THERMAL ANALYSIS OF FORCED CONVECTION IN DUCT PACKED WITH POROUS FIN +

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Abstract :

The effect of inserting porous fins pad in square duct on the forced convection heat transfer is studied experimentally for Reynolds number of (7590 – 18400). Thetested ductwalls are subjected to uniform heat flux in the range of (64.74 – 120.91 w/m²) and the pad consist of commercial galvanized steel screen with porosity ranging between (0.985 – 0.971). It is observed that the heat transfer is enhanced by using porous fin at (1.44, 1.24) times than empty duct in vertical position when porosityare(0.971, 0.985) respectively and (1.32, 1.15) times in horizontal. For that the best condition obtained to enhance heat transfer for all tests when using fin pitchof (20cm) and porosityof (0.971) in vertical position. Many experimental correlation have beenobtained.

Keyword: Forced convection – packed duct – porous fin.

التحليل الحراري للانتقال الحرارة بالحمل القسري خلال مجرى محشو بزعانف مسامية

صباح نعمه فرج

المستخلص :

"تم دراسة تأثير أضافة زعانف مسامية كحشوة داخل مجرى هوائي مربع على أنتقال الحرارة بالحمل القسري عمليا" لرقم رينولد يتراوح (7590-18400). سخن السطح الخارجي للمجرى الهوائي بواسطة فيض حراري ثابت – 64074 (12091 w/m²). تتكون الحشوة من مشبك سلكي معدني بمساميةتتراوح (0.985- 0.971). بينت النتائج أن التحسن في أنتقال الحرارة يكون عند أستخدام الزعانف المسامية ب (1.44, 1.24) مرة عن المجرى الفارغ بالوضع العمودي عندما تكون المسامية تساوي(1.15و1.32) مرة عند الوضع الأفقي. مما تقدم فأن أفضل حالة لتحسين إنتقال الحرارة لكل التجارب تكون عند (البعد بين زعنفة وآخرى20 cm)ومسامية (0.971) عند الوضع العمودي وتم الحصول على عدد من المعادلات التجريبية.

الكلمات الدالة : الحمل القسري – مجرى مسامي – زعنفة مسامية.

Nomenclature:

A_w	Duct surface	m^2
CP_a	Air specific heat at constant pressure	kJ/kg.k

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D_H	Equivalent duct diameter	m
h_x	Local heat transfer coefficient	$W/m^2.°C$
H	Average heat transfer coefficient	$W/m^2.°C$
H	Duct height	m
I	Heater electrical current	A
K_f	Air thermal conductivity	W/m.°C
K_m	Effective thermal conductivity	W/m.°C
K_s	Porous fin thermal conductivity	W/m.°C
L	Duct length	m
$\dot{m}_a$	Air mass flow rate	kg/s
N_{ux}	Local Nusselt number	
N_u	Average Nusselt number	
Pe	Peclet Number	
q_s	Duct heat flux	W/m^2
Q	Powersuppliedbyheater	W
Q_f	Heat transferrate to air	W
Q_{loss}	Heat lossesrate from test duct	W
R	Heater electrical resistance	Ω
Re	Reynolds number	
T_{f1}	Inlet air temperature	°C
T_{f2}	Outlet air temperature	°C
T_{fx}	Local mean air temperature	°C
ΔT_{lnx}	Local logarithmic mean temperature	°C
ΔT_{sx}	Local surface duct temperature	°C

U	Air velocity	m/s
W	Duct width	m
X	Duct length from duct entrance	m
$\mathcal{E}$	Porosity	
ϑ	Kinematic viscosity of air	m²/s
α_m	Effective thermal diffusivity of of porous fin	
H.P	Horizontal position	
V.P	Vertical position	

Introduction:

Convection heat transfer has get considerable attention in past years in different kind of engineering application , such as solar collectors, compact heat exchangers, geothermal energy system, electronic devices and cooling of nuclear reactors.Extended surfaces are used especially to enhance the heat transfer rate between a solid and fluid, thereis a continuous effort by the researchers as wellthe designers to determine the optimum shape of the fin that will maximize the rate of heat transfer for a specified fin volume or minimize the fin volume for a given heat transfer for rate[1].

Experimental study was conducted by [2] on amplifiedmodel of the finned tube heat exchanger by free convection. It consisted of a heated vertical cylinder (300 meter length and diameters of 30, 50 and 80 mm) dimensions and ring porous fin were manufactured from commercial aluminum screen density of2770 kg/m³, thermal conductivityof 177W/m.k. Porosity placed around the cylinder. It was found that the value of (Nu) under constant heat flux conditionsvaries with the variation of heat flux and as the cylinder diameter decrease, (Nu) increased.

Also, ref. [3,4] studied numerically enhancement of free convection heat transferin horizontal rectangular fin as results of introducing body modification perforations to the fin body to make a best use of the material and size ofa given fin, the modifications is a vertical equilateral triangular perforations made through the fin thickness. The effect of geometrical dimensions of perforated fin (thickness of fin from 1 to 50 mm) and fin thermal conductivity from (50 to 500 W/m.k) the finding of this study were the gain in heat dissipation rate for the perforated fin is a strong relation of both, the perforation dimension and lateral spacing and the perforation of fins enhances heat dissipation rates and at the same time decrease the expenditure of the fin materials. For that wire mesh used as fin to increase heat transfer without a significant addition of weight and size. Pavel etal [5] investigated experimentally and numerically the effect of metallic porous materials manufactured by commercial aluminum screen wire diameter 0.8 mm, density 2770 kg/cm³cutted out at diameter (25.4, 38.1, 50.8 and 63.5 mm) and then inserted on steel rods. The distance between two adjacent screensof (2.5, 5 and 10 mm).This pad was inserted in a copper pipe which subjected to a constant and uniform heat flux. Heat transfer enhancement could be achieved using porous inserts whose diameters approach the diameter of the pipe for a constant diameter of the porous medium improvement can be attained by using a porous insert with a smaller porosity and higher thermal conductivity.

As wellas, saradaetal[6] performed experiments that proved augmentations of turbulent flow heat transfer in horizontal tube by means of mesh inserts with air as the working fluid. Mesh screen diameters of (22, 18, 14 and 10 mm) for varying distance between the screens of(50,100,150, and 200 mm)in the porosity range of (99.73 to 94.98) were considered for experimentation. It was observed that the enhancement of heat transfer by using mesh inserts when compared to plain tube at the mass flow rate is more by a factor of (2 times) whereas the pressure drop is only about a factor of (1.45 times).Lin etal [7],they was approximately proved to be an effective way for heat transfer enhancement when aporous medium inserted at the core of tube. The heat transfer rate of the tube with porous inserts whose diameters approach the diameter of the tube is about (1.6 – 5.5 times)larger than the smooth tube in ranges of Reynolds number (1064 – 4172). Most of the studies reviewed were conducted to enhance heat transfer for flow of air in horizontal position with wire mesh inserted as porous fin. For that, this effect will be studied in this workin vertical and horizontal rectangular duct subjected to constant heat flux with the variable of porosity (number of mesh screen), Reynolds number and heat flux to find the augmentation in convection heat transfer coefficient.

Experimental :

The system used in Ref. [8]is developed in this work and usedto investigate the influence of porous fins on convection heat transfer through duct.The schematic diagram of test duct is shown in Figs. (1 and 2) and plate (1), the duct is made of galvanized steel with (10x10 cm) cross section area. Air from the centrifugal fan passes through (10 cm) honeycomb rectifier to obtain uniform velocity profile and remove eddies.

The(1500 cm) length ductconsist ofthe first sectionof (500 cm)length isused as developing region and the second sectionof(1000 cm) is the test duct which packed and heated by uniform heat flux obtained from electrical heater manufactured by (0.5 mm) diameter nickel- chrome wire isolated by ceramic spheres. The electrical circuit is designed for a voltage of (0 -220 V) with maximum current of (28 A). Heat flux can be varied by changing heater voltage and current.

To obtain the verticalorientation test duct a 90 °elbow fitting the fan to vary the direction of air flow. (10 cm) of glass wool is used to insulatethe outer surface of hot duct to reduce thermal losses to ambient and (2 cm) pieces of Teflon placed at the entrance and exit of test duct to prevent conduction heat transfer axially. The velocity of outlet airismeasured by vane anemometer with precisionof (±0.01 m/s), through test measuring velocity repeated several times to check the flow stabilization. Temperatures of the top surface of test duct are measured by nine k- type thermocouple along the axially centerline and distance of (10 cm) between one and another. Also nine thermocouple are inserted at the center line of the right test duct surface. To measure the inlet and outlet air temperature, two thermocouples were used at the entrance and the exit of the test section.

The porous fin used in this paper are commercial galvanized steel screen (wire diameterof 0.76 mm, densityof 2659 kg/m³, thermal conductivity of164 w/m.k) cut out at (10*10 cm) dimension square size as shown in fig (3) and plate(2). The porous fin (wire mesh) with different pitches and numbers are taken as shown in table (1).

A thin screwed stainless steel rod was inserted in the center of(24 and 48) pieces of screen at the same size and joined together by nuts with (40 and 20mm) distances respectively between two adjacent screens (pitch p). There are (ε= 0.985 and 0.971) correspond to the two different number of screen inserts (case 1 and 2) respectively, the porosity (ε) is calculated by the volume ratio of the porous inserts to all the screens.

Table (1) porous fin characteristics

Porous fin	No. of piece	Pitch (p) mm	Porosity (ε)
Case (1)	24	40	0.985
Case (2)	48	20	0.971

Data processing

After a (36) runs of experiment with a various boundary condition which described in Table (2) for horizontal and vertical test duct position. The followingequations are used for the calculations.

After (4-5) hour to obtain thermal steady state, the amount of heat provided from electric heater is:

$$\boldsymbol{Q = I^2 . R} \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots \ (1)$$

The average heat transferred to the air passing through heated duct is:

$$\mathbf{Q_f = \dot{m}aCp(}\boldsymbol{T_{f2}}\mathbf{ - T_{f1})} \ \dots\dots\dots\dots(2)$$

The heat loss by conduction and radiation from the test duct experiment is equal:-

$$\boldsymbol{Q_{loss} = Q - Q_f} \quad \dots\dots\dots\dots\dots\dots\dots \ (3)$$

Duct heat flux can be calculated from the relation

$$\boldsymbol{q_s = \frac{Q_f}{A_w}} \dots\dots\dots\dots \ (4)$$

Where: A_w= Test duct surfaces area (m²)and equal:-

$$\boldsymbol{A_W = 2(H * L) + 2(W * L) = 2L(H + W)}$$

The local mean temperature of the air (T_{fx}) at length (x) from the ductentrance is found as:

$$\boldsymbol{T_{fx}} = \mathbf{T_{f1}} + (\boldsymbol{T_{f2}} - \mathbf{T_{f1}})$$

$$\boldsymbol{T_{fx} = \mathbf{T_{f1}} + \frac{q_s(2(H+W))*X}{\dot{m}a\ Cp}} \quad \dots\dots\dots\dots\mathbf{(5)}$$

Local heat transfer coefficient h_x is introduced in terms of logarithmic meantemperature (ΔT_{lnx})[9].

$$\boldsymbol{\Delta T_{lnx} = \frac{\Delta T_{ex} - \Delta T_{ix}}{ln\frac{\Delta T_{ex}}{\Delta T_{ix}}}} \ \dots\dots\dots\dots\dots\dots\mathbf{(6)}$$

$$\mathbf{Where}\boldsymbol{\Delta T_{ex} = T_{sx} - T_{fx}}$$

$$\boldsymbol{\Delta T_{ix} = T_{sx} - T_{f1}} \quad \dots\dots\dots\dots\dots\dots \ \mathbf{(7)}$$

$$\boldsymbol{h_x = \frac{q_s}{\Delta T_{lnx}}} \ \dots\dots\dots\dots\dots\dots\dots\dots\dots \ (8)$$

The local Nusselt number (N_{u_x}) is based on the equivalent duct diameter (D_H) and effective thermal conductivity (k_m):

$$\boldsymbol{N_{u_x} = \frac{h_x D_H}{k_m}} \quad \dots\dots\dots\dots\dots\dots\dots\dots \ (9)$$

(k_m) is the effective thermal conductivity of the porous fin [10].

$$\boldsymbol{k_m = k_f^{\varepsilon} k_s^{1-\varepsilon}} \dots\dots\dots\dots\dots\dots\dots\dots \ \mathbf{(10)}$$

The Reynolds number (R_e) is determined as

$$R_e = \frac{UD_H}{\vartheta} \ldots\ldots\ldots\ldots\ldots\ldots\ldots (11)$$

And the peclet number (P_e) is found from:

$$P_e = \frac{\vartheta D_H}{\alpha_m} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (12)$$

(h) is the average heat transfer coefficient obtained from

$$h = \frac{1}{L}\int_{x=0}^{x=L} h_x \, dx \ \ldots\ldots\ldots\ldots\ldots\ldots (13)$$

(N_u) is the average Nussult number :

$$N_u = \frac{hD_H}{K_m} \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (14)$$

All properties of air (density, dynamic viscosity and thermal conductivity) based on the bulk temperature.

Table (2) Experimental condition

position	Vertical – Horizontal
Heat flux W/m²	64.74 - 93.37 - 120.91
Reynolds number	7992 - 12901 – 17689
Porosity	0.985 – 0.971

Results and Discussion :

According to the experimental results of the above cases, Figure (4) shows the relationship between the surface temperature and the length of the airway measured at a different thermal flow and Reynolds number in the horizontal position (H.P) and the vertical position (V.P). It can be seen from this figure that the same trends of temperature behavior with fin change (p) for all heat flow values at any Reynolds value for each site surface temperature increases near the end of the duct with increasing heat flow, decreasing the number of Reynolds and increasing the fin pitch (P) and porosity.

Figs.(5) shows that the surface temperature at (V.P) has a value lower than (H.P), due to the effect of buoyancy force flowing in the same direction as the external forces (fan) driving the forced load.

In the same position, we see that the surface temperature at the fin pitch (20 cm) is less than (40 cm), due to increased disturbance in airflow through the duct, which tends to increase heat transfer from the hot duct and then reduce surface heat. The maximum difference in surface temperature at mid-duct length between the fin pitch is (12%, 11.5%) percent at (H.P) and (V.P), respectively. The difference between (H.P) and (V.P) is (7%, 7.5%) in the fin pitch (20 cm, 40 cm), respectively.

Figure (6) shows the variation of the local heat transfer coefficient with the length of the airway, and the general shape of the curves shows that (h) decreases towards the end of the duct when the difference in temperature (surface temperature - air temperature) is increased, heat transfer by convection to go down. For each experimental case (h) decreases with a lower number of Reynolds, lower heat flow and increased fin pitch (porosity).

In Fig.(7), the coefficient of heat transfer in (V.P) is higher than (H.P) (9.8%, 6.1%) when the pitch fin (p) (20,40 cm) respectively. In (V.P), the coefficient of heat transfer at (p = 20 cm) was higher than (p = 40 cm) by 14.2% and 10.6% in (H.P).

Fig. (8) shows the local Nusselt number in the test duct has a similar trend of local heat transfer coefficient as shown in Fig. (7). It can be seen that the increase in the number of local Nusselt number with the number of Reynolds increased due to increased turbulence and decreased thickness of the thermal boundary layer. It was also found that local (Nu) in (V.P) is higher than (H.P).

The effect of the pitch fin and porosity is shown on the loading of the Nusselt number in Fig. (9). In (P = 20, 40 cm), the number of local Nusselti is higher in (V.P) than (H.P) and at fin pitch (p= 20 cm), local Nusselt number is higher than in (p=40 cm) for (H.P) and (V.P).

To estimate the correlation of the mean number of Nusselt number which calculated by eq. (14), and this requires drawing a relationship between the number of Nesselt and Reynolds number, shown in Fig. (10). It can be observed that all curves have the same direction, increasing the number of Reynolds leads to an increase in the number of Nusselt due to increased air flow velocity leading to increased disturbance and more heat transfer rate.

It was also observed that the highest mean values of Nusselt number were (70.9, 65.2) at fin (20 cm), for (V.P) and (H.P), respectively. In the fin pich (40 cm), the highest mean values are (63.3, 57.9) for (V.P) and (H.P).

In Fig. (10), the experimental correlation is plotted using a lower-square analysis (Re = 7590 to 18400).

- For the vertical position

$$\boldsymbol{Nu = 8.3\, Re^{0.206}} \text{ .... } \mathbf{(40\ cm)}$$

$$\boldsymbol{Nu = 12.7\, Re^{0.173}} \text{ .... } \mathbf{(20\ cm)}$$

- For horizontal position

$$\boldsymbol{Nu = 5.3\, Re^{0.24}} \text{ .... } \mathbf{(40\ cm)}$$

$$\boldsymbol{Nu = 12.04\, Re^{0.171}} \text{ .... } \mathbf{(20\ cm)}$$

Fig. (11) shows the rate of Nusselt (Nu) virus Peclet number for both position. The increase (Nu) is seen with the increase (Pe) and the correlation between (Nu) and (Pe) are:

- For vertical position

$$\boldsymbol{Nu = 9.69\, Pe^{0.205}} \text{ .... } \mathbf{(40\ cm)}$$

$$\boldsymbol{Nu = 13.84\, Pe^{0.175}} \text{ .... } \mathbf{(20\ cm)}$$

- For horizontal position

$$\boldsymbol{Nu = 6.37\, Pe^{0.24}} \text{ .... } \mathbf{(40\ cm)}$$

$$\boldsymbol{Nu = 13.43\, Pe^{0.170}} \text{ .... } \mathbf{(20\ cm)}$$

The value of average Nusselt number for both fin pitch (20 cm, 40cm) are highest than empty duct which calculated from ($Nu = 0.023Re^{0.8}Pe^{0.4}$) at (1.44, 1.24) times in vertical position and (1.32, 1.15) times in horizontal position respectively.

Above the heat transfer through forced convection of the hot duct in the air passing through can be obtained in the pitch fin (20 cm) and porosity (0.971) in the vertical position due to increased turbulence, reduced resistance formed by the thermal boundary layer near the adhesive surface and increase surface contact surface between air and porous fins leading to increased heat transfer.

Conclusion :

The finding in present work illustrated that:

1. Surface duct temperature at fin pitch (20 cm)is lower than (40 cm) and in (V.P) is lower than
(H.P). It decreased with increasing Reynolds number and decreasing heat flux.
2. Local heat transfer coefficient and local Nusselt number at (V.P) is higher than (H.P) and at fin pitch (20 cm) is higher than (40 cm) at all tests.
3. The highest values of average Nusselt number are (70.9, 63.3) in (V.P) and (65.2, 57.9) in (H.P) respectively for fin pitch (20 cm, 40 cm).
4. Average Nusselt number in packed duct increase at (1.44, 1.24) times in (V.P) than empty duct for (p= 20 cm, 40 cm) respectively, and (1.32, 1.15) times in horizontal.

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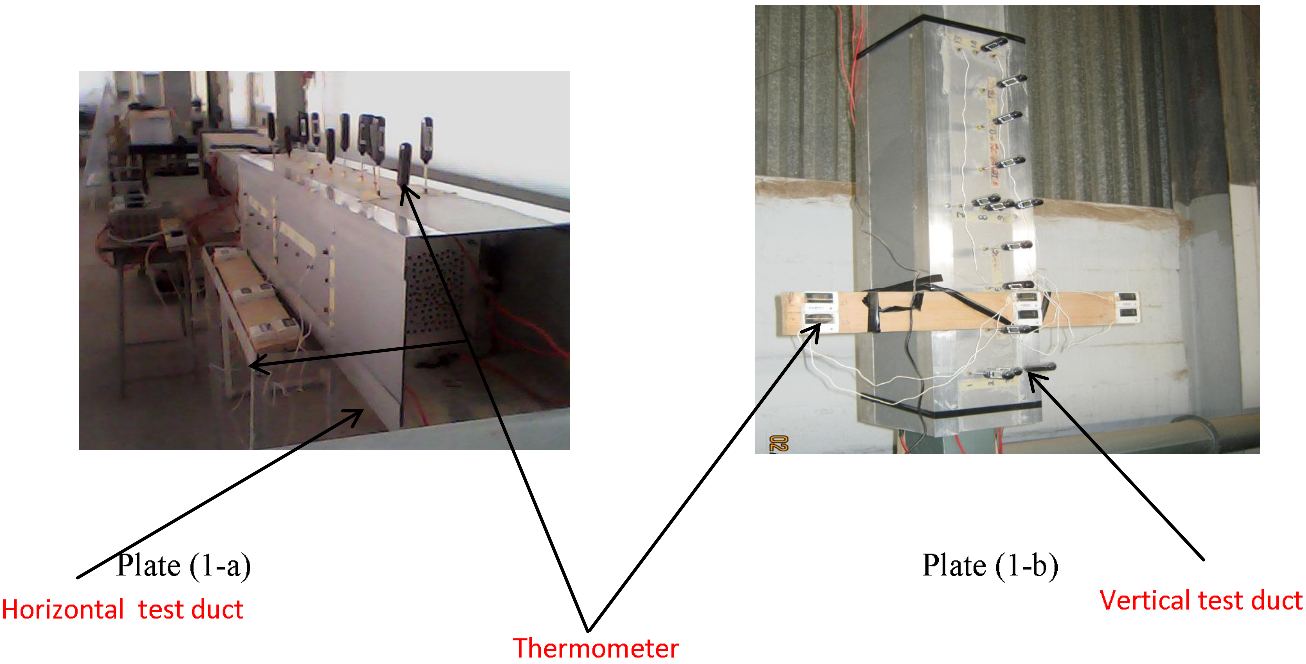

Plate (1) photographic of experimental apparatus

Plate (2) photographic view of porous fins

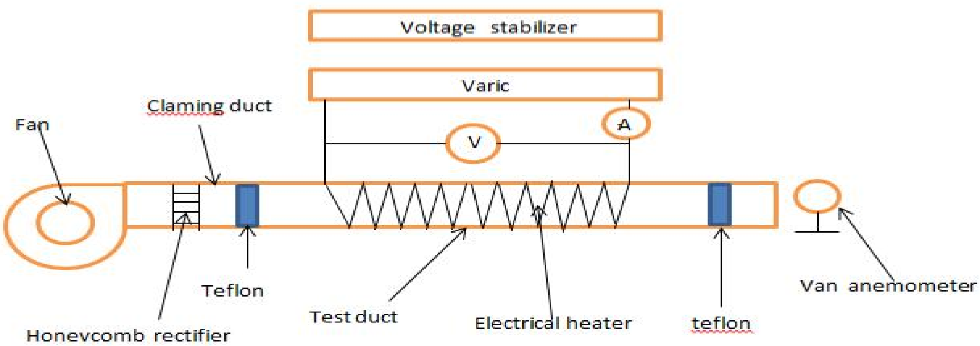

Fig. (1) Experimental apparatus for heat transfer of air

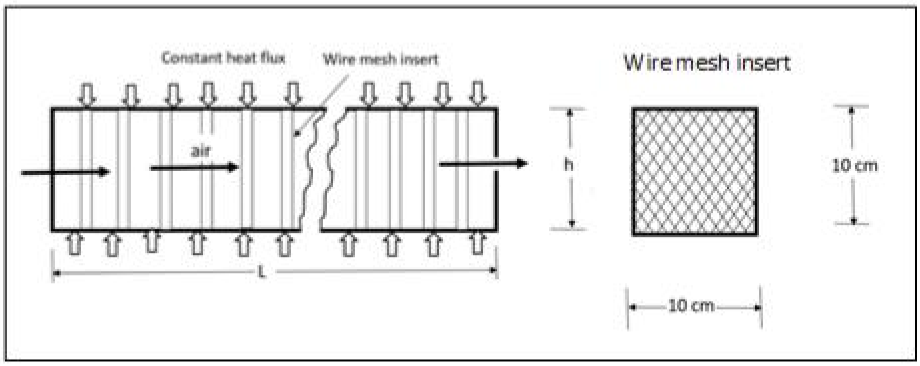

Fig. (2) Schematics diagram of the test section

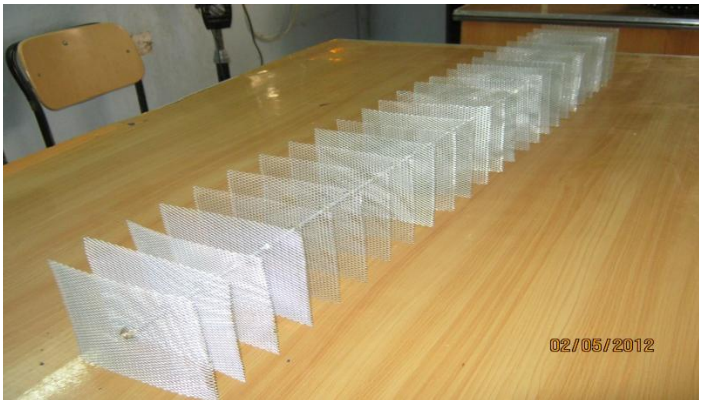

Fig. (3) Porous fins arrangement

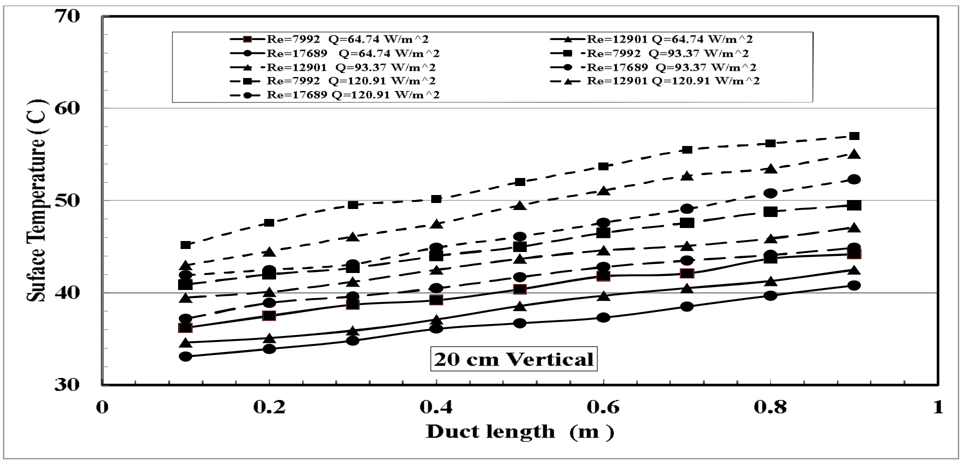

Fig. (4-a)The effect of Reynold number on the surface temperature(20 cm V.P)

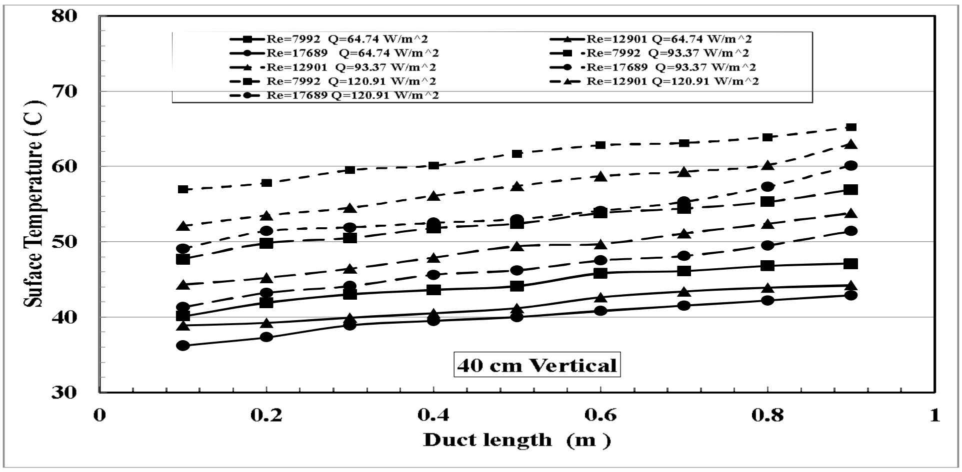

Fig. (4-b)The effect of Reynold number on the surface temperature(40cm V.P)

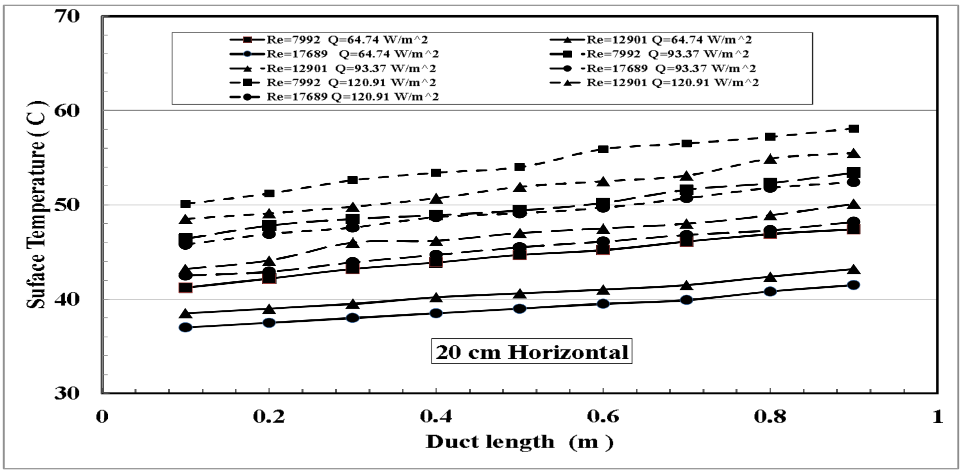

Fig. (4-c)The effect of Reynold number on the surface temperature(20cm H.P)

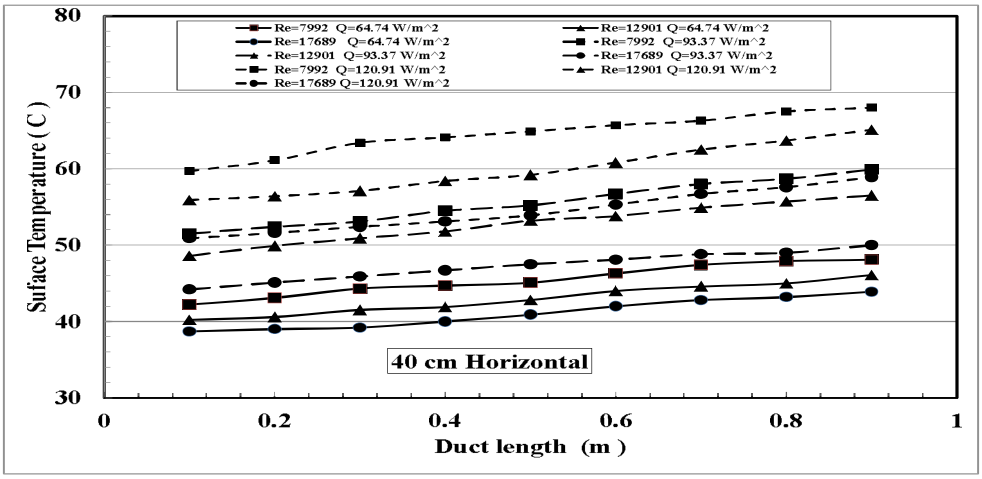

Fig. (4-d) The effect of Reynold number on the surface temperature(40cm H.P)

Fig. (4) The effect ofReynold number on the surface temperature

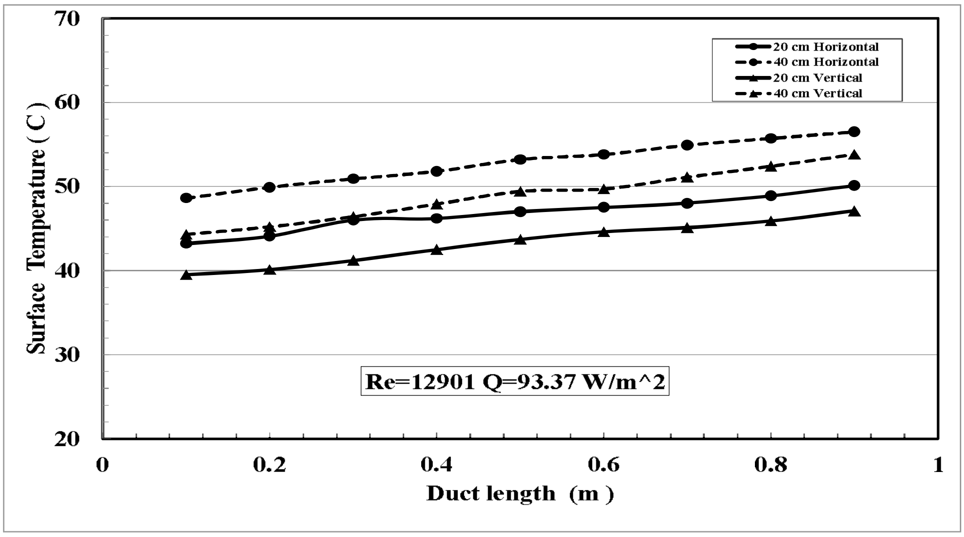

Fig. (5) The effect of fin pitch at vertical and horizontal position On surface temperature

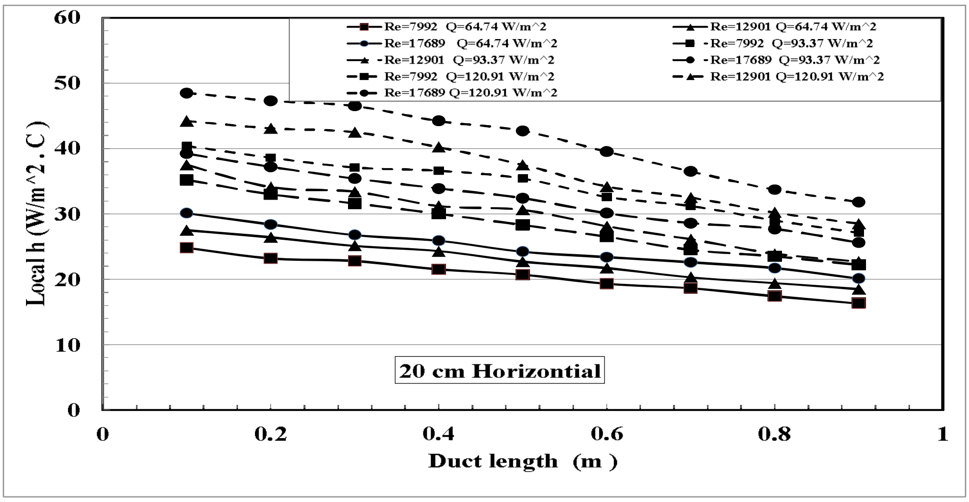

Fig (6-a)Variation of local heat transfer coefficient along test duct (20cm H.P)

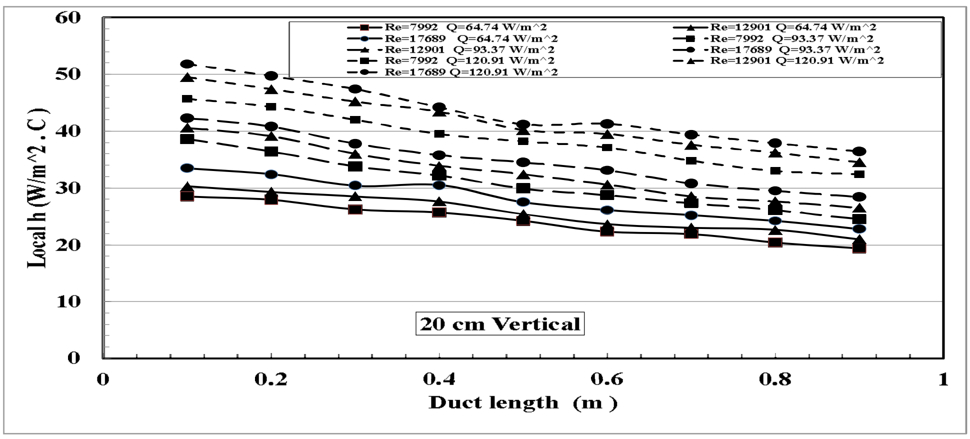

Fig. (6-b)Variation of local heat transfer coefficient along test duct (20cm V.P)

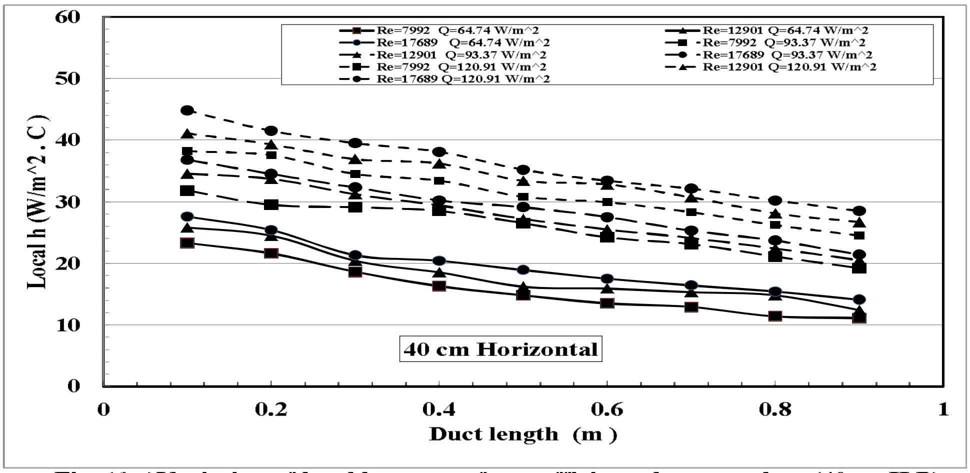

Fig. (6-c)Variation of local heat transfer coefficient along test duct (40cm H.P)

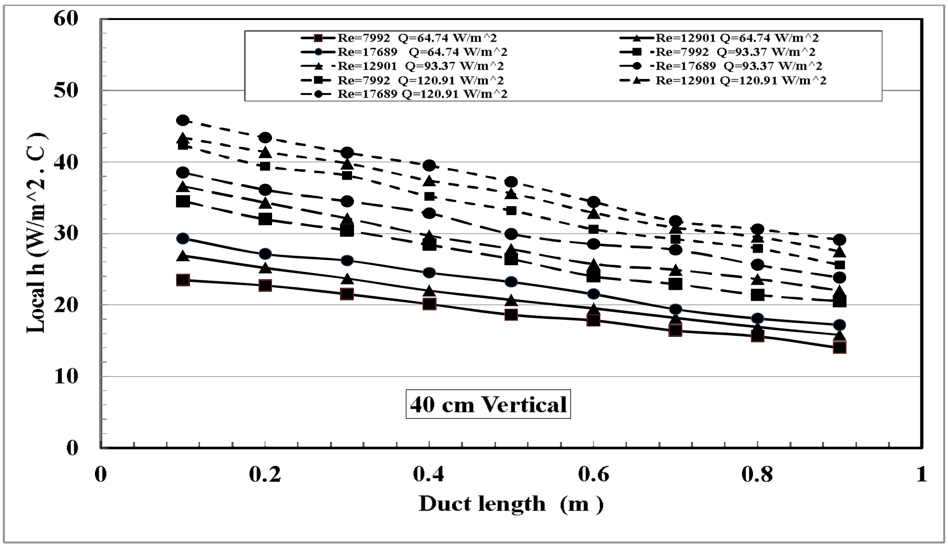

Fig. (6-d)Variation of local heat transfer coefficient along test duct (40cm V.P)

Fig. (6) Variation of local heat transfer coefficient along test duct

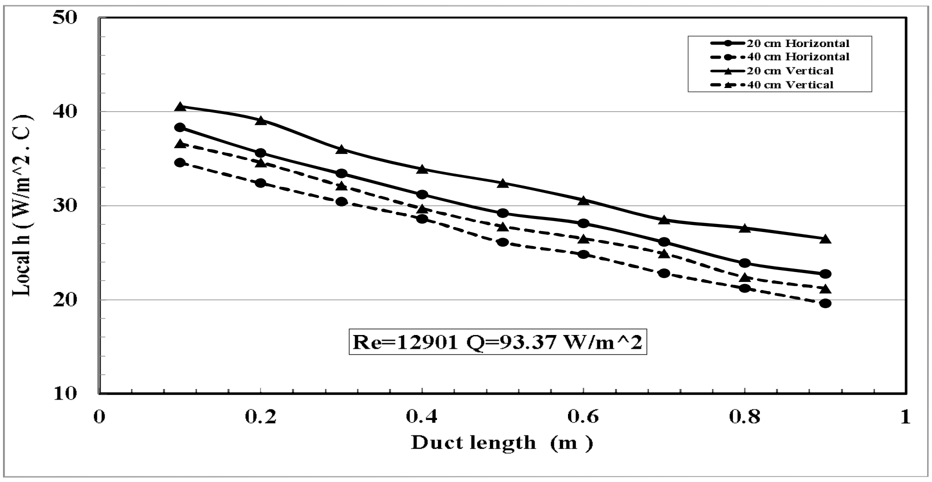

Fig. (7) Local heat transfer coefficient with variation fin Pitch and duct position.

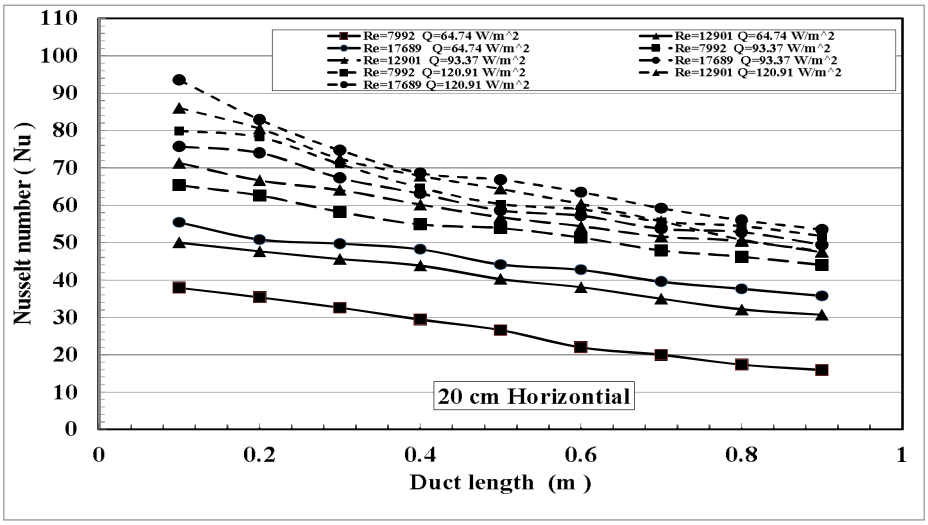

Fig. (8-a) Local Nusselt number against duct length at different experimental condition (20cm H.P)

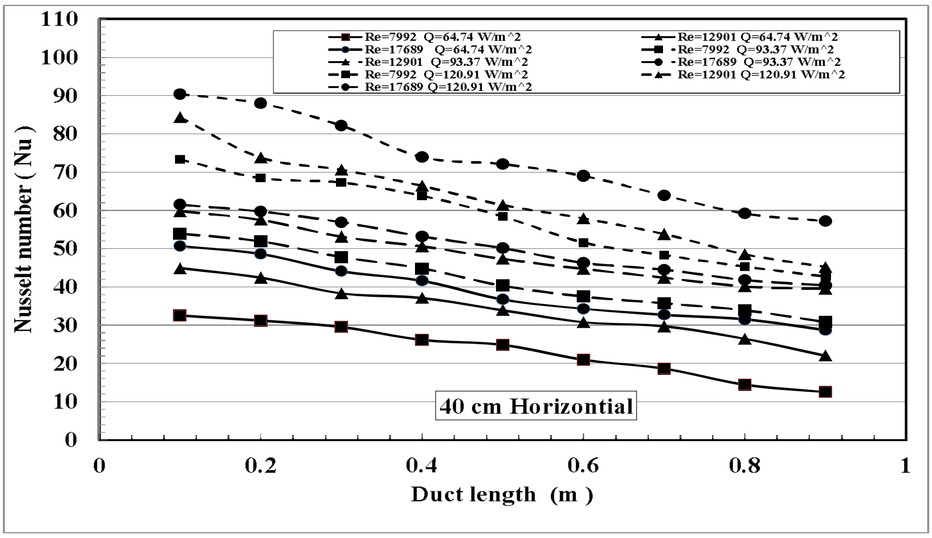

Fig. (8-b)Local Nusselt number against duct length at different experimental condition(40cm H.P)

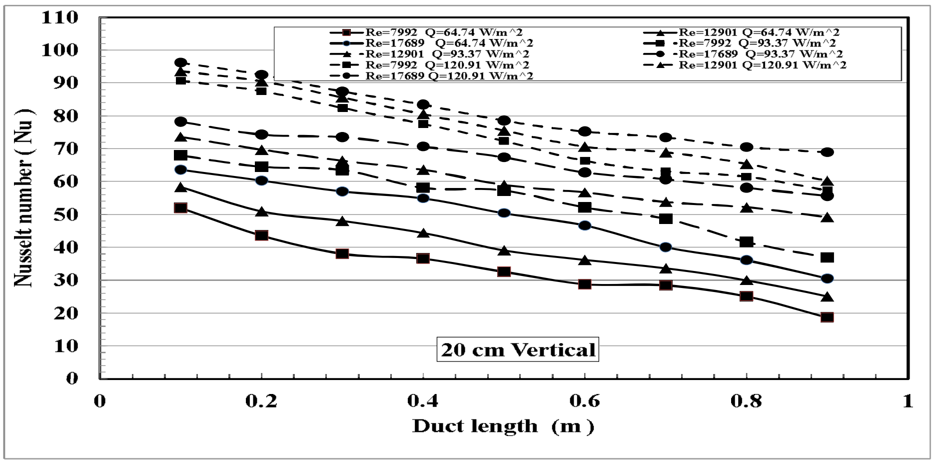

Fig. (8-c)Local Nusselt number against duct length at different experimental condition (20cm V.P)

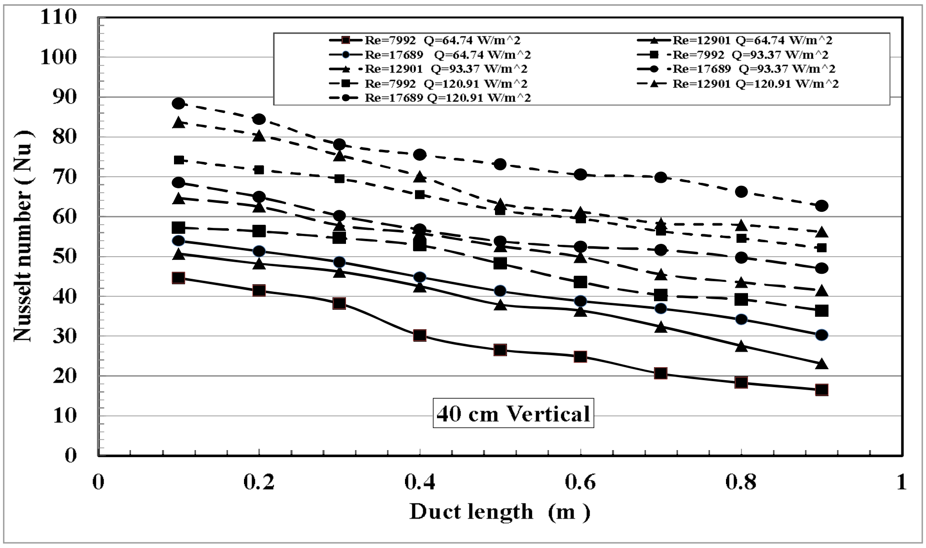

Fig. (8-d)Local Nusselt number against duct length at different experimental condition (40cm V.P)

Fig. (8) Local Nusselt number against duct length at different experimental condition

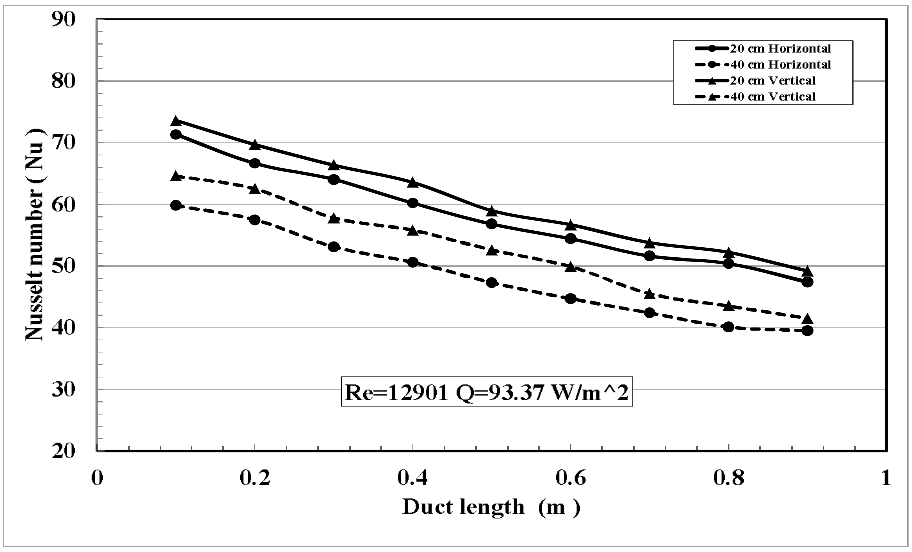

Fig. (9) Local Nusselt number with variation fin pitch and duct position

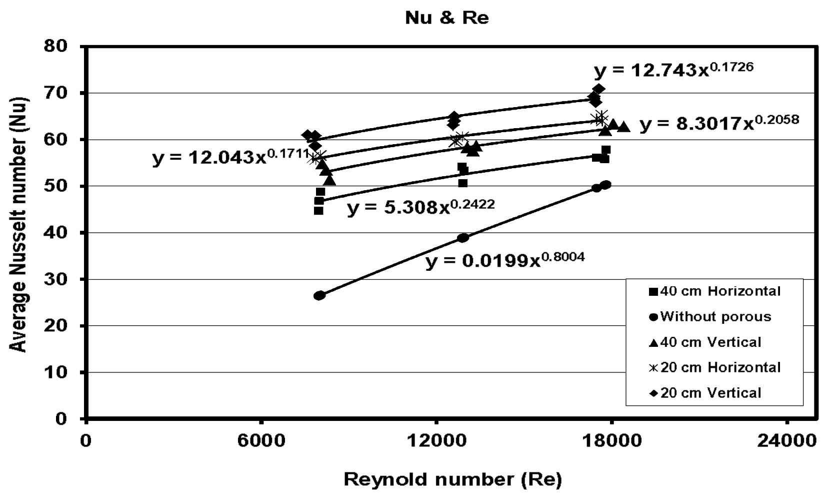

Fig. (10) Correlation of average Nusselt with Reynolds number

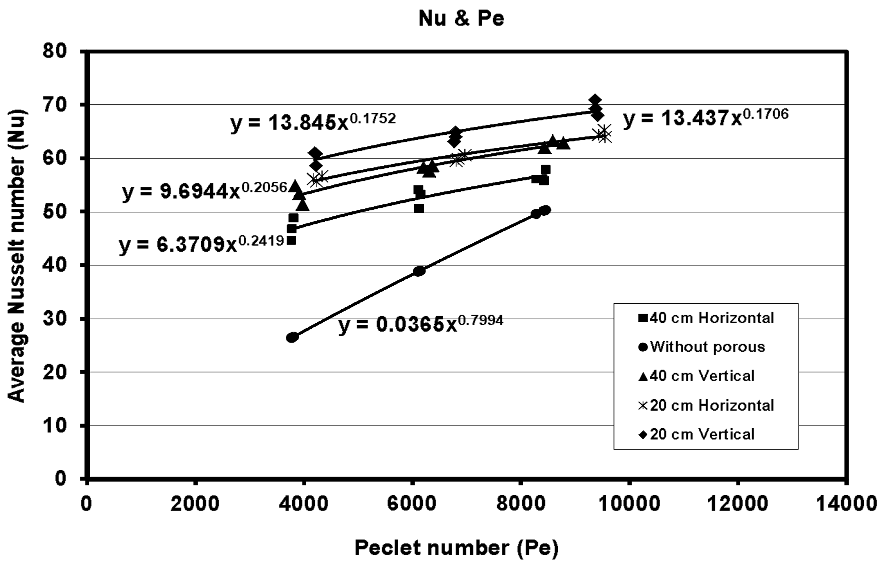

Fig. (11) Average Nusselt number with peclet number