An experimental simulation of free convection heat performance for rectangular-fins array inside an enclosure: the direction and inclination effect

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Abstract. In this work, an experimental simulation has been done to predict the thermal performance of the free convection from a heated rectangular-fins array to ambient air inside an open ended wooden enclosure. Three various directions of fins array named front, top and bottom. Also, two inclinations of front direction fins array are 30° and 60° are utilized. The fins array is heated from the base. Wide ranges of power input magnitudes to the fins array base have been employed. The effect of fins array direction, inclination angle of fins array, heat flux and surface temperature on the thermal performance of free heat convection have been simulated and investigatd experimentally. The experimental data refers to the front direction fins array gave best thermal performance in term of Nusselt number about 6% to 10% and 22% to 27% larger than top and bottom directions respectively. Three empirical correlation equations to predict Nusselt number values for the front, top and bottom directions rectangular-fins array are suggested. The present results are compared than previous researches and a good proximities in behavior is observed.

Keywords:
Rectangular-fins array, direction, inclination, free convection, experimental simulation.

1. Introduction
The methods of thermal performance improvement of heat engineering systems and industrial applications are a great importance subjects for researchers. The important of these methods are increasing the convection heat transfer from a heated surfaces to surrounding utilizing the extended surfaces (fins) or fins array with changing different parameters fins configurations, geometrical parameters of fins like fin height, fin thickness and fin spacing, inclinations and directions of fins or fins array. Fins arrays are one of very important methods for a direct-air cooling. The convection heat transfer occurs in a several industrial and engineering applications such as electronic devices cooling, high-power chips, power transformers, electrical motors, heat solar collectors, energy storage systems, heat exchangers, radiators.

Sparrow and Vemuri [1] investigated the effect of directions a on free thermal convection and radiation heat dissipation from pin-fins array. They concluded that the upward direction yielded the largest rates of heat transfer than those sideward and downward directions. Rao et al. [2] achieved a numerical investigation of combined free heat convection and radiation heat transfer from a vertical base-plate with rectangular-fins array with several fin spaces. They numerically solved the governing equations of problem using alternate direct implicit technique. They showed that the rates of free convection heat transfer increasing as fin spaces decreases. Al-Azawi [3] introduced an experimental study of free heat convection from a trapezoidal-fins array inside an enclosure to surrounding with different directions. He noted the values of convection heat transfer coefficient for sideward vertical-fins array are higher about 12% than that upward direction. Naidu et al. [4] experimentally and numerically studied of free convection from a fin arrays with different inclination angles for two
different geometries namely, vertical and horizontal fins array. They utilized the alternate direct implicit method for a numerical solution. They showed that the values of convection heat transfer coefficient of vertical-position fins array are larger than that horizontal fins array for all used inclination angles. Al-Azawi et al. [5] experimentally investigated of free heat transfer convection for pin-fins array with three different directions like upward, downward and sideward. They showed the coefficients of free heat convection for sideward pin-fins array are larger about 8% than that upward direction. Also, they noted that the temperatures distribution along the pin-fin of sideward direction has a uniform distribution. More et al. [6] presented a review study of free convection from a heated base plate with different geometries of the fin arrays and inclination angles. They showed that the geometries and inclinations of fins arrays enhanced the thermal performance and heat transfer rate with different percentages. Shehab [7] experimentally investigated of free heat convection from a rectangular and V-fins arrays with three different inclination angles are 15°, 30° and 45°. He showed the thermal performance of V-fins array best than rectangular-fins array. Also, he noted that the coefficient of free convection heat transfer decreases with inclination angles increasing about % 15 to 25% than that vertical position.

The objective of present work is to introduce an experimental simulation for the free heat convection from a rectangular-fins array to ambient air inside an open ended rectangular cross-section enclosure utilizing three different directions of fins array front, top and bottom with two different inclinations of front direction fins array (30° and 60°) for a wide ranges of heat fluxes.

2. Experimental test-rig

The experimental test-rig is manufactured to achieve the experiments of present work as shown photographically in figure 1. It consists of assembly of fins array, holder with inclination mechanical system, open ended wooden enclosure, voltage-regulator to govern of heat inputs to heating element and measurement instruments like digital multi-meter, data-logger thermometer and K type calibrated thermocouples.

The fins array is fabricated from a shiny aluminum sheet plate as one piece with base plate using CNC milling machine. The dimensions of base plate are 120 mm length × 80 mm width × 4 mm thickness. The fins array consists of six rectangular-fins with constant dimensions of fin (120 mm length × 16 mm height × 3 mm thickness) and fin spacing of 12.40 mm. It’s heated from a bottom face of base plate utilizing heating element allotted for this purpose. It is putted on top face of thermo stone’s block (200 mm length × 160 mm width × 80 mm thickness) to minimize the thermal conduction lost from ends and bottom face of fins array base plate and then is fixed with holder inside an open ended wooden enclosure from top and bottom. The holder is purposed to allow the moving of fins array assembly with multi directions and inclination angles as shown in the figure 2.

The surface temperatures of fins array base are measured by a five calibrated thermocouples are fixed for a suitable locations on bottom surface of base plate (four in the corners and one in the center). Another five thermocouples are used to measure the temperatures difference for bottom and ends of fins array to compute the thermal conduction lost through a thermo stone’s block. The thermocouples are join to a multi channels data-logger thermometer. Plus one digital thermocouple is utilized to read the ambient air temperature inside an enclosure. The steady-state conditions for all experiments are arrived after (60 to 75) minutes. The surface temperatures, voltage and current at steady-state are registered.
3. Calculation procedures

The present work is carried out for three various directions of fins array which are, front, top and bottom. Also, two inclination angles (Θ) of front direction fins array (30° and 60°) are used. Five various of heat fluxes (q) values, 2604, 5208, 7812, 10416 and 13020 W/m² are utilized. The rate of free heat convection (Q\text{conv}) can be computed from equation:

\[ Q_{\text{conv}} = V I - (Q_{\text{rad}} + Q_{\text{cond}}) \]  

(1)

Also, it can be calculated from cooling Newton’s equation [8, 9]:

\[ Q_{\text{conv}} = h A_s (T_s - T_{\infty}) \]  

(2)

Then, the free convection heat transfer coefficient (h) can be computed as follows:

\[ h = \frac{V I - (Q_{\text{rad}} + Q_{\text{cond}})}{A_s (T_s - T_{\infty})} \]  

(3)

and the average convection heat transfer coefficient (h_{av}) can be evaluated as follows:

\[ h_{av} = \frac{V I - (Q_{\text{rad}} + Q_{\text{cond}})}{A_s (T_{sav} - T_{\infty})} \]  

(4)

The loss of thermal radiation (Q_{rad}) between surfaces of aluminum fins array (has emissivity of \( \varepsilon = 0.05 \)) and air ambient temperature inside a wooden enclosure is evaluated utilizing Stefan-Boltzmann law [8].
The losses of thermal conduction ($Q_{\text{cond}}$) from bottom and ends of fins array across thermo stone’s block (has thermal conductivity of $k_t = 0.15 \text{ W/m.K}$) are calculated utilizing Fourier’s law [8]. The average surface temperature ($T_{\text{sav}}$) is:

$$T_{\text{sav}} = \frac{\sum_{i=1}^{N} T_{si}}{N} \quad (5)$$

The surface area of free convection heat dissipation for rectangular fins array ($A_s$) is calculated as:

$$A_s = (L^2 - ntL) + 2nHL \quad (6)$$

Where $T_s$ and $T_\infty$ are the local surface and ambient temperatures respectively, $H$, $L$ and $t$ are the height, length and thickness of fin respectively, $n$ is the fins number.

The Nusselt number ($Nu$) based on the fin spacing ($S$) as the characteristics length can be computed as [9]:

$$Nu = \frac{hS}{k_a} \quad (7)$$

Define Grashof number ($Gr$) and Rayleigh number ($Ra$) as follows [8, 9]:

$$Gr = \frac{g \cos(\Theta) \beta (T_s - T_\infty) S^3}{\theta^2} \quad (8)$$

$$Ra = Gr Pr \quad (9)$$

Where $\Theta$ is the inclination angle of fins array.

Air properties are used in the present work done at film temperature ($T_f$).

Figure 3 appears a comparison of top direction fins array for present work than Al-Azawi work. It is noted from the experimental data of comparison a good agreement in behavior.

![Figure 3. Comparison of present work for top direction fins array with Al-Azawi work.](image)

4. Results and discussion
The present work simulates and studies experimentally the effect of direction and inclination angle on the thermal performance of fins array under free convection conditions at a wide range of heat fluxes. Figure 4 illustrates the behavior of free heat convection coefficient against temperatures difference for three various directions (front, top and bottom). In addition to two inclination angles of front direction ($30^\circ$ and $60^\circ$) for surface heat flux, $q=7812 \text{ W/m}^2$. It is clear that the free heat convection coefficient ($h$) gradually increases as temperatures difference ($\Delta T$) increases. Also, the maximum values of free heat transfer coefficient occurs at front direction for vertical position ($\Theta=0 \text{ deg.}$) compared with those top and bottom directions and $30^\circ$ and $60^\circ$ inclination angles. They are greater about $12\%$ to $21\%$ than those cases of fins array with inclinations $30^\circ$ and $60^\circ$ respectively, because of the more fresh air enters from the lower of fins array channels and contacted the hot faces of fins and then the fresh air inflow is heated.
and exits from the top. In addition to, a little of fresh air enters from the top of fins channels. This occurs as a result and effect of bouncy force which produced with occurrence a difference in temperatures and change in air inflow density.

![Figure 4. Heat transfer coefficient versus temperatures difference for different directions and inclinations of fins array at surface heat flux (q= 7812 W/m²).](image)

Figure 4. Heat transfer coefficient versus temperatures difference for different directions and inclinations of fins array at surface heat flux (q= 7812 W/m²).

Figure 5 shows the behavior of average heat transfer coefficient against surface heat flux for front, top and bottom directions. It is observed that the average free heat convection coefficient ($h_{av}$) increases gradually with increasing values of wall heat flux (q) for all directions of fins array are studied. Also, the average heat transfer coefficient reaches a maximize at front direction because of a large temperatures difference between fin array surfaces and fresh inflow air than those other directions.

![Figure 5. Average heat transfer coefficient versus surface heat flux for different directions of fins array.](image)

Figure 5. Average heat transfer coefficient versus surface heat flux for different directions of fins array.

Figures 6, 7 and 8 illustrate the behavior of Nusselt number versus Rayleigh number for three various directions of fins array at three different levels of wall heat fluxes namely, q= 2604, 7812 and 13020 W/m². They are clear that the Nusselt number values increases frequently with increasing Rayleigh number values. Also, the maximum values of Nusselt number at front direction of fins array because of
surface temperature values decrease and reach a minimize at front direction of fins array. Then, the front direction fins array gave best thermal performance in term of Nusselt number about 6% to 10% and 22% to 27% greater than those cases of top and bottom directions fins arrays respectively.

Three empirical correlation equations are concluded to predict the Nusselt number (Nu) as a function of Rayleigh number for three cases of fins array directions namely, front, top and bottom as follows:

1. Front direction (vertical position):
   \[ Nu = 0.05 \cdot Ra^{0.6067} \]  \hspace{1cm} (10)

2. Top direction:
   \[ Nu = 0.027 \cdot Ra^{0.6537} \]  \hspace{1cm} (11)

3. Bottom direction:
   \[ Nu = 0.0598 \cdot Ra^{0.5495} \]  \hspace{1cm} (12)

![Figure 6](image6.png)

**Figure 6.** Nusselt number versus Rayleigh number for different directions of fins array at surface heat flux (q= 2604 W/m²).

![Figure 7](image7.png)

**Figure 7.** Nusselt number versus Rayleigh number for different directions of fins array at surface heat flux (q= 7812 W/m²).
Figure 8. Nusselt number versus Rayleigh number for different directions of fins array at surface heat flux \(q=13020\) W/m\(^2\).

5. Conclusions
In this study, experimental simulation has been achieved on three different directions, front, top and bottom and two various inclination angles of front direction, 30\(^\circ\) and 60\(^\circ\) for rectangular fins array under free heat convection condition. The fins array for front direction case gave best thermal performance in term of Nusselt number about 6% to 10% and 22% to 27% larger than that top and bottom fins array respectively. Values of free convection heat transfer coefficient detracts with inclination angle increasing. The heat convection coefficients for front direction for fins array with vertical position are greater about 12% to 21% than that fins array with inclinations 30\(^\circ\) and 60\(^\circ\) respectively. Three experimental correlations to predict the values of Nusselt number for the front, top and bottom directions rectangular-fins array are developed.

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Nomenclature

- \(A_s\) surface area, (m\(^2\))
- \(g\) Gravitational acceleration, (m/s\(^2\))
- \(Gr\) Grashof number
- \(h\) free convection heat transfer coefficient, (W/m\(^2\).K)
- \(H\) fin height, (m)
- \(I\) input current, (A)
- \(k_a\) air thermal conductivity, (W/m.K)
- \(L\) fin length (base plate length), (m)
- \(n\) number of fins
- \(Nu\) Nusselt number
- \(Pr\) Prandtl number
- \(q\) surface heat flux, (W/m\(^2\))
- \(Q_{\text{cond}}\) loss of thermal conduction, (W)
- \(Q_{\text{conv}}\) rate of free convection heat transfer, (W)
- \(Q_i\) electrical power input, (W)
- \(Q_{\text{rad}}\) loss of thermal radiation, (W)
- \(Ra\) Rayleigh number
S  fin spacing, (m)
 t  fin thickness, (m)
 $T_\infty$ air ambient temperature, (°C)
 $T_s$ surface temperature, (°C)
 V  voltage, (V)
 $\Delta T$ temperature difference, (°C)
 $\Theta$ inclination angle, (deg.)
 $\beta$ volumetric coefficient of thermal expansion, (1/K)
 $\nu$ kinematic viscosity of the air, (m$^2$/s)
 $\bar{v}$ Average

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