Experimental Study of Free-Convection from Rectangular Fins Array on a Heated Horizontal Plate with Notch Effects

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Abstract

In this work , experimental investigation has been done for free-convection heat transfer from rectangular fins array on a heated horizontal base plate to surrounding air in the steady-state flow condition with rectangular notch portion effects. Five cases of fins arrays have been employed . One case without notch and other cases with rectangular notches for different percentages of aspect ratio area removal from fin . The horizontal base plate has been heated with various electrical supplied power values. Different number of fins and spacing have been used . The effect of notches from fins on average convection heat transfer coefficient and average Nusselt number at constant aspect ratio of fin height to fin length and varying heat inputs to the heating element have been discussed . The experimental results show that the performance of notched fins array in term of average convection heat transfer coefficient is 28% to 45% higher than unnotched fins array. The present experimental results have been compared with previously works . The results show a good a greement .

Keywords: Fins arra , notches , experimental investigation , horizontal base .

Introduction

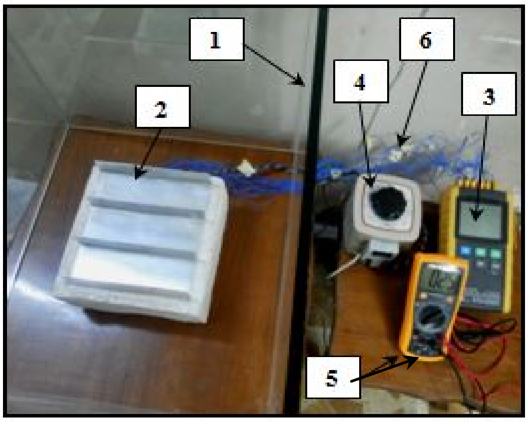
Fins or extended surfaces are used to enhancement performance and rate of heat transfer in a wide range of physical problems and engineering applications like radiators , condenser of refrigerators , compact heat exchangers , oil cooling systems , domestic convectors , thermal fluid systems , nuclear reactors , ground thermo-siphons , transformers , electrical motors , computer power supplies and electronic equipments [1-3]. Any small increase in heat dissipating rate from a fins of these systems or equipments decreases the power consumption and increases the life of them. The configurations of fins or fin arrays , geometrical parameters such as height , length , spacing of fin and fins modification by removing fin portion like perforated and notched fins effects on the convective heat transfer coefficient and heat dissipation rate . Yuncu and Anbar [1] submitted experimental study of free-convective from a fifteen sets of rectangular fin arrays on horizontal surfaces with different length and height of fins and heat inputs. They deduced experimental correlations for horizontal base plate with rectangular fins and without fins. Goshayeshi and Ampofo [2] used Fluent software version 6.3 to study natural-convection heat transfer over a vertical and horizontal plates using vertical fins array . They found that the vertical plate with vertical fins array gives best performance for cooling by natural-convection. Mahmoud et al. [3] investigated experimentally and computationally natural-convective from micro-fin arrays on horizontal microstructures. They used the micro-fin arrays with fin height ranging from 0.25 to 1.0 mm and fin spacing ranging from 0.5 to 1.0 mm. They found that the optimum design of fins array occurs at spacing 0.7 mm . Barhatte et al. [4] analyzed experimentally and computationally the natural convection heat transfer from a vertical rectangular fins array with and without notch. They used the commercially available CFD software, Fluent version 6.3 to computational analysis . They obtained a good agreement between experimental and computational results. Dixit et al. [5] investigated experimentally the free-convective heat transfer over a heated horizontal finned surface with and without notch. The performance of notched fins array was found 30% to 50% greater than fins array without notches as convective heat transfer coefficient. Wange and Metkar [6] performed an experimental and computational study for natural convection through inverted notched fin arrays by using commercially available CFD software, ANSYS Fluent version 12.0 for computational analysis. They discussed the effects of inverted notches on the performance of fins array and heat transfer rate. Dixit and Mishra [7] presented an experimental investigation of free-convection heat transfer over a horizontal rectangular fins array with inverted notch. They observed that the optimum fin spacing is in a band from 8 mm to 10 mm at 30% inverted notch . Tari and Mehrtash [8] studied theoretically natural-convective from finned heat sinks in a horizontal and inclined positions. They deduced the set of Nusselt number correlations covering a horizontal and inclination heat sinks with different inclination angles in upward and downward directions. They found a good agreement of the obtained numerical results with previous works. Wankhede and Meshram [9] presented an experimental study for natural and force heat transfer convective through a horizontal rectangular fins array with inverted notches. The effects of fin spacing , supplied power input , flow velocity and notches on the convection heat transfer coefficient were analyzed. Furthermore , the effects of inverted notch on the single chimney flow pattern were investigated. Lohar and Bhosale [10] studied the free and force-convection from horizontal rectangular fins array. They covered a wide range of height , spacing of fin , wind velocity and power input to obtained the optimum fin spacing that gives best performance of the heat transfer from fin arrays .

The objective of present work is to investigate and analyze experimentally the influence of notches on the performance and rate of steady-state free-convection heat transfer in term of average convection heat transfer coefficient and average Nusselt number. The present work has been carried out for a wide range of the geometrical parameters such as percentage of removal area (notched portion) from fin and fin spacing at fixed aspect ratio of fin height to fin length (H/L) with different supplied heat inputs .

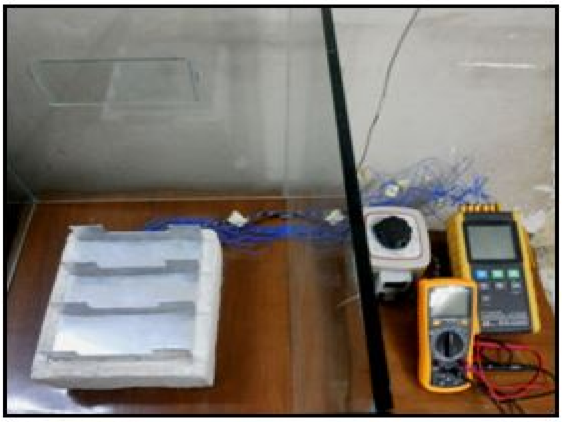
Experimental Setup

The experimental setup was shown photographically in Figures (1a and b). It consists of test section (Fins array assembly) , opened box ,digital voltage regulator and measurement devices like digital multi-meter , data logger thermometer with thermocouples wires . The test section involves the horizontal base plate manufactured from bright rolled aluminum to reduce the radiation heat loss with dimensions (180 mm length × 180 mm width × 2 mm thickness). The horizontal base is heated from a back surface by electrical heater wire is coiled around mica sheet and then is inserted between two symmetrical sheets of mica with same dimensions of the base plate and thin thickness of 0.5 mm to obtained uniform and homogeneous heating of the base plate and guaranteed the full insulated of wire heater. It's jointed with back surface of the horizontal plate using thermal super glue. The rectangular fins with notches and without notches are manufactured from same material of the base plate with dimensions (180 mm length × 60 mm height × 2 mm thickness) and fixed horizontally on the base plate. The assembly of fins array is housed in a 4 mm deep rectangular pocket milled on the top surface of a thermo-stone's slab (300 mm length × 280 mm width × 100 mm thickness) as shown in Figure (1c) . It's an excellent thermal insulator with a low thermal conductivity of 0.15 W/m . K to minimize the heat conduction loss. The whole assembly is placed in large box constructed of cast acrylic sheet of thickness 6 mm and opened from top face to provide free heat convection condition.

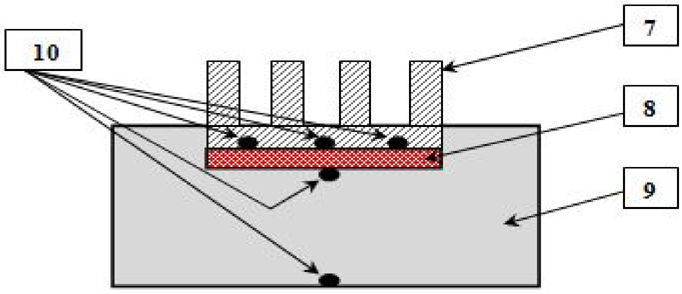
The surface temperatures of fins array are measured by nine type-K calibrated thermocouples are inserted at different suitable locations in the back face of horizontal plate. They are distributed by 3 rows and 3 columns with equal spaces. Other two same type-K calibrated thermocouples are used to measure the temperatures difference between two surfaces below the heating element to evaluate the heat conduction loss through a thermo-stone's block . Additional same calibrated thermocouple is used to measure the ambient temperature inside the box.



a. Fins array without notch



b. Fins array with notch



c. Section of the fins array assembly

1. Opened box **2.** Test section (fins array assembly) **3.** Data logger thermometer **4.** Digital voltage-regulator **5.** Digital multi-meter **6.** Thermocouple wires **7.** Fins array **8.** Heating element **9.** Thermo-stone's block **10.** Thermocouples locations

Figure 1: Photographs of the experimental setup and section of fins array assembly .

Experimental Processing and Calculations

Five configurations of rectangular fin arrays are used in the experimentations for the presented work. The first configuration of rectangular fins array is without notches and other configurations are with rectangular notches. The utilized different dimensions of notches and percentage of area removed (A_R) from fin are listed in Table (1). The fin length (L) is kept at 180 mm and fin height (H) is fixed at 60 mm in each configurations of fins array. Then, the aspect ratio is fixed at (H/L= 0.333). Different number of fins (N) like 4 , 6 , 11 , 16 and 21 are used. Then, fin spacing (S) is evaluated as follows :

$$S = \frac{W - Nt}{N - 1} \quad \text{... (1)}$$

Multi heater input levels namely 25 , 50 , 75 , 100 and 125 W are used. The surface temperature of fins array is measured by nine thermocouples. All readings of thermocouples are recorded at time intervals 45 minute up to steady-state condition.

The power input (Q_{in}) to the test section is transferred to the ambient by free-convection (Q_C) and radiation (Q_R) in addition to heat lost by heat conduction (Q_{Cd}) through a thermo-stone's slab. Then, free-convection heat transfer rate (Q_C) can be computed as :

$$Q_C = Q_{in} - Q_{Losses} \quad \text{... (2)}$$

The heat losses (Q_{Losses}) from the heating element is:

$$Q_{Losses} = Q_R + Q_{Cd} \quad \text{... (3)}$$

The radiation heat transfer rate (Q_R) is [11-13] :

$$Q_R = \varepsilon\ F_G\ \sigma\ A_S\ (T_{Sav}^4 - T_A^4) \quad \text{... (4)}$$

and the conduction heat transfer rate (Q_{Cd}) is calculated as [11-13] :

$$Q_{Cd} = \frac{\text{Thermal potential difference}}{\text{Thermal resistance}}$$

$$= \frac{\Delta T}{R_{th}} = \frac{k\ A_S}{X}\ \Delta T \quad \text{... (5)}$$

The radiation heat transfer (Q_R) from the heated horizontal base plate and fins array is evaluated to be less than 6% of electrical heat input (Q_{in}) for all cases. It is small because of the small value of emissivity (ε) of bright rolled aluminum used in manufacturing the base plate and fins which is 0.04 . The heat lost by conduction (Q_{Cd}) across thermo-stone's block is calculated to be about 3% .

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

$$Q_{in} = I\ V \quad \text{... (6)}$$

The heat losses (Q_{Losses}) are subtracted from electrical heat input (Q_{in}) , then equation (2) becomes :

$$Q_C = I\ V - Q_{Losses} \quad \text{... (7)}$$

The free-convection heat transfer rate (Q_C) from the fins array is computed according to Newton's equation of cooling as follows [11-14] :

$$Q_C = h_{av}\ A_S (T_{Sav} - T_A) \quad \text{... (8)}$$

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

$$h_{av} = \frac{I\ V - Q_{Losses}}{A_S (T_{Sav} - T_A)} \quad \text{... (9)}$$

where , T_{Sav} is the average surface temperature

$$T_{Sav} = \sum_{i=1}^{p} \frac{T_S}{p} \quad ...(10)$$

where, A_S is the surface area of heat transfer. It's evaluated in heat convection as follows :

For fins array without notches :

$$A_S = (N-1)SW + 2NHL \quad ...(11)$$

For fins array with rectangular notches

$$A_S = (N-1)SW + 2N(HL - A_n) \quad ...(12)$$

and A_n is the area removal (notched) from fin ,

$$A_n = L_n H_n \quad ...(13)$$

Then, the percentage of area removal from fin (A_R) is :

$$A_R = \frac{L_n H_n}{L H} \times 100 \quad ...(14)$$

The average Nusselt number (Nu_{av}) can be defined as [11-14]:

$$Nu_{av} = \frac{h_{av} S}{k} \quad ...(15)$$

All properties of air are computed corresponding to film temperature (T_f) . It calculated as an average of the ambient temperature (T_A) and average surface temperatures (T_{Sav}) of fins array as follows :

$$T_f = \frac{T_A + T_{Sav}}{2} \quad ...(16)$$

where, T_A and T_{Sav} in K .

Table 1: Cases of fin arrays geometry are used in experimental study.

Case No.	Dimensions of Rectangular Notch		Percentage of Area Removal A_R , %	Fins Number N	Fin Spacing S , mm
	Length L_n , mm	Height H_n , mm			
1	Without Notch		0 %	4	57.3
				6	33.6
				11	15.8
				16	9.8
				21	6.9
2	100	10	9.25 %	4	57.3
				6	33.6
				11	15.8
				16	9.8
				21	6.9
3	100	20	18.5 %	4	57.3
				6	33.6
				11	15.8
				16	9.8
				21	6.9
4	100	30	27.75 %	4	57.3
				6	33.6
				11	15.8
				16	9.8
				21	6.9
5	100	40	37 %	4	57.3
				6	33.6
				11	15.8
				16	9.8
				21	6.9

Results and Discussion

This work investigates experimentally the influence of rectangular notches of fins array on performance of free-convection heat dissipation as an average heat transfer coefficient and average Nusselt number at different percentages of area removal from fins , fin spacing and heat input when the aspect ratio (H/L) is constant .

Figure (2) illustrates the behavior of average convection heat transfer coefficient (h_{av}) with percentage of area removal (A_R) of the fin for ranging from 0% to 37% at different heat input ranges from 25W to 125W step 25W while fin spacing is kept at S= 15.6 mm. The average heat transfer coefficient (h_{av}) increases as the percentage of area notched increases because the notched surface area is removed from viewpoint of heat transfer from fin surface. It's observed that the average heat transfer coefficient (h_{av}) increases as heat input increases (Q_{in}) .

Figures (3) and (4) show variation in the average convection heat transfer coefficient (h_{av}) with fin spacing (S). Figure (3) shows the effect of fin spacing on (h_{av}) with changed heat input for 37% notched area fins array and without fins array. It's clear that as the fin spacing increases , the average convection heat transfer coefficient (h_{av}) increases. This increasing is sharp up to optimum fin spacing about 16 mm and then gradually rises because a more of fresh cold air enters between fins and contacted hot surfaces of fins array. The fresh air moves inwards between fins along single chimney profile and caused decreasing temperatures difference between heated surfaces of fins array and entered fresh air. This property was very clear in case of notched fins array by removed the non effective part from fin in form of the rectangular notch and gives good performance when fin spacing is optimum at S≃16 mm at aspect ratio of fin height to fin length (H/L) is 0.333. The same behavior of the average convection heat transfer coefficient (h_{av}) with fin spacing will be happened for different percentage of area removed when heat input is kept at Q_{in}= 125 W as shown in Figure (4) .

The average Nusselt number versus fin spacing for non-notched fins array is shown in Figure (5) with heat input (Q_{in}) as the parameter from 25W to 125W. It's observed that the average Nusselt number (Nu_{av}) increases as the fin spacing (S) increases. The increasing in fin spacing causes more flow of the fresh air between hot fins. It's also noted that the values of average Nusselt number are higher with higher supplied heat input.

Figure (6) presents a comparison of average Nusselt number for 37% notched and without notched fins arrays. It's observed that the free heat convection performance which represented by average Nusselt number (Nu_{av}) is about 45% higher in case of 37% notched area fins array than without fins array at the same heat input at Q_{in}= 125 W.

Figure (7) represents a comparison of average Nusselt number (Nu_{av}) for present experimental work for unnotched fins array at supplied power input Q_{in}= 125 W with Tari and Mehrtash correlation which is represented by [8] :

$$Nu_{av} = 0.0915(Gr'\,Pr)^{0.436} \quad \text{... (17)}$$

where ,

$$Gr' = [g\,\beta\,(T_{Sav} - T_A)\,S^3]\left(\frac{H}{L}\right)^{0.5}\left(\frac{S}{H}\right)^{0.38} / \nu^2 \quad \text{... (18)}$$

The result of comparison clears a good agreement .

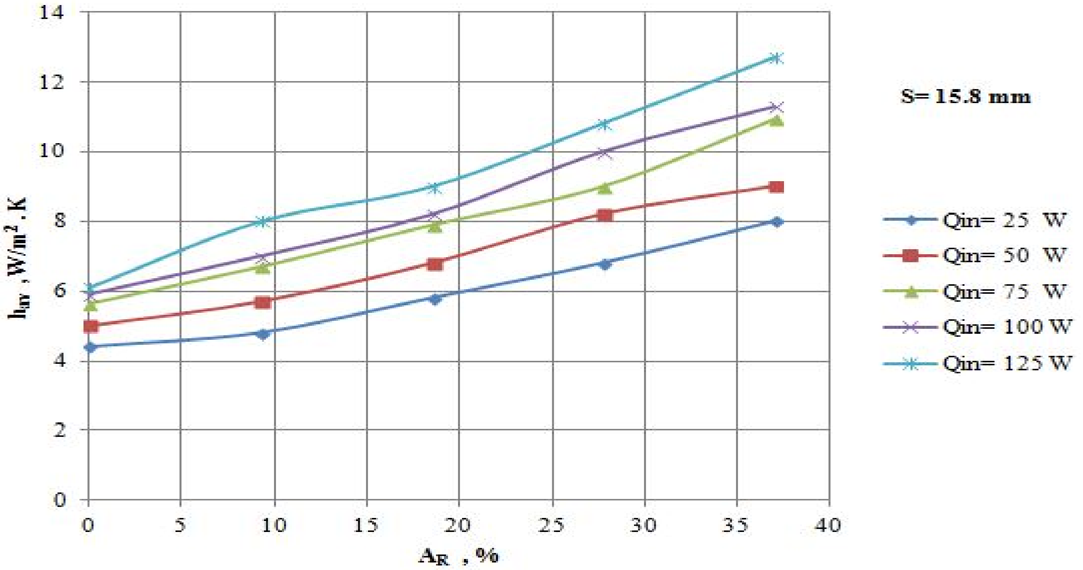


Figure 2: Average convection heat transfer coefficient versus percentage of area removal (notched) from fin.

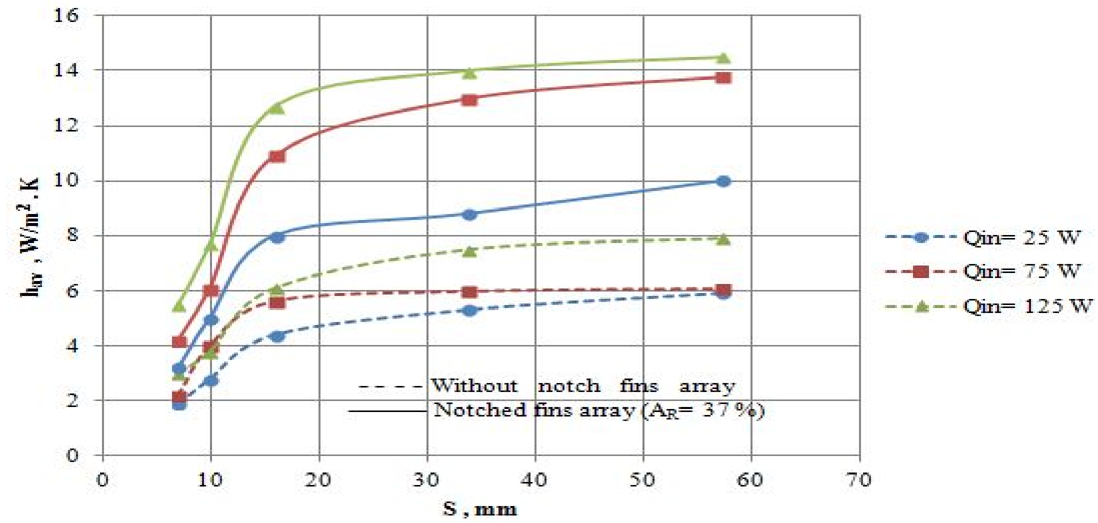


Figure 3: Average convection heat transfer coefficient versus fin spacing for different supplied heat input .

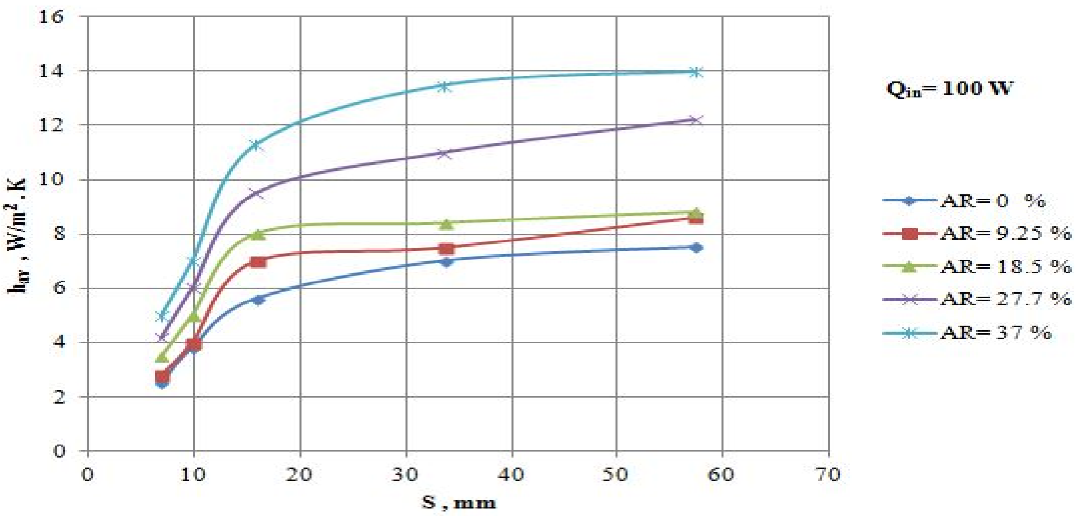


Figure 4: Average convection heat transfer coefficient versus fin spacing for different percentage of area removed (A_R) from fin .

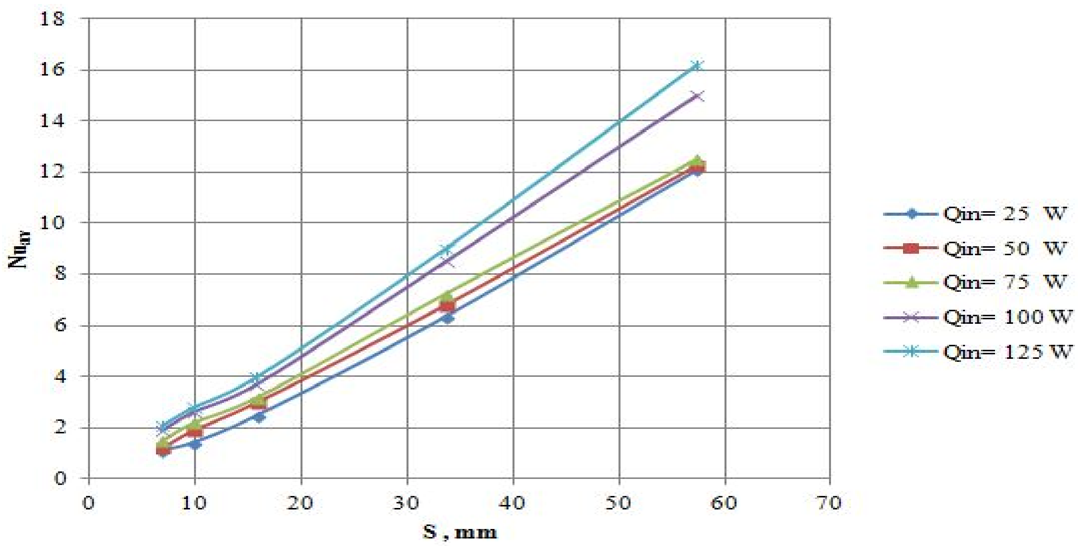


Figure 5: Average Nusselt number versus fin spacing for unnotched fins array at different heat input .

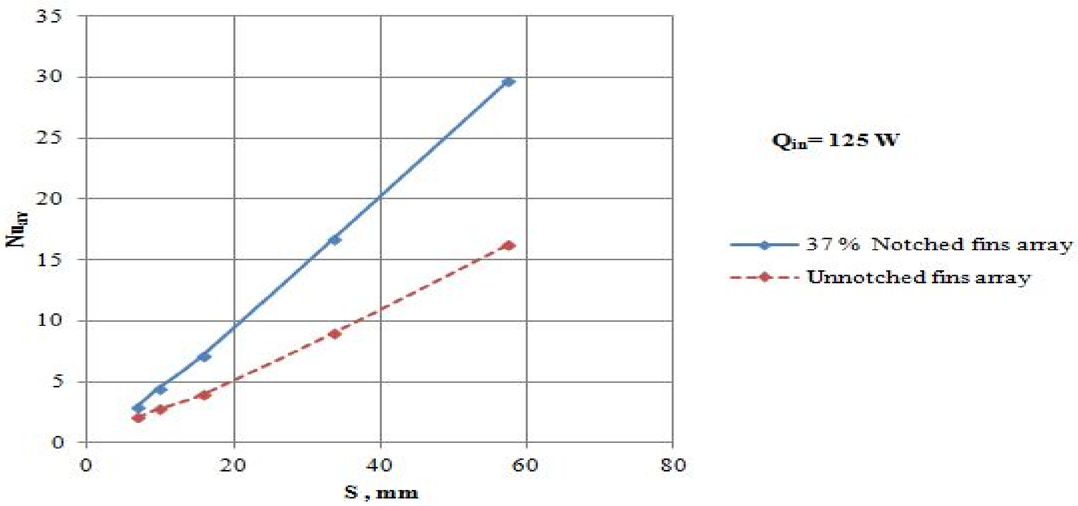


Figure 6: Comparison of average Nusselt number for notched and without notched fins array .

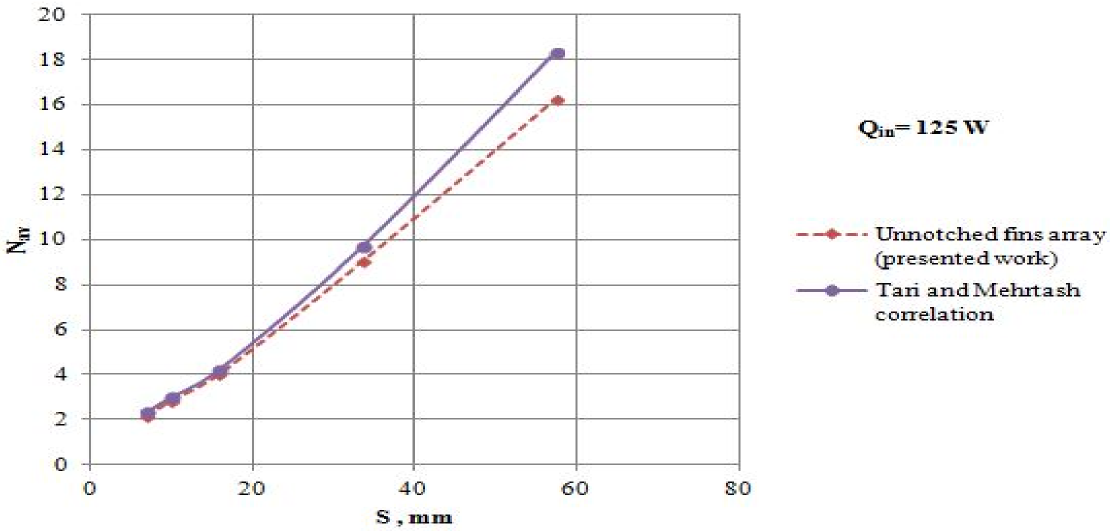


Figure 7: Comparison of average Nusselt number of the presented work for unnotched fins array with Tari and Mehrtash correlation at same conditions.

Conclusions

The main concluded remarks of present work are as follows:

1. Performance of the notched fins array in term of average convective heat transfer coefficient is 28% to 45% higher than that for fins array without notches at the sameconditions.
2. The average convective heat transfer coefficient increases as percentage of area removal (notched) from fins increases at constant fin spacing .
3. The average Nusselt number increases as fin spacing increases.
4. Average Nusselt number is about 45% higher in case of 37% notched fins array thanunnotched fins array.
5. The values of the average convection heat transfer coefficient and average Nusselt number increase with increasing in supplied heat input.

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Nomenclature

A_n Area removal (notched) from fin , (m^2)
A_R Percentage of Area removal , (%)
A_S Surface area of heat transfer , (m^2)
F_G Geometric function (view factor) , (-)
g Gravitational acceleration , (m/s^2)
Gr' Modified Grashof number , (-)
h Convection heat-transfer coefficient , ($W/m^2 . K$)
H Height of fin , (m)
H_n Height of notch , (m)
I Input current intensity , (A)
k Thermal conductivity (W/ m . K)
L Length of fins array (fin length) , (m)
L_n Length of notch , (m)
N Number of the fins in fins array , (-)
Nu Nusselt number , (-)
Pr Prandtl number , (-)
Q_C Convection heat transfer rate from fin arrays , (W)
Q_{Cd} Conduction heat transfer rate , (W)
Q_{in} Supplied heat input , (W)
Q_R Radiation heat transfer rate , (W)
Ra Rayleigh number , (-)
R_{th} Thermal resistance ($R_{th}= X / k A_S$) , (K/W)
S Spacing of fins , (m)
t Thickness of fin , (m)
T_A Ambient temperature inside the enclosure , (^oC)
T_f Film temperature , (^oC)
T_S Surface temperature , (^oC)
ΔT Thermal potential difference , (^oC)
V Voltage supplied , (V)
W Width of fins array , (m)
X Thermal insulation thickness , (m)

Greek Letters

β Volumetric coefficient of thermal expansion , (1/K)
ε Emissivity of the surface , (-)
σ Stefan-Boltzmann constant , ($\sigma = 5.67 \times 10^{-8}$ W/ $m^2 . K^4$)
$\boldsymbol{\nu}$ Kinematic viscosity , (m^2/s)

Subscript Symbols

av Average

دراسة تجريبية للحمل الحر من مصفوفة زعانف مستطيلة على قاعدة أفقية مسخنة مع تأثير الشقوق

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الخلاصـــة

يتناول هذا البحث دراسة تجريبية لأنتقال الحرارة بالحمل الحر من مصفوفة زعانف مستطيلة الشكل مثبتة على قاعدة أفقية مسخنة الى الهواء المحيط في حالة الجريان المستقر مع تأثير الشقوق (المساحات المزالة) المستطيلة الشكل في الزعانف. تم أستخدام خمس حالات لمصفوفات الزعانف . حالة واحدة بدون شقوق والحالات الأخرى بوجود شقوق مستطيلة الشكل في الزعنفة وبنسب مئوية مختلفة للمساحة المزالة. أستخدمت عدة مستويات من القدرة الكهربائية المجهزة لتسخين الصفيحة (القاعدة) الأفقية. أستخدم عدد مختلف من الزعانف والفراغات البينية لمصفوفة الزعانف. تم توضيح و مناقشة تأثير الشقوق (المساحة المزالة) في الزعنفة والمسافة البينية على معدل معامل أنتقال الحرارة بالحمل ومعدل رقم نيسلت بثبوت النسبة بين أرتفاع الزعنفة الى طول الزعنفة مع تغيير القدرة المجهزة لعنصر التسخين . أظهرت النتائج التجريبية أن أداء مصفوفة الزعانف بوجود الشقوق بدلالة معدل معامل أنتقال الحرارة بالحمل أعلى بنسبة 28% الى 45% مقارنة بمصفوفة الزعانف بدون الشقوق. تم مقارنة النتائج التجريبية للبحث المقدم مع بحوث سابقة وأظهرت النتائج توافق جيد.

الكلمات المفتاحية : مصفوفة زعانف , شقوق , دراسة تجريبية , قاعدة أفقية .