Experimental Study of Forced- Convection from Horizontal Rectangular Fins Array into Air Duct

Saad Najeeb Shehab

Department of Mechanical Engineering / College of Engineering/ Mustansiriyah University/ Baghdad
Email: saadnajeeb16@ uomustansiriyah.edu.iq

Abstract

In this work, an experimental study has been done to expect the heat characteristics and performance of the forced-convection from a heated horizontal rectangular fins array to air inside a rectangular cross-section duct. Three several configurations of rectangular fins array have been employed. One configuration without notches and perforations (solid) and two configurations with combination of rectangular-notches and circular-perforations for two various area removal percentages from fins namely 18% notches-9% perforations and 9% notches-18% perforations are utilized. The rectangular fins dimensions and fins number are kept constant. The fins array is heated electrically from the base plate with five different magnitudes of power-inputs. Five several air flow velocity into a duct are utilized. The influence of fin geometry, air flow velocity, Reynolds number and the surface heat flux on the heat-performance of forced heat convection have been simulated and studied experimentally. The experimental data indicates that the combination of 18% rectangular-notched and 9% circular-perforated rectangular fins array gave best forced heat performance in terms of average heat transfer coefficient about (25% - 45%) and (7% - 20%) compared than solid and 9% notches with18% perforations fins array respectively. Five empirical correlations to predict the average Nusselt number for the 18% notches with 9% perforations rectangular fins array at wide range of surface heat flux are deduced. The present data are compared with previous works and a good closeness in behavior is noticed.

Keywords: Fins array, circular-perforations, rectangular-notches, forced-convection, experimental study.

1. Introduction

The heat characteristics and performance enhancement is a very importance subject of thermal-engineering. In general, the heat transfer from heated surfaces are enhanced by increasing the coefficient of convection heat transfer between a heated surface and it's ambient or by fins and fins array to increase the surface area of heat transfer or by both methods. The convection heat dissipation occurs in a several engineering and industrial applications. The importance applications are cooling of electronic and microelectronic components, high power chips, transformers, electrical motors, solar collectors, systems of energy storage.

Toh and Ong [1] introduced an experimental investigation of forced heat convection from vertically fins array through a narrow duct with different air velocities. They suggested an empirical equation as a function of geometrical fin parameters and air velocities. Barhatte et al. [2] experimentally and numerically investigated and analyzed the natural heat convection over a rectangular fins array with and without notches. They utilized different notches types namely, rectangular, triangular and trapezoidal and multi heat inputs with constant dimensions of fins. They utilized the commercially available of fluent software version (6.3) to simulate the heat transfer from fins array to air ambient. They noted that more heat rate is transferred from triangular-notches fins array and found a very good agreement between experimental and

numerical results. Hossain et al. [3] experimentally investigated the natural and forced heat performance from an aluminum pin-fin arrays. They utilized seven pin fins with constant diameter of 8.2 mm and constant length of 70 mm was distributed in-line and various air flow rate. They concluded the forced convection heat transfer coefficient 43% greater than that free convection and they found the pin-fins increases the pumping power. Dhanawade et al. [4] experimentally studied the enhancement of forced convection heat transfer over a horizontal flat plate with three types of rectangular-fins array namely solid, square perforation and circular perforation. They used several sizes of perforations and different values of Reynolds number. They noted the size of perforation and Reynolds number has a larger effect on values of Nusselt number for both perforations type. Also, they observed the percentage of enhancement of the square-perforated fins array is more than that of fins array with circulated-perforated. Sivasubramaniyam [5] achieved an experimental investigation for forced circular pin-fins array with multi fin spacing and two different distributions like in-line and staggered in both directions of stream-wise and span-wise and different rate of air flow. He observed the heat transfer rate is larger in staggered distribution of fins with that of fins in-line distribution. Also, he showed the average convection heat transfer coefficient increases than air flow velocity and Reynolds number increase. Wankhede and Meshram [6] introduced an experimental investigation for free and forced-heat convection over a rectangular fins-array with and without inverted-notches in a horizontal position. The effects of fin spacing, surface heat flux, percentage of notched area and air-flow velocity on the characteristics and performance of convection heat dissipation were studied. They studied the influence of the inverted-notches on single-chimney flow pattern. They illustrated that the coefficient of convection heat transfer increases as percentage of notched area increases about 25% to 75% for cases of inverted notches fins array than that cases of fins array without inverted notches in natural and forced-heat convection. Baqir et al. [7] experimentally studied the enhancement heat transfer and reduction of friction factor from staggered perforated pins fins array into a rectangular duct for wide range of Reynolds number. They are studied three types of pin fins,

solid, horizontal-vertical perforations and horizontal-vertical-lateral perforations. They concluded the Nusselt number of pin fins includes horizontal-vertical perforations are 11% larger than those of solid pins and with horizontal-vertical-lateral perforations pins are 21% greater than those of solid pin fins. Mousa [8] presented an investigation of air flow characteristics and forced heat convection around different extended surfaces shapes like triangular, rectangular and wavy. He found the turbulence intensity of rectangular-extended greater than that of triangular and wavy-extended for same dimensions. Mane S.N. and Mane S.S. [9] presented a review on enhancement of forced heat convection from a several configurations of the notches fins array like rectangular, circular, triangular, and trapezoidal. They concluded that more heat transfer is from triangular-notches fins array than that other configuration of notches fins array. Also, they noted the size of notch is very effects on heat transfer enhancement. Shehab [10] experimentally simulated convection heat transfer over rectangular and V-fins array. He illustrated that the V-fins array gave better heat performance about 20% larger than rectangular fins array. Shehab [11] experimentally studied the natural heat convection over rectangular fins arrays on a horizontal heated aluminum base-plate to ambient air with and without rectangular-notches portions influence. He investigated multi parameters such as fins number, percentage of notch area from a fin and surface heat flux on the coefficient of convection heat transfer and Nusselt number. He observed that the heat convection performance for notch-fins array as coefficient of heat convection is 28% to 40% larger with solid-fins array. Mageswaran et al. [12] experimentally investigated the forced convection over perforated square pin fins array with staggered distribution inside a duct. They used five different numbers of perforations and three different diameters of perforations for pin fin. They concluded that the number and diameter of perforations effects on convection heat performance. Also, they found the best heat performance at the optimum perforated diameter (4 mm) and perforated number (3).

The aim of present work is to introduce an experimental study for the forced heat convection over a heated horizontal rectangular fins array to ambient into a rectangular cross-section air duct with nozzle-diffuser ends utilizing three different configurations of

rectangular-fins arrays, one configuration is solid and two configurations with combination of rectangular-notches and circular-perforations for different area removal percentages from fin. Also, the enhancement of forced heat convection using combination of notched and perforated fins. The influence of air flow velocity and surface heat flux are taken into consideration and studied.

2. Experimental Test-Rig

The experimental test-rig is designed and fabricated to cover the tests of present work as shown schematically and photographically in Figures (1a to 1c). It consists of air duct, rectangular fins array, air fan, digital voltage regulator and instruments of measurement namely, digital voltage-current reader, metal vane anemometer model (YK-80AM) to measure the inlet velocity of air flow in a duct, twelve-channels data logger thermometer and K-type of thermocouples wires.

The rectangular fins array is manufactured from bright rolled aluminum plate as one piece with base-plate by CNC machine with dimensions of (130 mm length × 130 mm width × 3 mm thickness of base plate). It is electrically heated from a lower surface of base-plate using electrical heater wire is coiled around a thin sheet of mica and then is putted between two layers of mica have same dimensions of base-plat with thickness of 0.4 mm to obtain an homogenous heating of fins array base-plate and ensured the complete insulated of electrical heater.

The dimensions of rectangular fins are (130 mm length × 26 mm height × 3 mm thickness). The fins array is housed in a 130 mm length × 130 mm width × 4.5 mm depth rectangular-hole

milled on the upper surface of thermo-stone's slab (200 mm length × 200 mm width × 100 mm thickness) to obtain a best thermal insulation and minimize the heat conduction losses from sides and lower of base-plate of fins array.

The whole assembly is placed in a halfway of rectangular cross-section air duct with nozzle-diffuser ends. The air duct is manufactured from wood with thickness of 15 mm and it's of 1000 mm long with an internal width (*w*) of 230 mm and internal height (*b*) of 100 mm, and the total length of duct with nozzle and diffuser ends is 1750 mm. The front face of air duct is made of 6 mm thickness clear acrylic sheet. Multi-velocities air fan is fitted at the diffuser end of duct to provide a forced heat convection condition.

The surface temperatures of the fins array are measured using five type-K calibrated thermocouples are fixed at various suitable positions in the lower surface of base-plate (four in corners and one in the center of base-plate). Another three same thermocouples are utilized to measure the temperatures difference at lower and sides of fins array base-plate to evaluate the heat conduction losses through a thermo-stone's slab as shown in Figure (2). Plus two thermocouples to record the inlet and outlet air temperatures inside a duct.

The conditions of steady state for all tests of present work are attained from 45 to 90 minutes depending on the electrical power input. The readings of thermocouples have been recorded when the temperatures difference between two readings is $\pm \leq 0.5$ °C within ten minutes approximately. Then, the surface temperatures of base-plate of fins array, air temperature into air duct, the voltage, and the current intensity passed to electrical wire heater are registered.
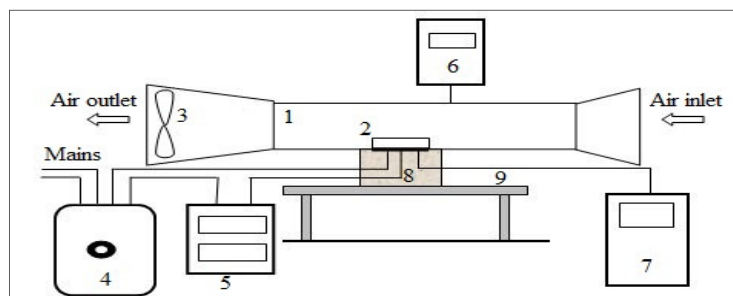
a. Sketch of an experimental test-rig.
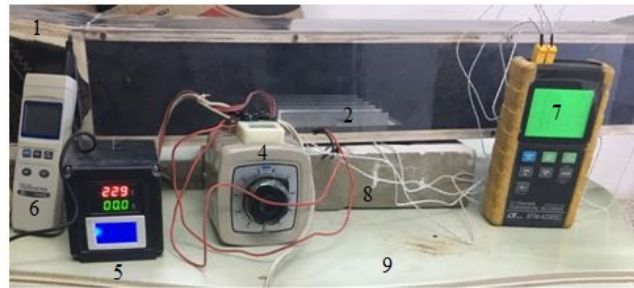
b. Photograph of an experimental test-rig.
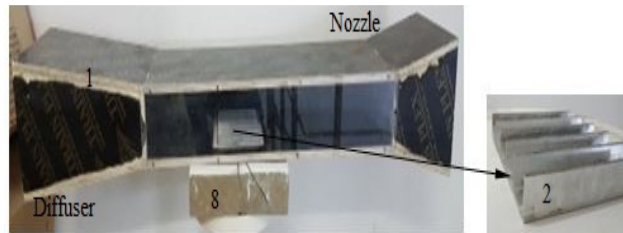
c. Photograph of an air duct.

1. Air duct 2. Rectangular fins array 3. Air fan 4. Digital Voltage regulator 5. Digital voltage-current reader 6. Metal vane anemometer 7. Twelve-channels data logger thermometer 8. Thermo-stone's slab 9. Flat table.
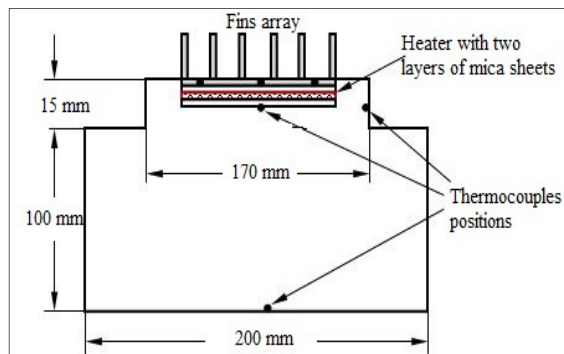
Fig. 1. Experimental test-rig.

Fig. 2. Schematic diagram of fins array assembly with thermocouples positions.

3. Calculation Procedures

The present work is carried out for three different configurations of rectangular fins arrays namely, solid, with combination of 18% rectangular-notch and 9% circular-perforations and with combination of 9% rectangular-notch and 18% circular-perforations as shown in Figure (3). The rectangular fin dimensions are fin length (L)= 130 mm, fin height (H)= 26 mm and fin thickness (t)= 3 mm and fins number (N)= 6 are kept constant for all configurations of fins array. Table (1) illustrate the dimensions of fins arrays utilized. Five different of surface

heat fluxes (q) values namely, 1479, 2958, 4437, 5917 and 7396 W/m^2 are utilized.

The electrical power input (Q_i) to the wire heater is evaluted as a product of voltage (V), current (I):

$$Q_i = V\, I \qquad \qquad ...(1)$$

It is transformed to heat energy and it transferred from a rectangular fins array to ambient air inside the duct by forced heat convection (Q_{cv}) plus heat losses due to heat radiation (Q_r) and heat conduction (Q_{cd}) lower and besides fins array. Hence,

$$Q_i = Q_{cv} + Q_r + Q_{cd} \qquad ...(2)$$

The loss of heat radiation between surfaces of polished rolled aluminum fins array with emissivity of $\varepsilon = 0.05$ and the air temperature inside a duct (T_∞) is computed from Stefan-Boltzmann law. The losses of heat conduction (Q_{cd}) from lower and sides of fins array base-plate across thermo-stone's slab (200 mm length × 200 mm width × 100 mm thickness) with thermal conductivity of k_i= 0.15 W/m.K are calculated using Fourier's law [13, 14].

Then, the forced-convection heat transfer (Q_{cv}) can be evaluted as follows:

$$Q_{cv} = V\, I - (Q_r + Q_{cd}) \qquad ...(3)$$

Also, it is can be evaluted from Newton's equation of cooling as [13, 14]:

$$Q_{cv} = h_{av}\, A_s (T_{sav} - T_\infty) \qquad ...(4)$$

The average forced-convection heat transfer coefficient (h_{av}) can be evaluted as:

$$h_{av} = \frac{V\,I - (Q_r + Q_{cd})}{A_s(T_{sav} - T_\infty)} \qquad \dots (5)$$

$$T_\infty = (T_i + T_o)/2 \qquad \dots (6)$$

$$T_{sav} = \frac{\sum_{i=1}^{p} T_{si}}{p} \qquad \dots (7)$$

where, T_i and T_o are the inlet and outlet air temperatures respectively, T_s is the local surface temperature.

A_s is the surface area of rectangular fins array, and it computed as follows:

- Solid rectangular fins array,

$$A_s = (L^2 - NtL) + 2NHL \qquad \dots (8)$$

- Combination of rectangular-notches and circular-perforations rectangular fins array,

$$A_s = (L^2 - NtL) + 2N\big[HL - (A_n + nA_p)\big] \qquad \dots (9)$$

where,

L is the fin length (base-plate side), N is the fins number, t is the fin thickness, H is the fin height, n is the perforations number in one fin, A_n is the notch area and A_p is the perforation area,

$$A_n = L_n H_n \qquad \dots (10)$$

$$A_p = \frac{\pi}{4} d^2 \qquad \dots (11)$$

The average Nusselt number (Nu_{av}) based on the hydraulic diameter of the air duct (D_h) as characteristics length can be calculated as [4, 15]:

$$Nu_{av} = \frac{h_{av}\,D_h}{k_a} \qquad \dots (12)$$

The Reynolds number (Re) for the flow into a duct can with the hydraulic diameter (D_h) be expressed as [15]:

$$Re = \frac{u\,D_h}{\nu} \qquad \dots (13)$$

and the hydraulic diameter of rectangular cross-section air duct is [15]:

$$D_h = \frac{2\,w\,b}{(w + b)} \qquad \dots (14)$$

where,

u is the velocity of air flow in a duct, w is the internal width of duct and b is the internal height of duct.

The air properties used in experimental calculations are taken at film temperature (T_f). It's an approximation to temperature of the fluid into a convection boundary layer and evaluted as the arithmetic mean of average surface temperature (T_{sav}) and ambient air temperature inside a duct (T_∞) in (K) [15].
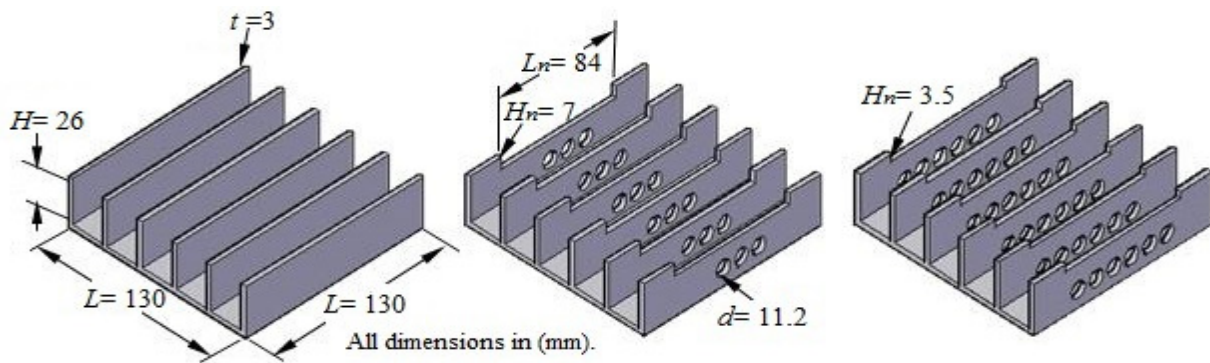
The percent error (%e) is the ratio of absolute error to the actual value, and computed as:

$$\%e = \frac{|V_{exp} - V_{cor}|}{V_{exp}} \times 100 \qquad \dots (15)$$

The mean relative squared-errors (*MRSE*) is evaluted as:

$$MRSE = \left[\frac{1}{j-1}\sum_{i=1}^{j}\left(\frac{V_{exp} - V_{cor}}{V_{exp}}\right)^2\right]^{1/2} \qquad \dots (16)$$

where, (V_{exp}) and (V_{cor}) are experimental and correlated values respectively, j is the data number.

a. solid b. 18% rectangular-notches with 9% circular-perforations c. 9% rectangular-notches with 18% circular-perforations.
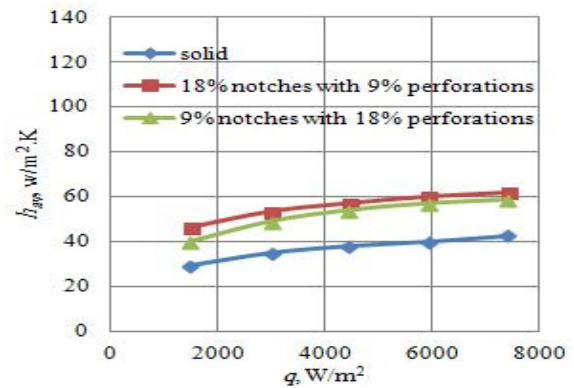
Fig. 3. Configurations of rectangular fins array.

Table 1,
Dimensions of fins array studied.

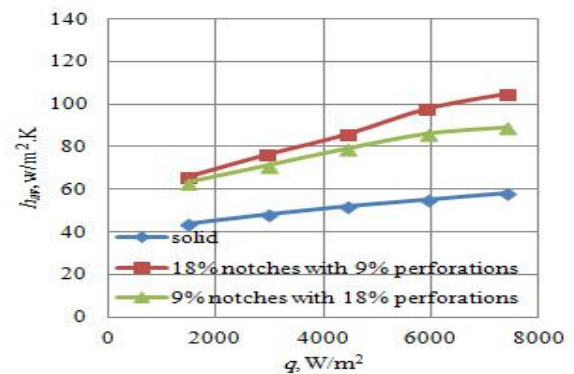| Fins array Configuration	Rectangular fin dimensions ($L \times H \times t$) mm	Fins number	Rectangular-notch dimensions ($L_n \times H_n$) mm	Circular-perforations dimensions($d$mm$\times n$)	Surface area (A_s) m^2
Solid	130× 26× 3	6	---	---	0.0°512
18% rectangular notches-9% circular perforations	130× 26× 3	6	84× 7	11.2× 3	0.0445
9% rectangular notches-18% circular perforations	130× 26× 3	6	84× 3.5	11.2× 6	0.0445

4. Results and Discussion

The present work simulates experimentally effect of the fins configurations (solid, with combination of rectangular-notches and circular-perforations), the percentage of area removal from fin, surface heat flux and the air flow velocity on forced-convection heat performance of a heated horizontal fins array inside air duct.
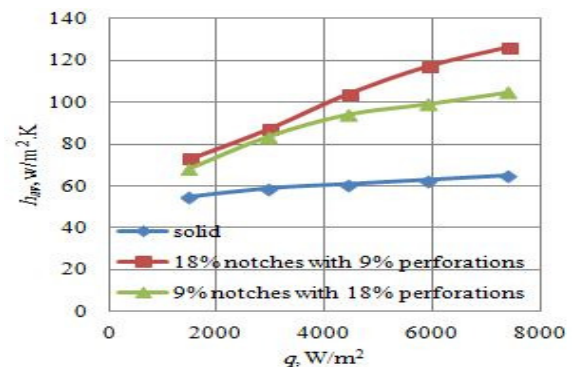
Figures (4a to 4c) illustrate behavior of the average forced heat convection coefficient versus surface heat flux for different configurations of rectangular fins array and different air velocities (u) ranging from 1.2 m/s to 3.6 m/s. They are clear that the average heat transfer coefficient (h_{av}) gradually increases as surface heat flux (q) increases. Also, the average heat-transfer coefficient (h_{av}) increases with increasing air flow velocity in the duct. Additionally, values of the average forced heat convection coefficient for 18% notches with 9% perforations rectangular fins array are larger about 25% to 45% than that solid fins array and larger about 7% to 20% than that 9% notches with 18% perforations rectangular fins array because of the combination of notches and perforations allow more fresh air comes inwards between fins channels and due to large increases in convection heat transfer coefficient. The notch of fin is removed from a non-effective part near of the stagnation zone of fin, while the perforations help to increase the turbulence of air flow into a duct and this, cause increases in forced-convection heat transfer rate from fins array surfaces.

a. Air flow velocity, u= 1.2 m/s

b. Air flow velocity, u= 2.4 m/s

c. Air flow velocity, u= 3.6 m/s

Fig. 4. Average forced-convection heat transfer coefficient versus surface heat flux for three configurations of rectangular fins array at five different air flow velocities, (u= 1.2, 2.4 and 3.6 m/s).
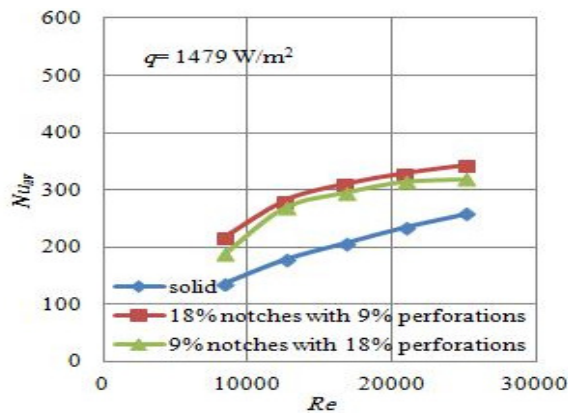
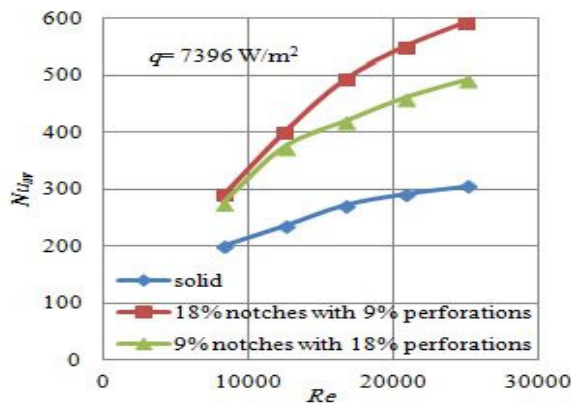**Fig. 5. Average Nusselt number versus Reynolds number for surface heat flux, $q= 1479$ W/m^2.**

Fig. 6. Average Nusselt number versus Reynolds number for surface heat flux, $q= 7396$ W/m^2.

Figures (5) and (6) show the behavior of average Nusselt number (Nu_{av}) against Reynolds number (Re) for solid, 18% notches with 9% perforations and 9% notches with 18% perforations rectangular fins arrays at surface heat flux, $q= 1479$ and 7396 W/m^2. It is observed that the average Nusselt number increases greatly with increasing Reynolds number for all cases studied because the surface temperatures decrease with increasing air flow velocity into a duct. Also it's clear that the average Nusselt number values for 18% notches with 9% perforations rectangular fins array are larger than those of solid and 9% notches with 18% perforations fins arrays. Hence, the best fins array is 18% notches with 9% perforations because the combination of notches and perforations which increases the forced-convection heat dissipation rate and turbulence of flow into an air duct and allow more cold air enters between fins channels and contacted the fins array hot surfaces. Also, increasing the removed area of non-effective part from 9% to

18% of fins area improves the forced-convection heat performance. Furthermore, it's observed that the average Nusselt number improves about 15% to 42% when the surface heat flux increases because the average heat convection coefficient increases with percent reach about 45% larger than those solid fins array.

Figure (7) appears five correlations between average Nusselt number and Reynolds number for 18% notches with 9% perforations rectangular fins array for a multi-levels of surface heat flux (q) from 1479 W/m^2 to 7396 W/m^2. It is clear that the average Nusselt number increases with increasing Reynolds number. The following five empirical correlations are concluded:

$$Nu_{av} = 5.3475(Re)^{0.4145} \qquad ... (17)$$
$$Nu_{av} = 4.2607 \, (Re)^{0.4537} \qquad ... (18)$$
$$Nu_{av} = 1.9742(Re)^{0.5459} \qquad ... (19)$$
$$Nu_{av} = 1.1661(Re)^{0.611} \qquad ... (20)$$
$$Nu_{av} = 0.7885(Re)^{0.6578} \qquad ... (21)$$

Figures (8a and 8b) show a comparisons of the correlated average Nusselt numbers (Nu_{av-cor}) with those experimental (Nu_{av-exp}) for 18% notches with 9% perforations rectangular fins array at surface heat fluxes, $q= 1479$ W/m^2 and 7396 W/m^2 using Eqs. (17) and (21) respectively. They are appear that the percentage error ($e\%$) of the experimental values situated between +10% and -10% of correlated values for surface heat fluxes, $q= 1479$ W/m^2 and between +12% and -10% of correlated values for surface heat fluxes, q= 7396 W/m^2, and the mean relative squared-errors ($MRSE$) are 0.145 and 0.187 respectively.

Furthermore, the test is carried out on a smooth air duct using the same base-plate without fins array. Then, the following empirical correlation is concluded:

$$Nu_{av} = 0.1522 \, (Re)^{0.6775} \qquad ... (22)$$

and compared with Dittus and Boelter [1°], and Dhanawade et al. [4] correlations:

$$Nu_{av} = 0.024 \, (Re)^{0.8} \, (Pr)^{0.4} \qquad ... (23)$$
$$Nu_{av} = 0.015 \, (Re)^{0.816} \qquad ... (24)$$

This is valid for turbulent flow in a rectangular smooth air duct as shown Figure (9). It can be seen a good similarity in the behavior, but the present results gave values of average Nusselt number higher those done in Dittus and Boelter and Dhanawade et al. correlations because the different in duct dimensions, air flow velocity and surface heat flux levels.
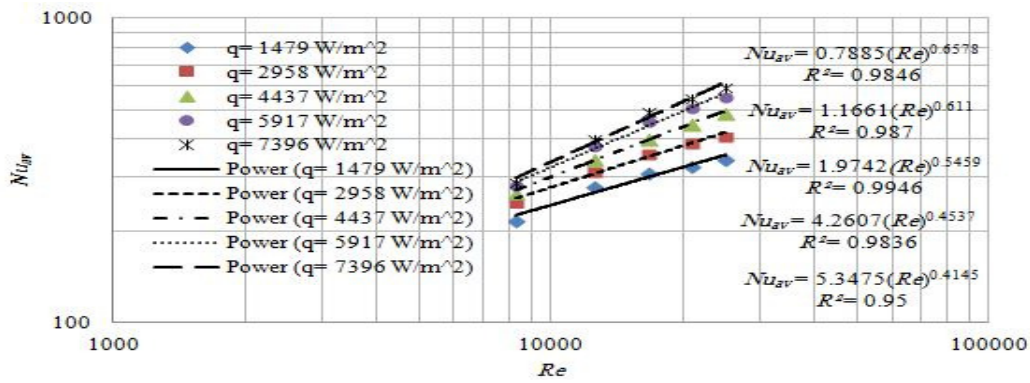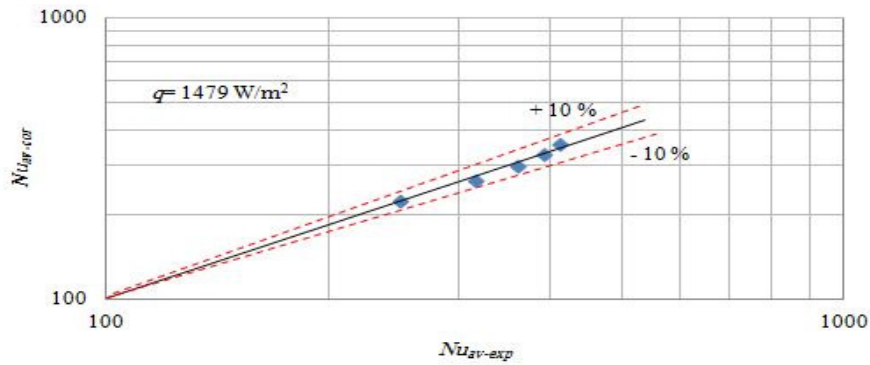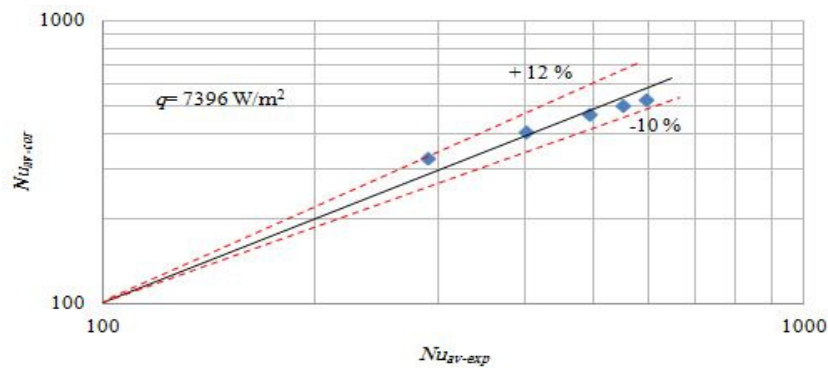
Fig. 7. Analysis of present data for 18% notches-9% perforations rectangular fins array with concluded empirical correlations.

a. Surface heat flux, q= 1479 W/m²

b. Surface heat flux, q= 7396 W/m²

Fig. 8. Comparison of the correlated average Nusselt numbers (Nu_{av-cor}) with those of experimental (Nu_{av-exp}) for 18% notches-9% perforations rectangular fins array.
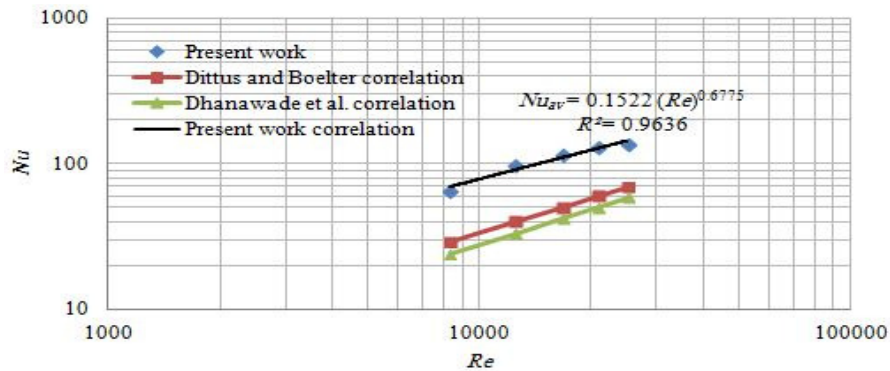

Fig. 9. Comparison of the present work for a smooth duct with Dittus and Boelter [1°] and Dhanawade et al. [4] correlations.

5. Conclusions

The present work has achieved an experimental study of an electrically heated horizontal rectangular fins array inside an air duct subjected to forced-convection. Three configurations of rectangular fins namely, solid, combination of 18% rectangular-notches with 9% circular-perforations and combination of 9% rectangular-notches with 18% circular-perforations are utilized with constant fins number and dimensions. The important conclusions can be drawn:

1. The rectangular fins array includes combination of 18% notches with 9% perforations gave best forced heat performance in terms of average heat convection coefficient about (25% to 45%) and (7% to 20%) larger than those solid and 9% notches with 18% perforations fins array respectively.

2. Increasing of the removed area of non-effective part (notches) from 9% to 18% of fins area improves the heat characteristics and gave maximum values of average heat transfer coefficient (hav-max= 126 W/m2. K) and Nusselt number (Nuav-max=592).

3. The average Nusselt numbers increases strongly from 34% to 48% approximately with increases Reynolds numbers for all studied rectangular fins arrays.

4. The average Nusselt number improves clearly about %15 to 42% with increasing surface heat flux for all studied cases.

5. Five empirical correlations to predict the average Nusselt number for the 18% notches with 9% perforations rectangular fins array at wide range of surface heat flux are concluded.

Nomenclature

A_n notch area, (m²)
A_p Perforation area, (m²)
A_s surface area of heat transfer, (m²)
b internal height of air duct, (m)
d diameter of perforation, (m)
D_h hydraulic diameter of air duct, (m)
h forced-convection heat transfer coefficient, (W/m².K)
H fin height, (m)
H_n notch height, (m)
I input current, (A)
k_a thermal conductivity of air, (W/m.K)
L fin height (base-plate side), (m)

L_n notch length, (m)
N number of fins
n perforations number
Nu Nusselt number
q surface heat flux, (W/m²)
Q_{cd} loss of heat conduction, (W)
Q_{cv} rate of forced-convection heat transfer, (W)
Q_i electrical power input, (W)
Q_r loss of heat radiation, (W)
Re Reynolds number
t fin thickness, (m)
T_∞ air temperature into a duct, (°C)
T_f film temperature, (K)
T_i inlet air temperature, (°C)
T_o outlet air temperature, (°C)
T_s surface temperature, (°C)
u velocity of air flow, (m/s)
V voltage, (V)
w internal width of air duct, (m)

Greek Letters

v kinematic viscosity, (m²/s)

Subscript Symbols

av average

6. References

[1] Toh, K.C. and Ong, Y.K. "Forced Convection Heat Transfer from Vertically Stacked Fin Arrays with a Shrouding Wall ", International Communications in Heat and Mass Transfer Journal, Vol. 19, Issue 3, pp.373-384, 1992.

[2] Barhatte, S.H., Chopade, M.R. and Kapatkar, V. N., "Experimental and Computational Analysis and Optimization for Heat Transfer Through Fins with Different Types of Notch ", Journal of Engineering Research and Studies, V. 2, Issue 1, pp. 133-138, 2011.

[3] Hossain, M. S., Ahamed, J. U., Akter, F., Das, D. and Soha, S. "Heat Transfer Analysis of Pin Fin Array ", Proceedings of the International Conference on Mechanical Engineering and Renewable Energy, Bangladesh, 24-27 December 2013.

[4] Dhanawade, K. H., Sunnapwar, V. K. and Dhanawade, H. S. "Thermal Analysis of Square and Circular Perforated Fin Arrays

by Forced Convection", International Journal of Current Engineering and Technology, Special Issue-2, pp. 109-114, February 2014.

[5] Sivasubramaniyam, R. "Performance Analysis on Array of Circular Fins through Forced Convection", International Journal of Engineering Research and Technology, Vol. 7, No. 1, pp. 1-8, 2014.

[6] Wankhede, S.D. and Meshram, C.B. "Experimental Study and Investigation of Heat Transfer from Horizontal Rectangular Inverted Notch Fin Arrays (INFAS) under Natural and Forced Convections ", International Journal Researchers, Scientists and Developers, Vol. 2, No. 1, pp. 13-16, 2014.

[7] Baqir, A.S., Qasim, A. and Adnan, A. "Experimental Study for Staggered Perforated Array of Pins Like Fins in a Rectangular Air Cross Flow", The Iraqi Journal for Mechanical and Material Engineering, Vol. 14, No. 2, pp. 261-275, 2014.

[8] Mousa, N. A. "Study on Flow Characteristics and Heat Transfer Behavior around Different Geometrical Corrugated Extended Surfaces", Al-Khwarizmi Engineering Journal, Vol. 11, No. 2, pp. 31-44, 2015.

[9] Mane, S.N. and Mane, S.S. "A Review on Heat Transfer from Different Types of Notch Fin Arrays under Forced Convection to Increase Heat Energy

Transfer", International Journal on Recent Technologies in Mechanical and Automobile Engineering, Vol. 2, Issue 6, pp. 15-18, 2015.

[10] Shehab, S.N. Shehab, S.N. "Experimental Simulation of Natural Heat Convection from Finned Vertical Plate with Different Inclinations', Al-Khwarizmi Engineering Journal, Vol. 12, No. 3, pp. 61-71, 2016.

[11] Shehab, S.N. "Experimental Study of Free-Convection from Rectangular Fins Array on a Heated Horizontal Plate with Notch Effects", Al-Nahrain Journal for Engineering Sciences, Vol. 20, No. 1, pp. 140-148, 2017.

[12] Mageswaran, C., Karthikeyan, R. and Muthukumaran, R. "Heat Transfer Enhancement of Perforated Square Pin-Fin Array in a Rectangulr Duc", International Journal for Research in Applied Science and Engineering Technology, Vol. 5, Issue IX, pp. 1826-1832, 2017.

[13] Cengel, Y. A. "Heat Transfer: A Practical Approach", McGraw-Hill Companies, Second Edition, USA, 2008.

[14] Rajput, R.K. "Heat and Mass Transfer", S. Chand and Company LTD., Fifth Revised Edition, India, 2012.

[15] Bejan, A. "Convection Heat Transfer", John Wiley and Sons, Inc., Fourth Edition, USA, 2013.

دراسة تجريبية للحمل القسري من مصفوفة زعانف مستطيلة أفقية داخل مجرى هواء

سعد نجيب شهاب

قسم الهندسة الميكانيكية / كلية الهندسة / الجامعة المستنصرية / بغداد

البريد الالكتروني: saadnajeeb16@ uomustansiriyah.edu.iq

الخلاصـة

يتناول هذا البحث دراسة تجريبية للتنبؤ بالأداء والخصـائص الحرارية للحمل الحراري القسري من مصـفوفة زعانف مستطيلة أفقية مسـخنة الى الهواء داخل مجرى مستطيل المقطع. أستخدمت ثلاثة أشكال مختلفة من مصفوفات الزعانف المستطيلة، الشكل الأول يتضمن زعانف صلبة (بدون شقوق وثقوب) والشكلان الآخران يتضمنان شقوق مستطيلة الشكل وثقوب دائرية الشكل بنسبتين مختلفتين للمسـاحات المزالـة من الزعنفة هي، 18% شقوق- 9% ثقوب و 9% شقوق- 18% ثقوب. تم تثبيت أبعاد الزعانف وعددها لجميع الحـالات. تم تسخين مصفوفة الزعانف كهربائيا من القاعدة بخمس قيم مختلفة من القدرة الداخلة. أستخدمت خمس سرع لجريان الهواء داخل المجرى. تم محاكاة تـأثير شكل الزعانف ودراستها وسرعة جريان الهواء وعدد رينولد والفيض الحراري للسطح على الأداء الحراري للحمل القسري عمليا. أظهرت البيانات العملية أن مصفوفة الزعانف المستطيلة التي تتضمن شقوق مستطيلة بنسبة 18% وثقوب دائرية بنسبة 9% تعطي أداء حرارياً قسرياً أفضل بدلالة معدل معامل أنتقال الحرارة أكبر بحوالي (25%- 45%) و (7%- 20%) مقارنـة بمصـفوفتي الزعانف الصـلبة والمتضـمنة 9% شقوق مـع 18% ثقوب. أستنتجت خمس علاقـات أرتباطية عملية للتنبؤ بمعدل رقم نسلت لمصفوفة الزعانف المتضمنة 18% شقوق مـع 9% ثقوب لمدى واسـع من الفيض الحراري للسطح، قورنت البيانات المقدّمة مع دراسات سابقة وأظهرت تقارباً جيداً في السلوك.