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Experimental Simulation of Natural Heat Convection from Finned Vertical Plate with Different Inclinations

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Abstract

In this work an experimental simulation is made to predict the performance of steady-state natural heat convection along heated finned vertical base plate to ambient air with different inclination angles and configurations of fin array. Two types of fin arrays namely vertical fins array and V-fins array on heated vertical base plate are used with different heights and spaces. The influence of inclination angle of the plate , configuration of fins array and fin geometrical parameters such as fin height and fin spacing on the temperature distribution, base convection heat transfer coefficient and average Nusselt number have been plotted and discussed. The experimental data are correlated to a formula between average Nusselt number versus Rayleigh number for vertical plate and vertical fins array. The results indicate that the configuration of V-fins array gave best natural-convection heat transfer performance as base heat transfer coefficient about 20% greater compared with vertical fins array. Experimental simulation data and correlations of the present work are compared with a previous works shows good agreement.

Keywords: *Natural-convection, experimental simulation, finned plate, fins array.*

1. Introduction

The configurations of fins and inclinations play an important role in natural-convective heat transfer. Fins and fin arrays are extended surfaces used to enhance and increase heat transfer rate and heat dissipation from heated surfaces to the surroundings in electronic devices and engineering applications. Many configurations of fin arrays like vertical fins, triangular fins, trapezoidal fins, tree-shaped fins , pin fins ,V-shaped and V-shaped with bottom spacing. Fin arrays on vertical base plates of various shapes are of great importance in industrial applications such as cooling of the electronic and micro- electronic equipments , high voltage power transformers , motors, high power chips, nuclear reactor cores , compact heat exchangers , radiators, heating of the houses , energy storage system and solar collectors .

Churchill and Chu [1] deduced an empirical correlation to predict the Nusselt number for steady-state free-convection heat transfer from vertically heated base plate in a laminar and turbulent flow conditions .Vermeulen and Baudoin [2] obtained the experimental free-convection heat transfer correlations for vertical and inclined flat plate after several experiments. They measured the temperatures by using infrared (IR) thermography. The results show that the increase in convection heat transfer coefficient of about 10% when roughness elements (fins) are used than flat surfaces. Rao et al. [3] investigated numerically the combined free-convection and radiation heat transfer from vertical plate with fins array. They solved the governing equations using Alternate Direct Implicit (ADI) method. They noted that the convection heat transfer rate increasing with fin spacing decreases and fin length increases. Abid [4] studied experimentally the effects of fin shape on laminar natural convection. He used vertical and pin fins array and developed empirical correlations for vertical fins and pin fins array in laminar condition flow. Sable et al. [5] investigated enhancement of free-convection heat transfer on heated vertical plate by multiple V-fins array. The results show that the V-fins array gave better heat transfer performance than vertical fins array and V-fins with bottom spacing array. Naidu et al. [6] studied the problem of natural convection experimentally and numerically by using Alternate Direct Implicit (ADI) method from fin arrays with different inclination angles of two geometric orientations, vertical and horizontal fins array. The results show that the convection heat transfer rate of vertical fins array is great than horizontal fins array for the same inclinations angles. Fahiminia et al. [7] presented computational analysis of the natural-convection on extended vertical base surfaces by using finite volume method (FVM). They concluded that the heat convection rates increases with increasing fin space reaches to optimum spacing . More et al. [8] presented review study of natural-convection from heated plate with different configurations of the fin arrays and inclinations. The results of study show that the all configurations of fin arrays are improved thermal design and heat dissipation rate of the heated surfaces with different percentages. Hireholi et al. [9] investigated experimentally and theoretically the heat transfer by free-convection of heat sink used for cooling of the electronic chips. They found the optimum fin spacing. They compared that the experimentally measured temperatures of heat sink with theoretically predicted temperatures using two-dimensional model and show a very good agreement. Tiwari and Malhotra [10] studied heat transfer of laminar natural-convection over a flat plate bounded by enclosures with effects of the surface roughness , ambient temperature , flow velocity and surface inclinations on the convection heat transfer coefficient at different heat source input. They noted that the increase in heat source input increases the heat transfer rates and the increase in surface inclination decreases the heat transfer coefficient.

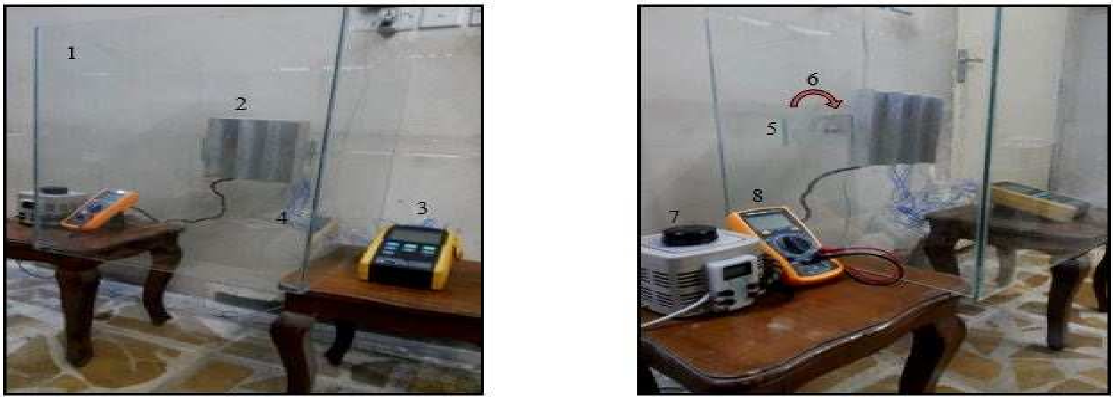
The present work concentrates experimentally on the effects of configuration of fins array, base plate inclinations (ϕ), fin height (H) and fin spacing (S) on performance of natural-convection heat transfer over a heated vertical base plate. This is assist to predict of the temperatures distribution along a vertical heated base plate and to calculate convection heat transfer coefficient with and without fin arrays to choose the best configuration and design of fins array.

2. Experimental Work

2.1. Experimental Rig

The experimental test rig and it’s components shown in Fig. 1 consists of an aluminum square base plate of 200 mm side has thickness 2.0 mm. Vertical plate and two configurations of fin arrays vertical fins and V-fins are joined on the base plate are tested as shown in Fig. 2 with different fin height (H) and fin spacing (S) having thickness 2.0 mm. The vertical base plate is heated from backside using an electrical heater wire is coiled around mica sheet and then is sandwiched between two symmetrical square sheets of mica with same dimensions of the base plate and thickness of 0.5 mm to obtain homogeneous heating of the base plate and ensured the electrical insulation of heater wire. It’s fixed on the backside of base plate by thermal super glue. The backside of base plate assembly is a good insulation using polyurethane foam layer with 60 mm thickness to minimize the conduction heat loss. Also the sides are framed by plywood frame. The whole assembly of base plate is fixed in a vertical position with an adjustable support allowing different angles of inclination in a square enclosure constructed of cast acrylic sheet of thickness 6 mm, with dimensions (500 mm length × 500 mm width × 600 mm height) opened from upper and lower ends under guaranteed a good free-convection heat transfer conditions. The inclination angles of base plate are measured with vertical position by a fixed Protractor. The heating element is supplied up to 300 W with stabilized alternating current (AC) and voltage of 220 V through a contact type voltage regulator with digital reader type SAKO-TDGC$_2$ to control on the electrical heat input and digital multi-meter type VICTOR-VC890C$^+$ to measure voltage and current.

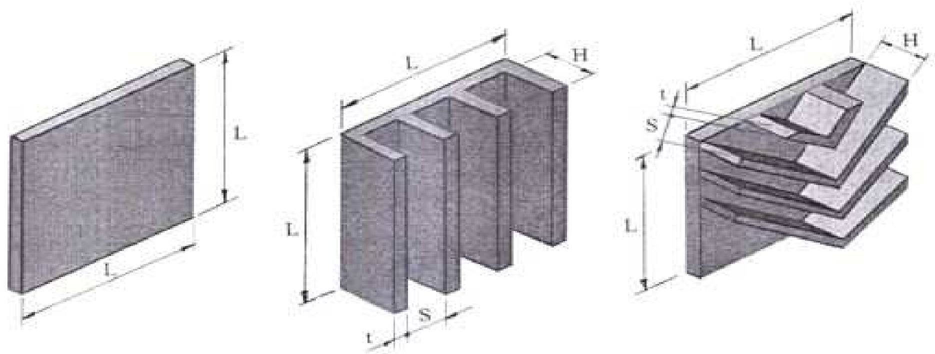
Twelve K-type calibrated thermo-couples are embedded at different suitable locations in back surface of the vertical base plate to measure surface temperatures. They are matrix form distributed (4 rows × 3 columns) with equal distances. The thermocouples are connected to twelve channels digital temperature recorder type BTM-4208SD. Another two same K-type calibrated thermocouples are joined in digital multi-meter to record temperatures difference to evaluate the conduction heat loss from the backside of heated base plate. Additional two separate K-type digital calibrated thermocouples are used to measure the ambient temperature inside the enclosure.



a. Front photo b. Side photo

1. Opened enclosure 2. Finned plate assembly. 3. Digital temperature recorder. 4. Thermocouple wires. 5. Adjustable support. 6. Protractor. 7. Voltage regulator. 8. Digital multi-meter.

Fig. 1. Photos of the experimental rig.



a. Vertical plate b. Vertical fins array c. V-fins array

Fig. 2. Tested vertical plate and fin arrays configurations.

2.2. Experimental Procedure and Calculations

Three cases are studied , vertical base plate , vertical base plate with vertical fins array and vertical base plate with V-fins array. Three angles of inclination with respect to vertical position are used (ϕ= 15^o , 30^o and 45^o). Cases of fin arrays is carried out for different fin height (H= 15 , 30 and 45 mm) and fin spacing (S= 14.5 , 20 , 31 and 64 mm) depending on numbers of fins as Table 1. Multi- range of the electrical heater input Wattage namely 25 , 50 , 75 , 100 , 125 and 150 W are used. All readings of the temperatures are recorded under steady-state conditions every 45 minutes approximately and when varying of the temperature readings less than 0.5 ^{o}C.

Table 1,
Number of fins against fin spacing in fins array.

Number of fins, N	4	7	10	13
Fin spacing (S), mm	64	31	20	14.5

The electrical heat input (Q_{in}) to the heating element is :

$$Q_{in} = I\,V \qquad \dots(1)$$

It's transferred to the ambient mainly by natural-convection (Q_C) in addition to radiation (Q_R) and conduction (Q_{Cd}) heat transfer losses. Then, the rate of natural-convection heat transfe (Q_C) can be evaluated as :

$$Q_C = I\,V - Q_R - Q_{Cd} \qquad \dots(2)$$

The radiation heat transfer rate (Q_R) is [11] :

$$Q_R = \varepsilon\sigma A_S F(T_{Sav}^4 - T_A^4) \qquad \dots(3)$$

The conduction heat transfer rate (Q_{Cd}) is calculated by Fourier's equation as [11,12] :

$$Q_{Cd} = k\ \frac{A_b}{X}\ \Delta T \quad \text{... (4)}$$

The radiation heat transfer (Q_R) from the heated base plate and fins array is small because the emissivity (ε) of bright rolled aluminum used in manufacturing the base plate and fins is 0.04 and it's found to be less than 5% of electrical heat input (Q_{in}) for all cases. The loss by heat conduction (Q_{Cd}) is minimized using a thick layer of polyurethane foam with very low of thermal conductivity about 0.028 W/m.K . It's found to be approximately 2% of heat input (Q_{in}) .

According to Newton's law of cooling, the net heat transfer from the base heated plate by natural-convection is expressed as [11] :

$$Q_C = h_b A_b (T_{Sav} - T_{Aav}) \quad \text{... (5)}$$

Hence, the base convection heat transfer coefficient (h_b) is :

$$h_b = \frac{I\,V - (Q_R + Q_{Cd})}{A_b (T_{Sav} - T_{Aav})} \quad \text{... (6)}$$

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

$$h_{av} = \frac{I\,V - (Q_R + Q_{Cd})}{A_t (T_{Sav} - T_{Aav})} \quad \text{... (7)}$$

The total surface area of free-convection heat transfer (A_t) for vertical fins array is :

$$A_t = (A_b - NLt) + 2NHL \quad \text{... (8)}$$

and for V-fins array is :

$$A_t = \left(A_b - t\sum_{i=1}^{N} L_{Vi}\right) + 2H\sum_{i=1}^{N} L_{Vi} \quad \text{... (9)}$$

where , L_V is the length of V-fin.

The average Nusselt number (Nu_{av}) at the heated plate is computed by [11,12]:

$$\mathrm{Nu_{av}} = \frac{\mathrm{h_{av} L_C}}{\mathrm{k}} \quad \text{... (10)}$$

The average Grashof number (Gr_{av}) can be defined as follows [11,12] :

$$Gr_{av} = \frac{buoyancy\ forces}{viscous\ forces} = \frac{g\beta cos\phi (T_{Sav} - T_{Aav}) L_C^3}{\vartheta^2} \quad \text{... (11)}$$

Define Rayleigh number (Ra) as the product of the Grashof and Prandtl numbers [11] :

$$Ra = Gr_{av}\,Pr \quad \text{... (12)}$$

The average surface temperature (T_{Sav}) is:

$$T_{Sav} = \sum_{i=1}^{n} \frac{T_S}{n} \quad \text{... (13)}$$

and the average of ambient temperature (T_{Aav}) is sum of two readings of ambient temperatures inside an enclosure (T_{A1}) and (T_{A2}) divided by two to obtained a more accuracy :

$$T_{Aav} = \frac{T_{A1} + T_{A2}}{2} \quad \text{... (14)}$$

All properties of air are taken at film temperature (T_f) :

$$T_f = \left(\frac{T_{Sav} + T_{Aav}}{2}\right) + 273.18 \quad \text{... (15)}$$

where, L_C is the characteristics length of geometry , L_C= L for vertical base plate, L_C= S for vertical fins array and V-fins array configurations [11,12] .

3. Results Analysis and Discussion

The utilized empirical correlations for comparison with the experimental average Nusselt number (Nu_{av}) of natural-convection heat transfer over heated vertical plate under steady-state and laminar flow conditions are listed in Table 2 .

Table 2,
Available empirical correlations.

Empirical correlation	Expression with limitations
McAdam's [11, 13]	$Nu_{av} = 0.59\ (Ra)^{0.25}$ ... (16) ($10^4 <$ Ra $< 10^9$)
Churchill and Chu's (first relation) [1,11]	$Nu_{av} = \left[0.825 + \frac{0.387 Ra^{1/6}}{\{1 + \left(\frac{0.492}{Pr}\right)^{9/16}\}^{8/27}}\right]^2$ ... (17) ($10^{-1} <$ Ra $< 10^{12}$)
Churchill and Chu's (second relation) [1,11]	$Nu_{av} = 0.68 + \frac{0.67 Ra^{1/4}}{\{1 + \left(\frac{0.492}{Pr}\right)^{9/16}\}^{4/9}}$ ... (18) ($10^{-1} <$ Ra $< 10^9$)
Churchill and Usagi's [11]	$Nu_{av} = \frac{0.67 Ra^{1/4}}{\{1 + \left(\frac{0.492}{Pr}\right)^{9/16}\}^{4/9}}$ ... (19) ($10^5 <$ Ra $< 10^9$)

Also the experimental simulation data of laminar natural heat convection from vertical fins array case are compared with empirical correlation of Abid [4] :

$$Nu_{av} = 0.045(Ra^{'})^{0.75} \quad \dots(20)$$

where ,

$$Ra^{'} = Ra \left(\frac{S}{L}\right) \quad \dots(21)$$

Variation of the base convection heat transfer coefficient (h_b) with average surface temperature (T_{Sav}) for vertical base plate , vertical fins and V-fins array at fin heights (H) ranging (from 15 to 45 mm) and spacing (S= 64 mm) are shown in Figures (3) and (4). The values of base convection heat transfer coefficient (h_b) in V-fins array are greater than vertical fins array case because of created a low pressure suction region in the corners of V-shaped fins on the down-stream side of V-fins therefore , the cold air flows into the separation region. It's caused to increase the base convection heat transfer coefficient (h_b) and the heat dissipation rate.

Figures (5-7) show variation in the base convection heat transfer coefficient (h_b) with inclination angle (ϕ) at heat input ranging from 25 W to 150 W when fin height (H= 30 mm) and spacing (S= 64 mm) for vertical base plate , vertical fins and V-fins array respectively. As the fin height (H) increased , the average surface temperature (T_{Sav}) decreased because a more of cold air enters between fins and the base convection heat transfer coefficient (h_b) increased . In general, the base heat transfer coefficient (h_b) decreases with inclinations (ϕ) increasing and heat input decreasing.

Figures (8) and (9) illustrate variation in the base convective heat transfer coefficient (h_b) with fin spacing (S) at different fin height (H) while heat input kept at 100 W for vertical fins and V-fins array respectively . The convection heat transfer coefficient (h_b) as a function of fin spacing at first increases up to a peak value with fin spacing (S) increasing and then decreases gradually because the average surface temperature (T_{Sav}) decreases down to a minimum with fin spacing (S) increasing and then increases gradually as shown in Figures (10) and (11). The optimum fin spacing ($S_{optimum}$) value maximizing base convection heat transfer coefficient (h_b) and minimizing average surface temperature (T_{Sav}) for all three fin height is about 20 mm for vertical fins and V-fins array configurations.

The experimental simulation data are correlated to a formula between average Nusselt number (Nu_{av}) versus Rayleigh number (Ra) for vertical plate and vertical fins array and compared with a previous works for same conditions as shown in Figures (12-14). A correlation is suggested to predict the average Nusselt number for vertical plate :

$$\mathrm{Nu_{av}} = 0.563\ (\mathrm{Ra})^{0.25} \quad \dots(22)$$

for $(4.3\times10^{6} < Ra < 4\times10^{7})$.

The squared correlation coefficients (R^2= 94.5%). The absolute percentage relative of error between suggested correlation and experimental data is ($e \leq 11\%$). Another experimental correlation is adopted for vertical fins array at optimum fin spacing ($S_{optimum}$= 20 mm) and fin height (H= 15 mm) :

$$\mathrm{Nu_{av}} = 0.0523\ \left(\mathrm{Ra}^{'}\right)^{0.75} \quad \dots(23)$$

for $(780 < Ra^{'} < 2460)$.

The squared correlation coefficients (R^2= 99%) , the absolute percentage relative of error between suggested correlation and experimental data is ($e < 3.5\%$). The relative error between the suggested correlation of vertical plate with other correlations is ($e < 10\%$). The agreement of present work is very good especially with McAdam's correlation ($e < 5.5\%$), also the relative error between the suggested correlation of vertical fins array with Abid correlation is ($e < 15\%$) at same conditions according to the Figures (13) and (14) .

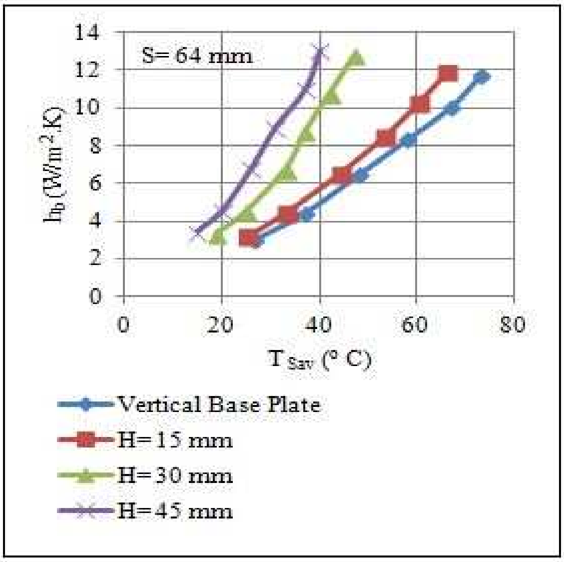


Fig. 3. Base convection heat-transfer coefficient versus average surface temperature for vertical plate and vertical fins array.

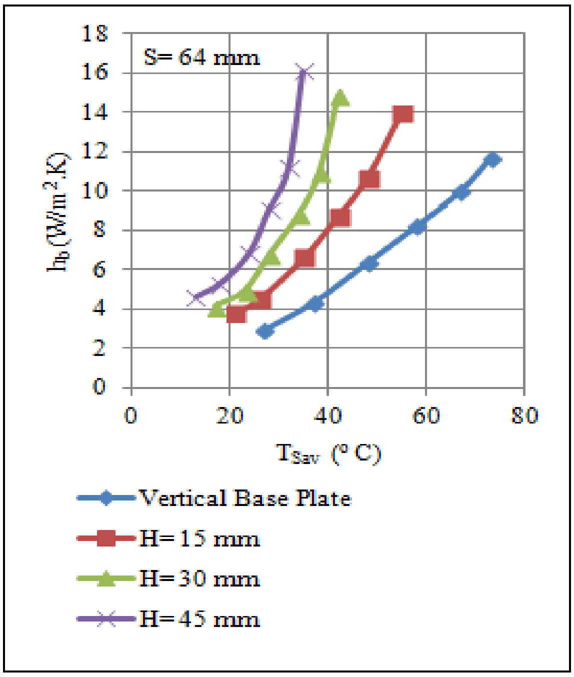


Fig. 4. Base heat transfer coefficient versus average surface temperature for vertical plate and V-fins array.

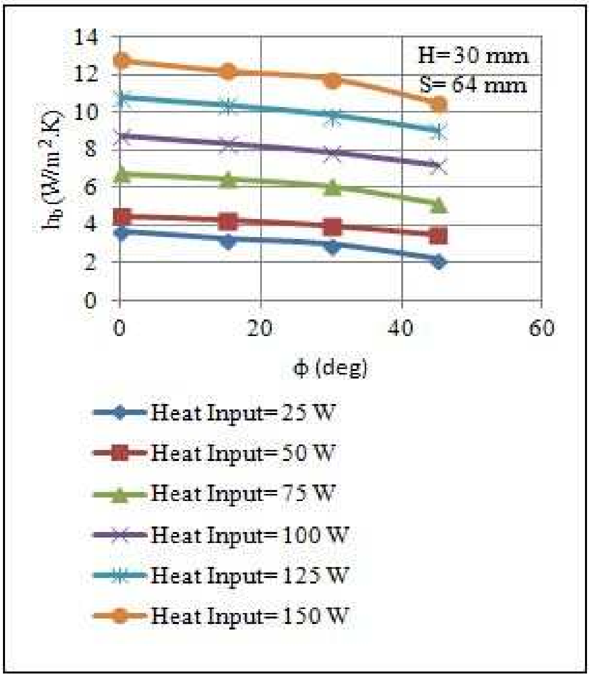


Fig. 6. Base heat transfer coefficient versus inclination angle for vertical fins array.

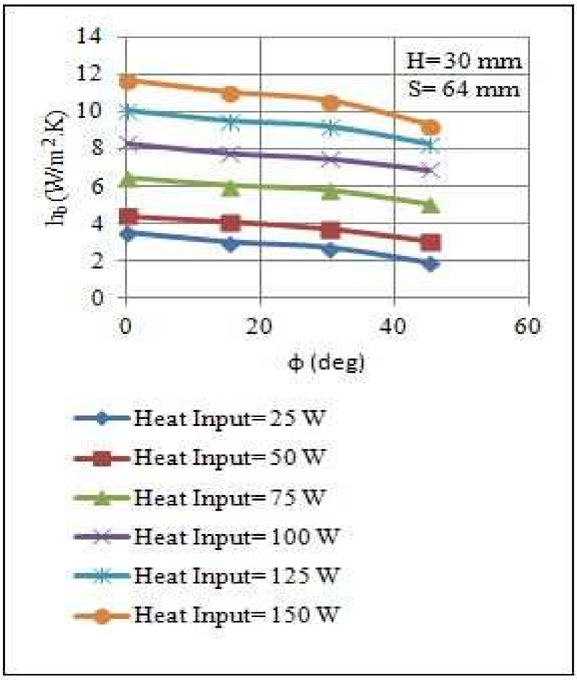


Fig. 5. Base heat transfer coefficient versus inclination angle for vertical plate.

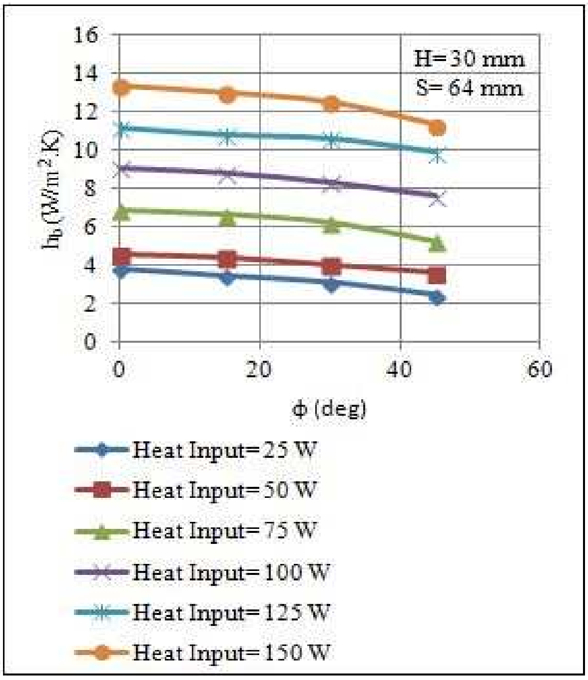


Fig. 7. Base heat transfer coefficient versus inclination angle for V-fins array.

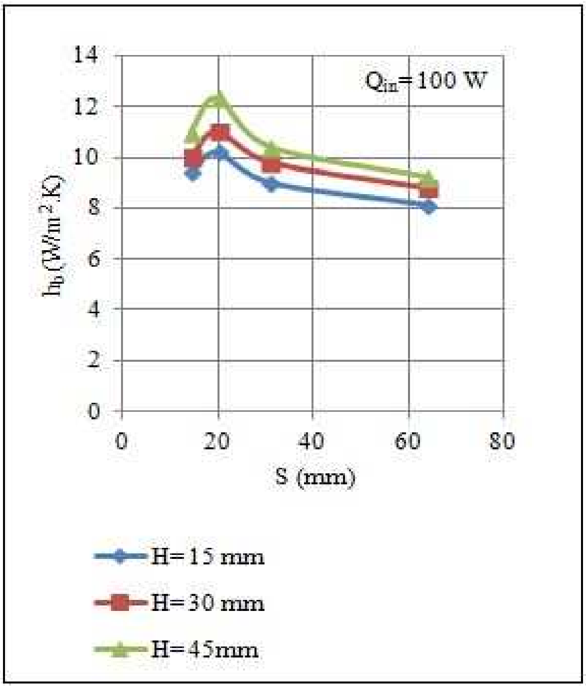


Fig. 8. Base heat transfer coefficient versus fin spacing for vertical fins array.

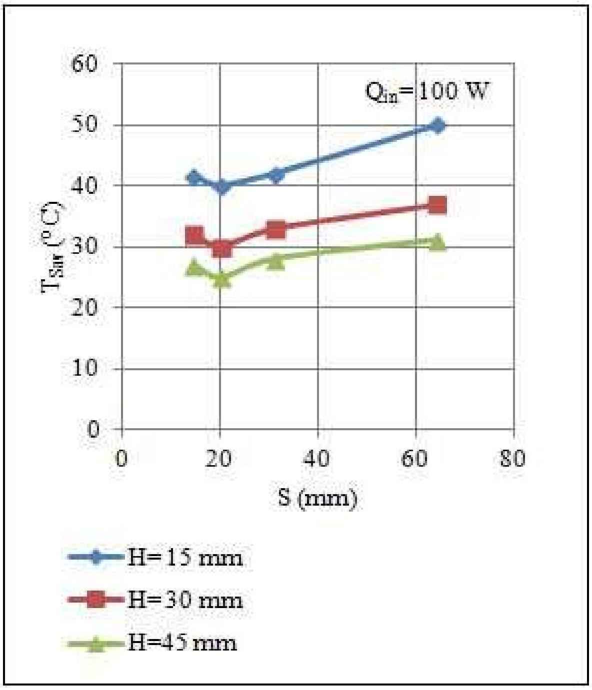


Fig. 10. Average surface temperature versus fin spacing for vertical fins array.

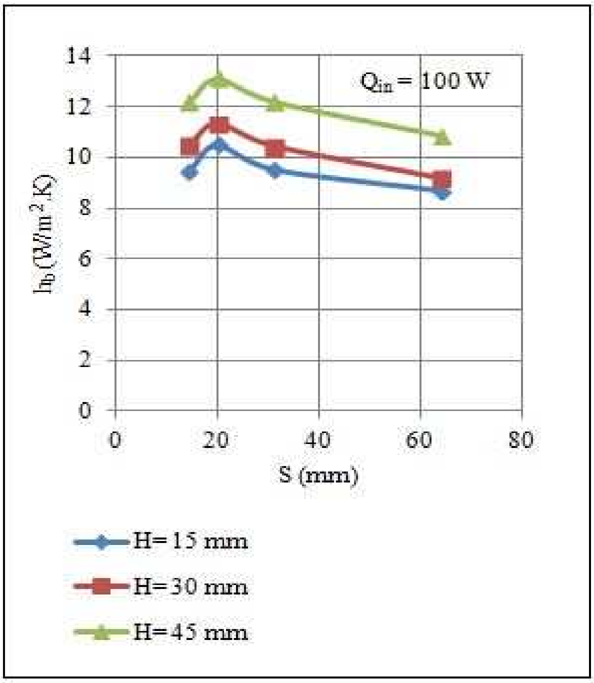


Fig. 9. Base heat transfer coefficient versus fin spacing for V-fins array.

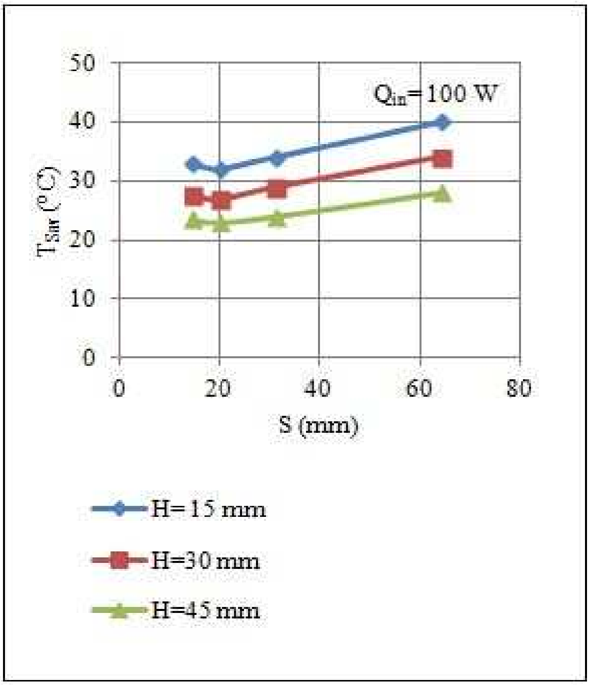


Fig. 11. Average surface temperature versus fin spacing for V-fins array.

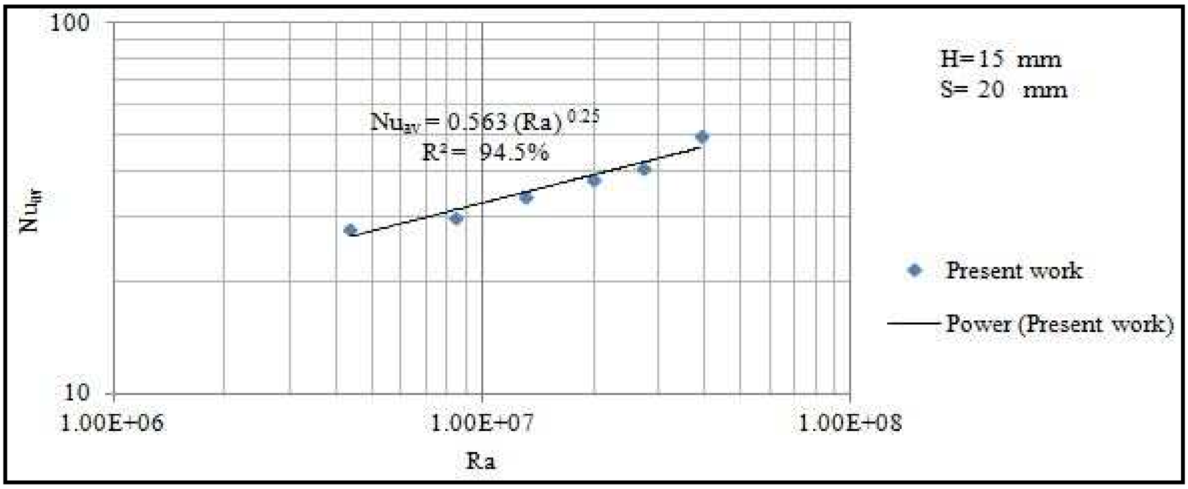


Fig. 12. Analysis of the experimental simulation data of present work for vertical base plate to the power function fits.

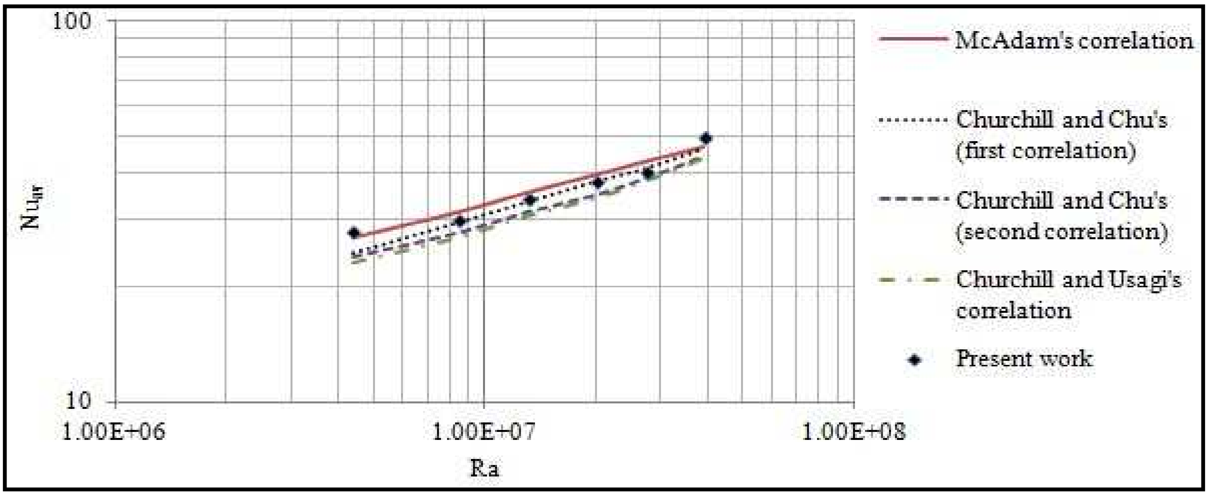


Fig. 13. Comparison of average Nusselt numbers (Nu_{av}) for vertical base plate with available empirical correlations .

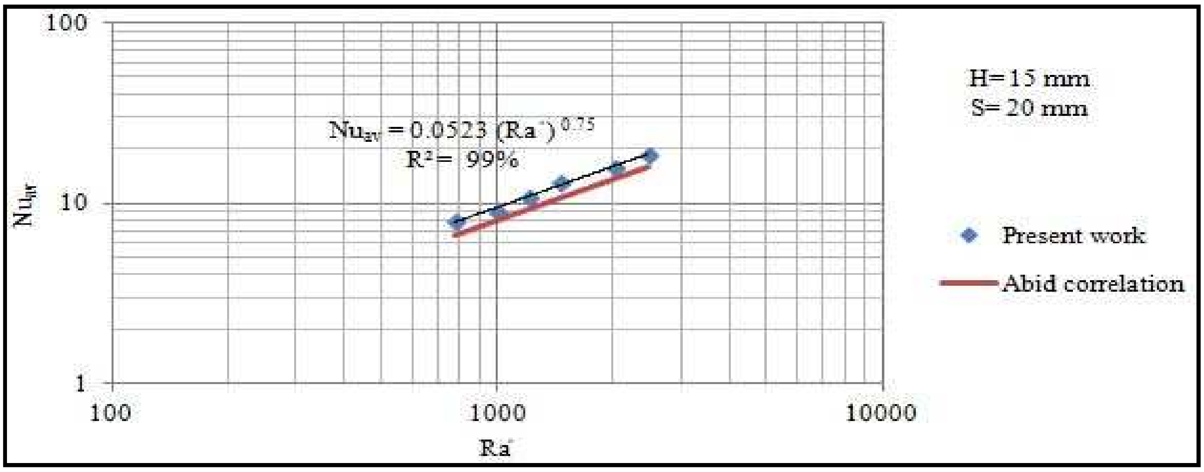


Fig. 14. Experimental simulation data of present work for vertical fins array compared with Abid correlation.

4. Conclusions

Natural-convective heat transfer from the heated vertical base plate, vertical fins array and V-shaped fins array are studied experimentally and focused on the determination of the geometrical configuration giving the best natural-convection heat transfer performance. The following conclusions can be drawn:

1- The type of V-fins array gave better convection heat transfer performance in term of base convection heat transfer coefficient about 20% bigger compared than vertical fins array.

2- As the height of fin increased, the average surface temperature decreased and the base convection heat transfer coefficient increased.

3- The base convection heat transfer coefficient decreases with inclinations increasing and heat input decreasing.

4- The optimum value of fin spacing is about 20 mm for vertical fins and V-fins arrays when maximizing base convection heat transfer coefficient.

5- Experimental correlations are suggested to predict the average Nusselt number for vertical plate and vertical fins array.

Nomenclature

A_b	Base area, (m^2)
A_S	Surface area of heat transfer, (m^2)
A_t	Total surface area of convection heat transfer from fin-arrays, (m^2)
F	Radiation shape factor, (--)
g	Gravitational acceleration, (m/s^2)
Gr	Grashof number, (--)
h	Convection heat-transfer coefficient, ($W/m^2.K$)
H	Fin height, (m)
I	Input current intensity, (A)
k	Thermal conductivity of material, (W/m.K)
L	Length / Height of vertical plate or fins array, (m)
L_C	The characteristics length of geometry, (m)
N	Number of fins, (--)
Nu	Nusselt number, (--)
Pr	Prandtl number, (--)
Q_C	Convection heat transfer rate, (W)
Ra	Rayleigh number, (--)
Ra'	modified Rayleigh number, (--)
S	Fin spacing, (m)
t	Fin thickness, (m)
T_A	Ambient temperature, (^oC)
T_S	Surface temperature, (^oC)
ΔT	Temperature difference, (^oC)
V	Voltage supplied, (V)
X	Thickness, (m)

Greek Letters

β	Volumetric coefficient of thermal expansion, (1/K)
ε	Emissivity of the surface, (--)
ϑ	Kinematic viscosity of the air, (m^2/s)
σ	Stefan-Boltzmann constant, ($\sigma = 5.67 \times 10^{-8}$ W/ $m^2.K^4$)
ϕ	Inclination angle of the plate with vertical position, (deg)

Subscript Symbols

av	Average
b	Base

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محاكاة عملية للحمل الحراري الطبيعي من صفيحة عمودية مزعنفة بزوايا ميل مختلفة

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الخلاصة

في هذا البحث، أجريت محاكاة عملية للتنبؤ بأداء أنتقال الحرارة بالحمل الطبيعي في الحالة المستقرة من صفيحة عمودية مزعنفة و مسخنة بزوايا ميل مختلفة وصفوف زعانف مختلفة الأشكال. أستخدم شكلان مختلفان من صفوف الزعانف على الصفيحة العمودية المسخنة هما مصفوفة الزعانف العمودية ومصفوفة الزعانف نوع-V وبأرتفاعات ومسافات بينية مختلفة. تم دراسة تأثير كل من زاوية ميل الصفيحة المعدنية وشكل مصفوفة الزعانف والمتغيرات الهندسية للزعنفة كأرتفاع الزعنفة والمسافة بين زعنفة وأخرى على التوزيع الحراري و معامل أنتقال الحرارة بالحمل و معدل رقم نيسلت . تم التوصل الى علاقات تجريبية من البيانات العملية بين معدل رقم نسلت ورقم ريليه للصفيحة العمودية ومصفوفة الزعانف العمودية. بينت النتائج أداء أفضل لأنتقال الحرارة بالحمل الطبيعي لمصفوفة الزعانف نوع-V ومعامل أنتقال حرارة أكبر بحدود 20% مقارنة بمصفوفة الزعانف العمودية. تم مقارنة نتائج المحاكاة العملية و العلاقات التجريبية للبحث المقدم مع بحوث سابقة وكان التوافق جيد .