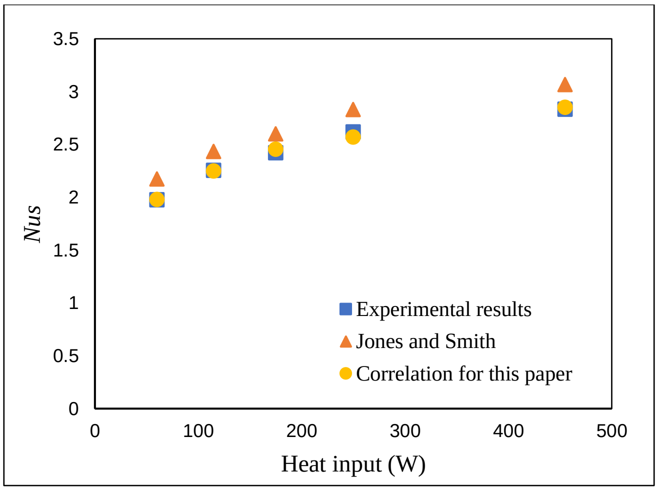

Fig. 10 Nusselt number as a function the heating input for the horizontal case.

3.2.2. Average Nusselt number correlation

This study was carried out to present an empirical correlation used by the Rayleigh number as a parameter to calculate the Nusselt number based on the characteristic length S. The following equation describes the heat transfer process by natural convection.

$$Nu_s = C(Ra_s)^n \tag{25}$$

Where C is the constant coefficient and n is the constant exponent, which are affected by the surface geometry. From equation 25, the constant-coefficient (C) of 0.1408 and the power (n) of 0.306 and the squared correlation coefficient R^2 of the fit is 0.9491. These values are valid in range $6.7 \times 10^3 < Ra_s < 2.27 \times 10^4$ and for fin spacing 17 mm these values show a very well fits with the experimental data. Thus, a new empirical correlation relation is present.

$$Nu_s = 0.1408(Ra_s)^{0.306} \tag{26}$$

for $6.7 \times 10^3 < Ra_s < 2.27 \times 10^4$

4.2.3. Effect heating input

Figures 11 and 12 show the convective heat transfer coefficient and Nusselt number as a function of the heating input. It can be seen that the convective heat transfer coefficient and Nusselt number increase as heat input increases. It is due to an increase in the temperature difference between the heat sink fin array and the surrounding area that enhances natural convection. Further, the results show a maximum heat transfer coefficient of (4.9816 W/m^2 K) and Nusselt number of (3.079), respectively. All experiments are in good agreement with the experimental correlations with a maximum error of less than 5 %.

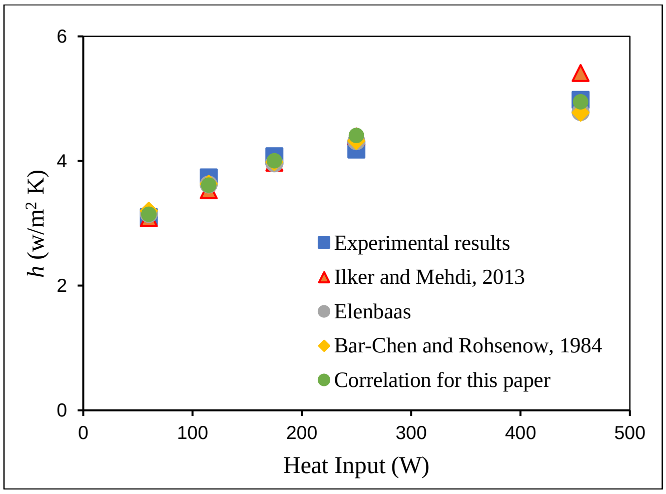

Fig. 11 Heat transfer coefficient as a function the heating input for the vertical case.

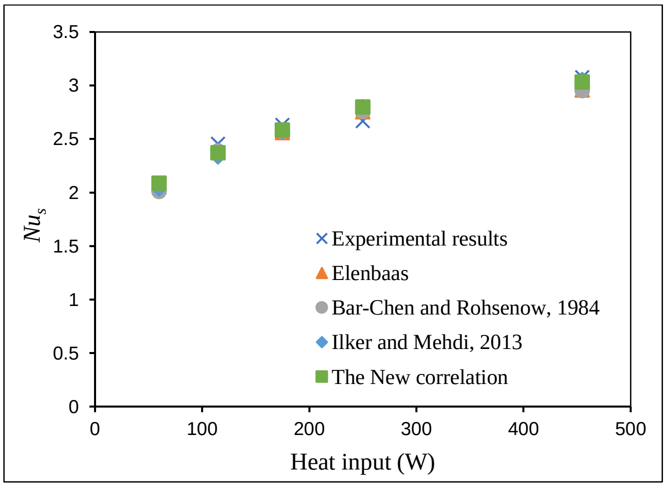

Fig. 12 Nusselt number as a function the heating input for the vertical case.

3.2.4. Effect inclination angle

The experimental tests were carried out on the effect of changing the direction of acceleration by changing the angle of inclinations (θ) of fin array on the convection heat transfer coefficients and Nusselt number at different heater inputs. The angle of inclinations is 0, 30, 45, 60, and 90 degrees from vertical position. Figs. 13 and 14 show that the heat transfer coefficient and Nusselt number are influenced by heat input and changing the incline angle. This is due to changes in gravitational acceleration, which had a significant effect on buoyancy forces. It has been noted that the heat transfer coefficient and Nusselt number are at maximum values at 30 degrees of inclination from vertical, with the maximum heat transfer coefficient (5.305 W/m^2 K) and maximum Nusselt number (3.243) at this angle. The percentage improvement in the total heat transfer coefficient is found at approximately (6%) for five heating inputs.

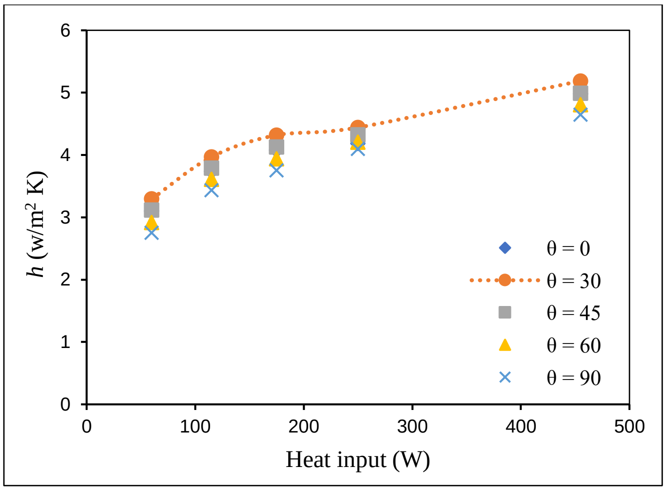

Fig. 13 Heat transfer coefficient as a function of heat supplied for different Inclination angles.

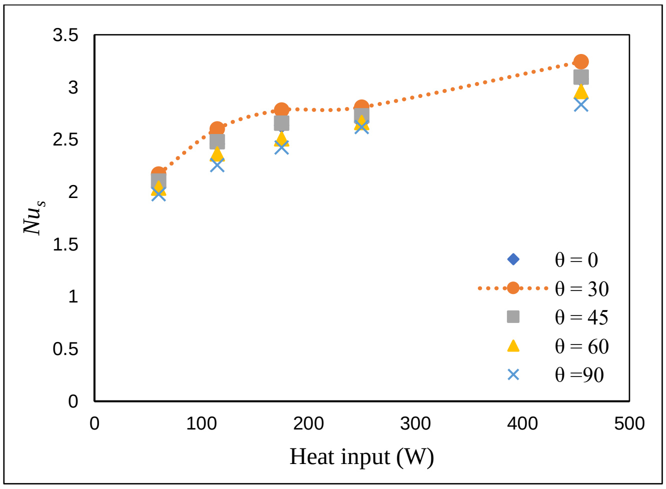

Fig. 14 Nusselt number as a function of heat supplied for different inclination angles.

4. Conclusions

An experimental study was conducted in this paper to investigate the effect of increasing the heating input on steady-state natural convection from heat sink fin arrays in horizontal, vertical, and inclined cases. According to the results, when heating input increases, the heat transfer coefficient increases by 38 %, and 40.78 %, in horizontal and vertical cases, respectively. As for the inclined case, it was found that a 30 degree inclination increased the heat transfer coefficient by 6%. For horizontal and vertical cases, new correlations have been presented to calculate the Nusselt number influenced by the Rayleigh number. Radiative heat transfer rates are calculated in the channels of heat sinks using the view factor method with high accuracy. The results showed that the radiative heat transfer was less than 5 % of the total heat input for all experiments.

Nomenclature		
Symbol	**Description**	**SI Units**
Q_{tot}	Total power input	W
Q_{conv}	Convection heat transfer rate	W
Q_{rad}	Radiation heat transfer rate	W
h_{av}	Average heat transfer coefficient	$W/m^2 K$
A_s	The total surface area	m^2
A_{base}	Aluminum plate surface area	m^2
A_{side}	Fin side surface area	m^2
A_{tip}	Fin tip surface area	m^2
T_b	Bulk temperature	K
T_∞	The ambient temperature	K
T_f	The film temperature	K
k	Thermal conductivity of air	W/m K
S	Fin spacing	m
Ra	Rayleigh number	-
Gr	Grashof number	-
Pr	Prandtl number	-
g	Gravitational acceleration	m/s^2
L	Fin length	m
H	Fin height	m
N	Fins number	-
Nu	Nusselt number	-
F	The view factor of the shape	-
	Greek Symbols	
Symbol	**Description**	**SI Unit**
β	Thermal expansion coefficient	1/K
ν	Kinematic viscosity	m^2/s
α	Thermal diffusivity	m^2/s
σ	Stefan-Boltzmann constant	$W/m^2 K^4$
ε	Heater surface emissivity	-

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