An Experimental Study on Heat Transfer Enhancement for Porous Heat Exchange in Rectangular Duct

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ABSTRACT

Forced convection heat transfer of air in porous rectangular duct was investigated experimentally . The pad consist of (zig-zag) metallic wire mesh insert with two different porosities (ε) namely (0.97) and (0.99) .The experiments were carried out for Reynolds number (7682,12497 and 17323) and constant heat flux (192,297 and 422 W/m^2) .The results indicated that Nusselt number was increased with increasing Reynolds number and heat flux but decreased with increasing pad porosity .The Nusselt number in the porous duct was increased up to (144%) for ($\varepsilon = 0.97$) and (72%) for ($\varepsilon = 0.99$) when compared to clear duct at the same tested condition . For optimization between heat transfer enhancement and pad weight added into clear duct , it is found that [(Nu_{porous} - Nu_{clear}) / weight of pad] equal to (84.34) and (40.49) for ($\varepsilon = 0.99$ and $\varepsilon = 0.97$) respectively and this is good improvement in heat transfer through porous rectangular duct and reducing size and weight of original duct .

Keywords: Porous rectangular duct , Heat transfer , Wire mesh insert .

دراسة تجريبية لزيادة انتقال الحرارة لتبادل حراري مسامي في مجرى مستطيل المقطع

الخلاصة

انتقال الحرارة بالحمل القسري للهواء المار خلال مجرى مسامي مستطيل المقطع تم دراسته عمليا في هذا البحث . تتكون الحشوة من مشبك سلكي معدني متعرج بمسامية تتراوح بين(0.97 ، 0.99) . اجريت التجارب لرقم رينولدز (Re) (7682 , 12497 , 17323) وفيض حراري ثابت (q) (192 , 297 , 422 واط / م^2) . بينت النتائج ان رقم نسلت (Nu) يزداد بزيادة رقم رينولدز (Re) والفيض الحراري (q) ولكن يقل بزيادة مسامية الحشوة (ε) . رقم نسلت للمجرى المسامي يزداد بمقدار (144 %) عند ($\varepsilon = 0.97$) و (72 %) عند ($\varepsilon = 0.99$) عند مقارنته مع المجرى الخالي من الحشوة عند نفس الظروف التشغيلية ومن المفاضلة بين الزيادة في انتقال الحرارة ووزن الحشوة المضافة الى المجرى الخالي ، وجد ان[($Nu_{خالي}$ - $Nu_{مسامي}$) / وزن الحشوة المضافة] يساوي (84.34) و (40.49) عند مسامية (0.99) و (0.97) على التوالي وهذا يعتبر

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تحسين جيد في انتقال الحرارة خلال المجرى المستطيل المقطع اضافة الى تقليل حجم ووزن المجرى الاصلي .

INTRODUCTION

The low convective heat transfer coefficient between the symmetric heating duct and flowing fluid through it leads to develop many techniques for enhancing convective heat transfer . Porous media can be used as an effective heat transfer augmentation technique . Porous structures intensify the mixing of the flowing fluid and increase the contact surface area and consequently enhance the convective heat transfer [1] . One of the important porous media characteristics is represented by an extensive contact surface between solid and fluid surface . The extensive contact surface enhances the internal heat exchange between the phases and consequently results in an increased thermal diffusivity [2] . Several numerical and experimental studies had investigated enhancing convection heat transfer in engineering application by adding high thermal conductivity porous substrates . Paved etal [3] experimentally and numerically investigated the effect of metallic porous materials inserted in a pipe on the rate of heat transfer . The porous media used for experiments were manufactured from commercial aluminum screen wire diameter (0.8 mm) , density (2770 kg/m^3) , thermal conductivity (177 W/m^2.K)cut out at various diameter (25.4 , 38.1 , 50.8 and 63.5 mm) and then inserted on steel rods , the distance between two adjacent screens were (2.5 , 5 and 10 mm) and test were carried out using a rig composed of four copper pipe sections (31.75 mm radius), joint together by flanges and screws . The pipe was subjected to a constant and uniform heat flux . The effects of porosity , porous material diameter and thermal conductivity as well as Reynolds number (Re = 1000-4500) on the heat transfer rate and pressure drop were investigated . The results are compared with the clear flow case where no porous material was used . Heat transfer enhancement can 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 . Huang etal [4] further studied the heat transfer enhancement over a range of Reynolds number about (1000 – 19000) covering laminar , transitional and the turbulent regime and the heat flux is determined to make the temperature of the air rise about (10 ^{0}C) after it flows through the tube (internal diameter 17mm and its external diameter19 mm). The porous media are developed by cutting the commercial capper screen (8978 kg/m^3 in density and 387.6 W/m^2 .K in thermal conductivity) to many circular pieces with same diameter , inserting them on thin capper rod evently from the center then soldering to fix them together .

The heat transfer rate of the tube with porous inserts whose diameters approach the diameter of the tube is about (16-15.5 times) larger than the smooth tube cases in laminar , transitional and turbulent ranges of Reynolds number. Because of the random structures of porous media , they are different in geometry , the authors in Ref. [3 , 4] used metallic wire mesh to enhance heat transfer and Jengetal [5] used another porous materials which packed by brass beads with average diameter of (2 , 4 and 6 mm) . They filled spherical packing in the asymmetrically heated rectangular channels.

The channel width was fixed to be (60 mm) , variable parameters were the relative length of packed channel (length / beads diameter L/d_p = 5-60) and the relative height of packed channel (High / beads diameter H/d_p=1.67 – 15) . The experiments were carried at Reynolds number depending on hydraulic diameter(Re_D = 755 – 792) and particle Reynolds number was set to be (Re_{dp}= 38 – 2703) .The results indicated that the bead diameter (dp) rather than the hydraulic diameter (D_h)may be a proper parameter to generalize the data for heat transfer in a packed channel . The particle Nusselt number (Nu_{dp}) increased with decreasing (L/d_p) , while the (H/d_p)was not sensitive to (Nu_{dp}) .

Venkatestenetal[6] used another type of metallic porous structures which prepared from commercially available (250 *27*1.5 mm) dimensions perforated brass sheets having stamped holes (3 mm) diameter . The perforated sheets are assembled together by means of washers (1 mm thick) arranged between the perforated sheets in such a way that the holes in the consecutive plate were in staggered fashion .This porous samples were packed in vertical duct (390 * 250 * 62 mm) at different porosity (0.85 , 0.89 and 0.92) and number of perforated sheets used were (55, 37 and 23) respectively . Results indicated , that at a given heat input and a fixed Reynolds number , the Nusselt number increased with decreasing porosity , the porous insert of smallest porosity (0.85) gave the best heat transfer performance for which the highest increase in the average Nusselt number is approximately (4.52) times higher than that for clear flow case .

In earlier studies the porous media that had been employed in the rectangular channel / tube were made of conventional spherical beads and according to the above review of literature survey there are number of research that used the metallic porous media however not being used in large rectangular channel .

The present research will use a new arrangement of corrugated metallic wire mesh (zig- zag) packed to enhance the convection heat transfer in (12.5 *12.5*100 cm) channel heated with constant heat flux.

Experimental setup

The experimental study investigate the heat transfer enhancement over arange of Reynolds number of (7862 – 17323) and heat flux applied to channel wall ranging between (192-422 W/m^2) .

The schematic diagram and a photograph of the experimental set up is shown in Fig. (1) and plate (1) receptively . The apparatus consists of a centrifugal fan fitted with a horizontal rectangular galvanized steel duct (12.5* 12.5 cm) . Air flow through (5 cm) honeycomb rectifier before entering the duct in order to remove eddies and obtained more uniform velocity profile . (2 m) long calming duct follows the honeycomb rectifier for the air flow to be hydrodynamic fully developed . Then air entered the test section which shown in Fig.(2) , its dimension (12.5*12.5*100 cm) heated electrically isolated by ceramic spheres . The magnitude of the heat flux was adjusted by varying the intensity of the current measured with the ammeter and controlled by a varic connected to the heaters . To prevent heat loss to the environment , two layers of insulation are used , (0.25 mm) thickness asbestos tape wounded tightly then insulated with (200 mm) thickness of glass wool .

The axial conduction loss from test duct is prevented by using two pieces of Teflon at the entrance and exit of the duct .Eighteen k-type probe digital thermocouples of accuracy ($\pm$ 0.1^{0}C) are placed longitudinally at mid – width of the top and side plate . Also two thermocouples are located to measure inlet and outlet air temperature .

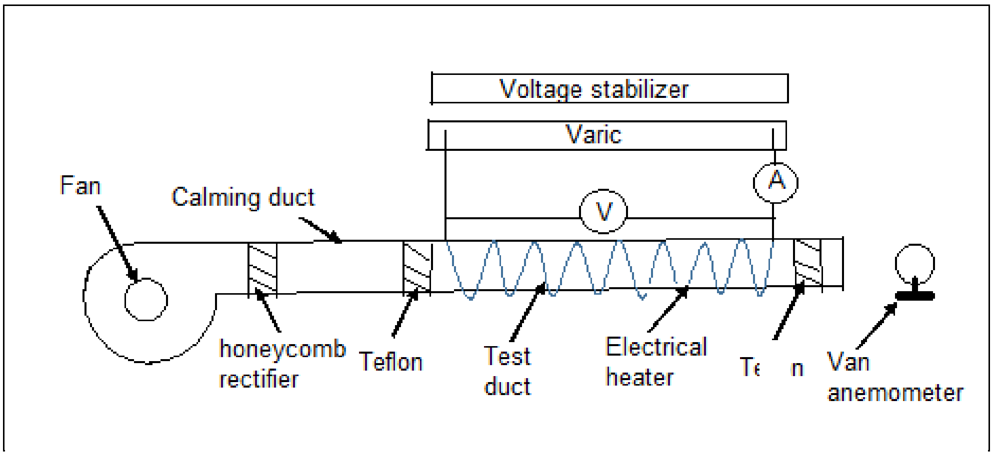

Figure (1) : Experimental apparatus for heat transfer of air

Plate (1) :Photographic of experimental apparatus

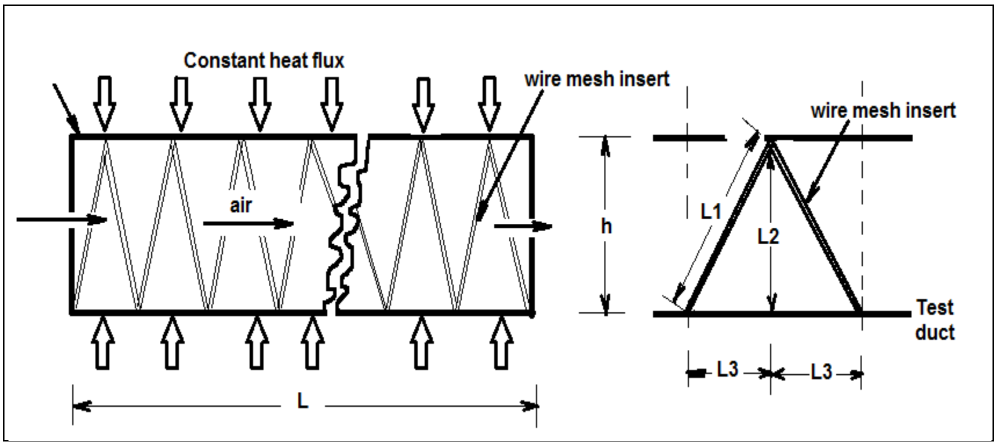

Figure (2) : Schematic diagram of the test section

The velocity of the outlet heated air is measured by vane anemometer with precision ($\pm$ 0.01 m/s) and returned several times through test to check the flow stabilization . The test section can be heated to different levels of heat flux by controlling the input to the heater by varic . A comparison of fluid sensible heat rise with electric power input showed that the maximum heat loss from the test duct does not exceed (6%) of the a power input .

The porous media used in the experiment are commercial galvanized steel screen (wire diameter 0.76 mm , density 2659 kg/m^3 , thermal conductivity 164 W/m.K) cut out at two dimension rectangular size as shown in Fig.(3) and plate (2) , many rectangular pieces with dimension size presented in Table (1) are arranged courrgeatly in (V ends) shape with varying of pieces number and inserted them on a thin screwed stainless steel rod evenly from the center and joined together by nuts . Each insert is taken and inserted into the test duct axially , the porosity of each porous medium is calculated by the volume ratio of the porous insert to all the screens on .These are ($\varepsilon = 0.97$ *and* 0.99) correspond to the two different number of screen inserts of (1 and 2) respectively .

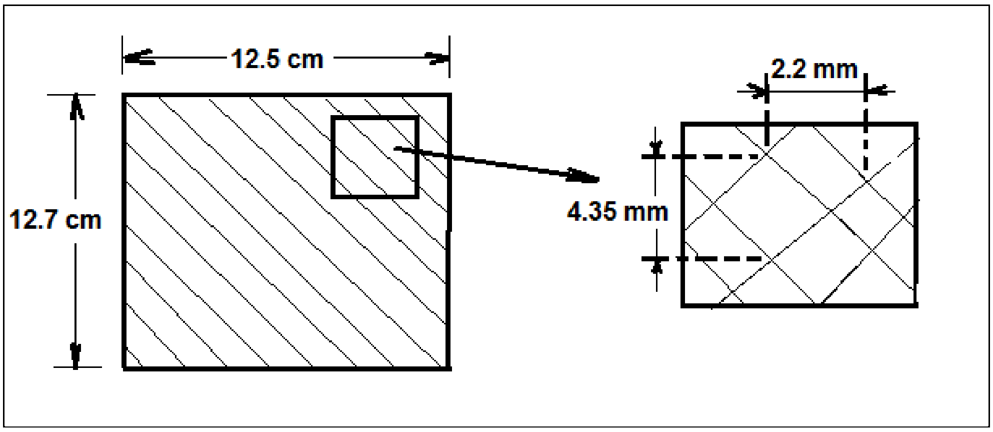

Figure (3) :Wire mesh insert inserts manufactured from galvanized steel screen

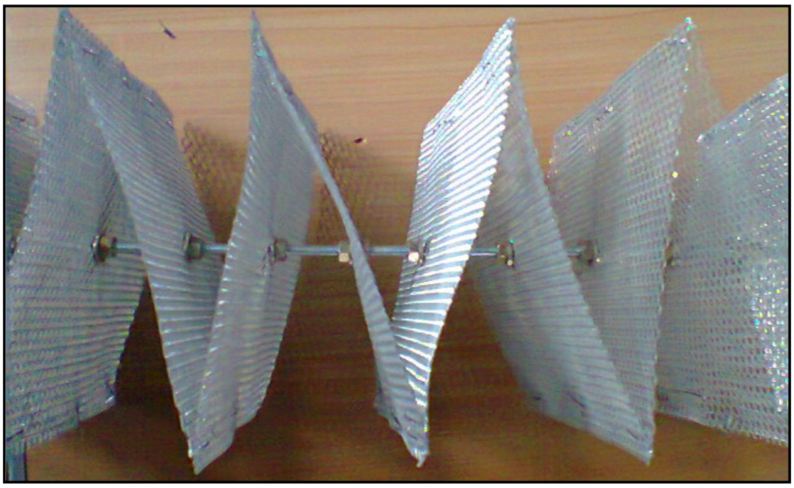

Plate (2) : Wire mesh insert

Table (1) Porous Pad Characteristics

	No.of Piece	L_1 (cm)	L_2 (cm)	L_3(cm)	ε
Wire mesh insert (1)	50	12.7	12.5	2	0.97
Wire mesh insert (2)	12	12.7	12	4	0.99

Experimental Processing

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

$$Q = \dot{m}_a * Cp_a * (T_{a2} - T_{a1}) \quad ..(1)$$

And the duct heat flux can be calculated from the relation :

$$q_w = \frac{Q}{A_s} \quad ...(2)$$

The local heat transfer coefficient (h_x) is defined in terms of the temperature difference between the heated wall and the temperature of air at desired distance from duct entrance :

$$h_x = \frac{q_w}{(T_{wx} - T_{ax})} \quad ....(3)$$

The local mean temperature of the air is calculated from the energy balance as :

$$T_{ax} = T_{1a} + \frac{q_w(2w + 2H)}{\dot{m}_a Cp_a} x \quad ...(4)$$

The local value for the Nussett number can be found from :

$$Nu_x = \frac{(h_x * D_h)}{K_m} \quad ....(5)$$

The hydraulic diameter (D_h) found as Ref. [8] defined it :

$$D_h = 4(W * H) / 2 * (W + H) \quad ..(6)$$

The effective thermal conductivity of the porous structures are defined as [7] :

$$k_m = k_f^{\varepsilon} k_s^{1-\varepsilon} \quad ...(7)$$

The Reynolds number is defined based on the hydraulic diameter of the duct and determined from the following expressions :

$$Re_D = \frac{UD_h}{\vartheta} \quad .....(8)$$

The Peclet number is computed as :

$$Pe = \frac{UD_h}{\alpha_m} \quad ....(9)$$

(h) is the average heat transfer coefficient calculated from :

$$h = \frac{1}{L}\int_{x=0}^{x=L} h_x d_x \quad ..(10)$$

and the average Nusselt number (Nu) is :

$$Nu = \frac{hD_h}{k_m} \quad ....(11)$$

All properties of air are evaluated corresponding to bulk air temperature .

For plain duct ,Nusselt number from the experimental data for rectangular duct were compared with the correlation recommend ended by Dilttus – Boelter [8] :

$$Nu = 0.023 \ Re^{0.8} \ Pr^{0.4} \qquad \ldots(12)$$

Results and Discussion

Experimental results for the effect of inserting corrugated wire mesh pads through rectangle duct with the influence of porosity , Reynolds number and heat flux are discussed, the following values were conducted in this research :

Porosity (ε) : Mesh insert 1 (0.97) and Mesh insert 2 (0.99)

Reynold number (Re) : 17323 , 12497 and 7682

Heat flux (q_w) = 422 , 297 and 192 W/m^2

Firstly , heat exchange through clear duct (without porous pad) would be examined in five experiments and compared Nusselt number obtained from experimental work with the value obtained from Dittus – Boelter correlation [8], it is shown good agreement at (10 %) percent at maximum difference , for that we used this correlation to examine the enhancement in heat transfer through porous duct.

Figs. (4 & 5)show the profile of surface duct temperature along duct length recorded in Mesh insert (1) and Mesh insert (2) , the general shape of all curves showed that the temperature increases towards the end of the duct and this temperature increases as heat flux increases while decreases as the Reynolds number increases .

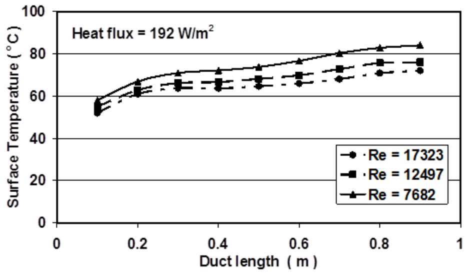

Fig. (4-a)

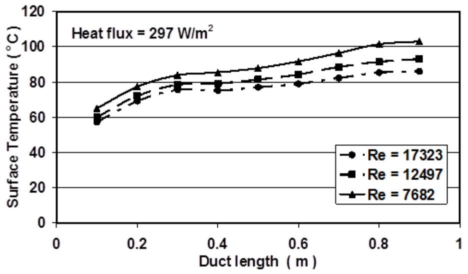

Fig. (4-b)

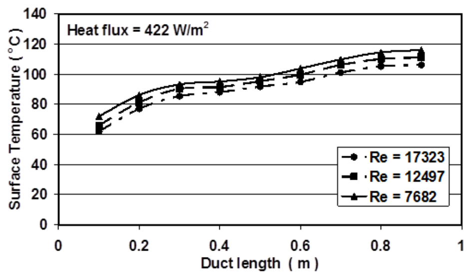

Fig. (4-c)

Figure.(4) : Surface duct temperature with the variation of Reynolds number and heat flux for wire insert (1) (ε =0.97)

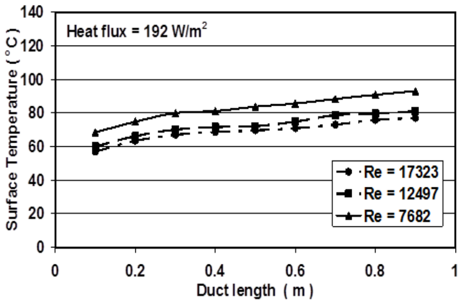

Fig. (5-a)

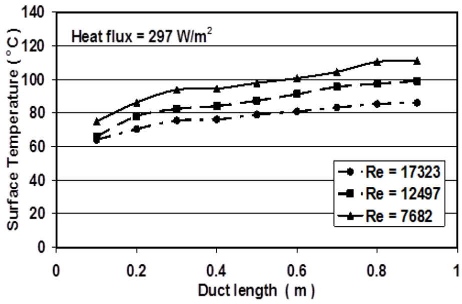

Fig. (5-b)

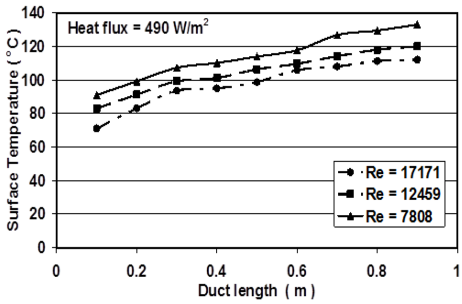

Fig. (5-c)

Figure.(5) : Surface duct temperature with the variation of Reynolds number and heat flux for wire insert (2) (ε =0.99)

The wire mesh insert greatly decreased duct surface temperature compared to the empty duct due to decrease thermal boundary layer thickness and increase turbulence of flow near the surface in addition to duct core . When comparing the rise in temperature of the two cases (wire insert 1 & wire insert 2) , we note that the highest temperature will be at wire insert 2 . Because of clearance between the duct surface and porous media , building thermal boundary layer near the surface will be occurred and resisted heat to transfer into air.

Figs.(6& 7) illustrate the variation of the local heat transfer coefficient along duct length for (wire insert 1 & wire insert 2) , tested at the same heat flux and Reynolds number . It showed that , the local heat transfer coefficients decrease with the increase of duct length from entrance due to increasing the local fluid temperature which lead to reducing the difference with the local surface duct temperature ,increasing the thickness of thermal boundary layerand thermal resistance then induce reducing heat transfer rate as the air moves through the duct . At the same heat flux , as the Reynolds number increases , the local heat transfer coefficient increases because of the increased turbulence resulting from the presence of the wire mesh pad.

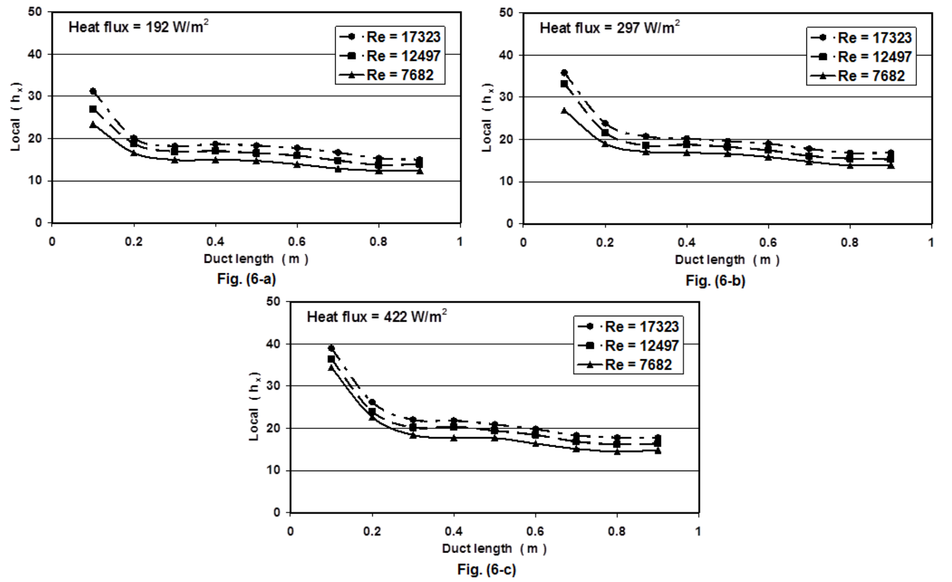

Figure.(6) : Variation of local heat transfer coefficient along duct length for (wire mesh insert(1) (ε =0.97)

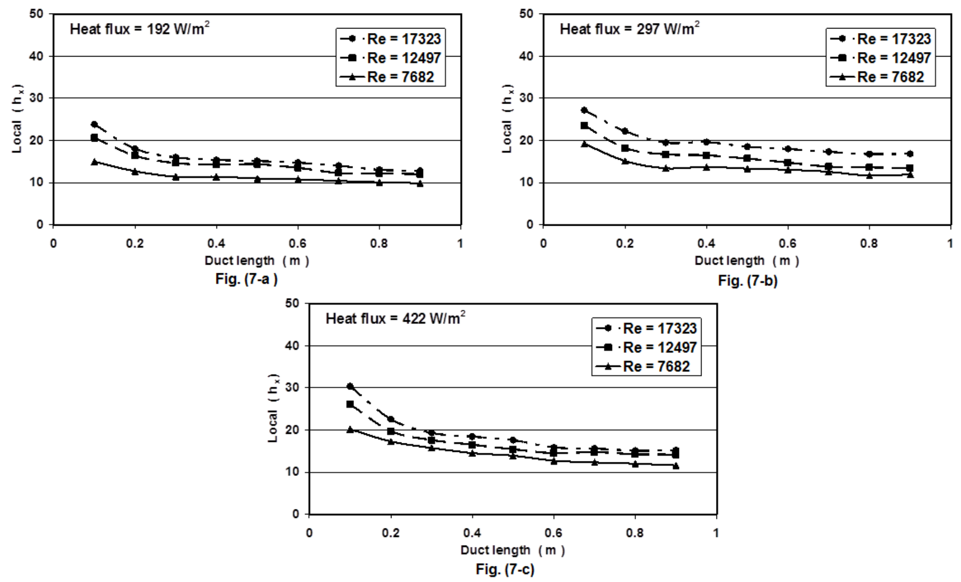

Figure.(7) : Variation of local heat transfer coefficient along duct length for (wire mesh insert(2) (ε =0.99)

The results of local Nussult number calculated from Equation (5) which depend on effective thermal conductivity are presented in Figs.(8& 9) . It can be seen that (Nu_x) take similar behavior of local heat transfer coefficient .The variation in porosity (ε) has a strong influence upon the local Nusselt number, at ($\varepsilon = 0.97$) leads to higher (Nu_x) and the cross – sectional area available for fluid flow smaller than($\varepsilon = 0.99$) and thus results a higher turbulence causes heat transfer enhancement .

Also at ($\varepsilon = 0.97$) , the wire mesh insert (1) are in contact with the inner walls of the duct and the high thermal conductivity of the pad , leads to high heat transfer from duct into two way the first to the air from surfaces duct directly by convection and the second from duct to the wire mesh by conduction then air convective heat . This is in addition to turbulence and reduce the thickness of the thermal boundary layer , which lead to increased heat transfer and increase Nusselt number . In ($\varepsilon = 0.99$), the wire mesh insert (2) made a void between the mesh and the walls of the duct resulting to convection heat transfer to the air as well as to increase turbulence mixing of flowing air and increase the heat transfer to the air .

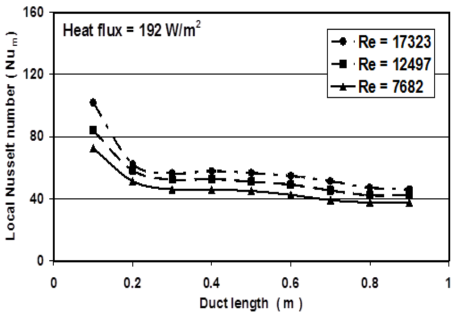

Fig. (8-a)

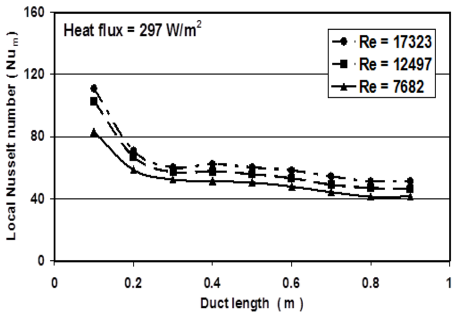

Fig. (8-b)

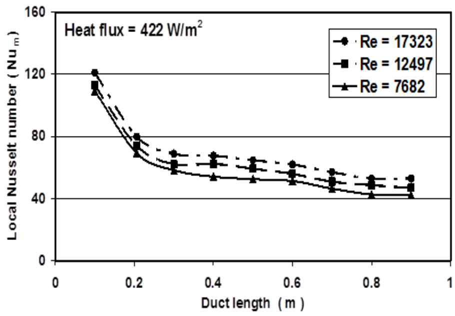

Fig. (8-c)

Figure.(8) : Distribution of the local Nusselt number along the duct length for (wire mesh insert(1) (ε =0.97)

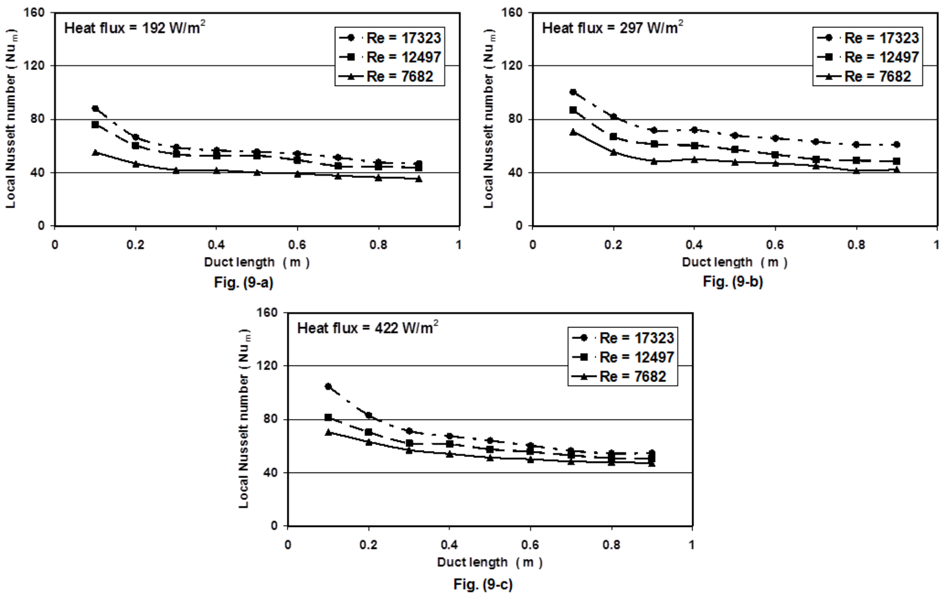

Figure.(9) : Distribution of the local Nusselt number along the duct length for (wire mesh insert(2) (ε =0.99)

Fig .(10) illustrates the relation between average Nusselt number and Reynolds number for cases with and without wire mesh inserts The value of (Nu) for both pad (mesh insert 1 & mesh insert 2) are highest than clear duct which calculated from equation (12) at (144%) and (72%) present respectively .At (Re=7600) and decreased with increasing Reynolds number until its reached (47%) and (20%) percent respectively at (Re=17400).

The measured data yield the following relationship between (Nu) and (Re):

$$Nu = C_1 (Re)^{C_2} \qquad \ldots\ldots(13)$$

Where (C_1)and (C_2) presented in Table (2)

Table (2) The corresponding factors of correlation (13)

	C_1	C_2
Wire mesh insert 1	10.16	0.2002
Wire mesh insert 2	1.2873	0.3926
Without wire mesh	0.0158	0.8138
Clear duct [8]	0.0198	0.8009

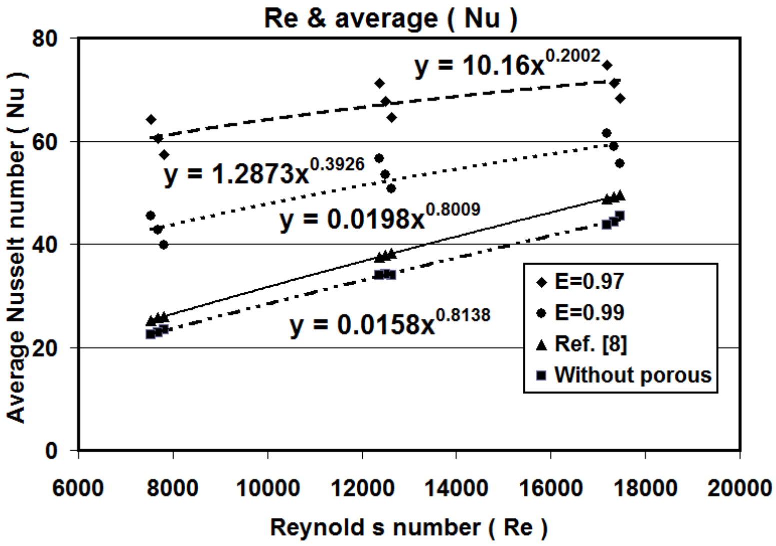

Figure.(10) : Average Nusselt number as a function of Reynolds number for mesh 1&2

The average Nusselt number (Nu) against the Pecletnumber is plotted in Fig.(11) , it is noted that a general increase in (Nu) with increasing of (P_e).
As is known the definition of Peclet number is the ratio of (the thermal energy convected to the fluid) to(the thermal energy conducted with in the fluid) , for that (Nu) and (Pe) increase as the average wall temperature with respect to the air inlet temperature decreases.
From Fig .the relation between (Nu) & (Pe) can be expressed as :

$$Nu = C_3(Pe)^{C_4} \qquad \ldots\ldots(14)$$

Where (C_3) and (C_4) illustrated in Table (3)

Table (3) Constants (C_3) and (C_4) for Equation (14)

	C_3	C_4
Wire mesh insert 1	11.925	0.199
Wire mesh insert 2	1.7572	0.3906
Clear duct	0.0371	0.7979

There is anew factor that can be used to find relationship between Nusselt number and weight of wire mesh insert added to the duct in order to analysis the practical results that were obtained and Table (4) shows the relation Reynolds number with the percent of the increase in Nusselt number of porous duct and clear duct into weight of pad added.
From Table we note that this factor is higher in the case of ($\varepsilon = 0.99$) because it will be a little added weight to get this optimization ,and highest lowest percent are (84.348 and 25.871) respectively. This is considered good improving to heat transfer through the duct even if the weight is an important factor in the design and we can

reducing the size and weight of the original duct in addition to the improving heat transfer by added porous pad into clear duct .

Table (4) The ratio of difference of Nu in porous duct and clear duct into weight of pad at different (Re)

Wire mesh insert (1)		Wire mesh insert (2)
$\varepsilon = 0.97$		$\varepsilon = 0.99$
Weight of pad (WOP) = 0.889 kg		Weight of pad (WOP) =0.2134 kg
Re	$(Nu_{porous} - Nu_{clear})$ /WOP (1/kg)	$(Nu_{porous} - Nu_{clear})$ /WOP (1/kg)
7600	40.494	84.348
13000	32.620	65.604
17400	25.871	46.860

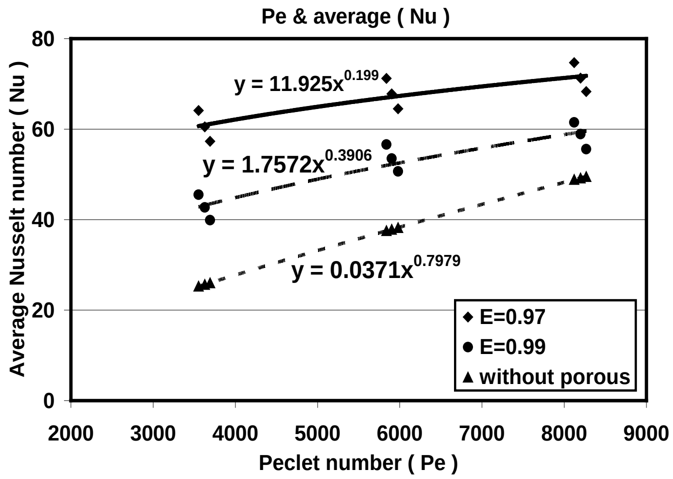

Figure.(11) : Average Nusselt number versus Peclet number for mesh inserts 1&2

Results of average Nusselts number versus Reynolds number at($\varepsilon = 0.97$) shown in Fig. (10) are compared with results in Fig .(12) at ($\varepsilon = 0.975$) which obtained from Ref.[4] . It is shown good agreement between two works with a percentage difference ratio (6.19 %) as shown in Table (5) .

Table (5) Validation between present work and Ref.[4]

Re	Average Nu		
	Percent work	Ref.[4]	Difference %
	$\varepsilon = 0.97$	$\varepsilon = 0.975$	
7600	61	47	29.7
13000	68	68	0
17400	72	80	-11.11

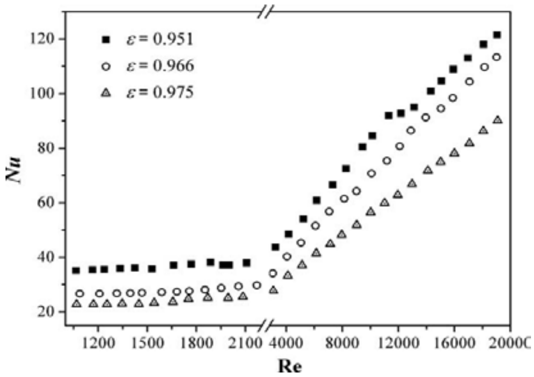

Figure.(12) Experimental results of Nusselt number from Ref.[4]

CONCLUSIONS

In the present paper , heat transfer enhancement by inserting (zig – zag) wire mesh pad through rectangular duct heated with constant and uniform heat flux has been investigated experimentally . It are shown that, the Nusselt number increases with decreasing porosity , while at fixed porosity , the Nusselt number increases with increasing heat flux and Reynolds number .At tested conditions , the porous inserts of ($\varepsilon = 0.97$ & $\varepsilon = 0.99$) gives highest value of Nusselt number than clear duct at (144 %) and (72 %) percent respectively . A new coefficient was used to determine the relationship between increasing in heat transfer and added weight of pad , it shows that the heighest increase in the Nusselt number ranging between (84) and (47) per (1 kg) of the pad at ($\varepsilon = 0.99$) .

Nomenclature

A_S	Duct surface area	m^2
Cp_a	Air specific heat at constant pressure	kJ/kg.K
D_h	Hydraulic diameter	m
d_p	Bead diameter	mm
h_x	Local heat transfer coefficient	$W/m^2.K$
h	Average heat transfer coefficient	$W/m^2.K$
H	Duct height	cm
K_f	Air thermal conductivity	W/m.K
K_m	Effective thermal conductivity	W/m.K
K_s	Wire mesh thermal conductivity	W/m.K
L	Duct length	cm
L_1	Wire mesh insert height	cm
L_2	Wire mesh (zig-zag) wave amplitudes	cm
L_3	A quarter of wavelength (wire mesh (zig-zag)wave	cm
Nu_{dp}	Particle Nusselt number	-
Nu_x	Local Nusselt number	-
Nu	Average Nusselt number	-
$\dot{m}_a$	Air mass flow rate	Kg/s
Pr	Prandtl number	-

q_w	Duct heat flux	W/m^2
Q	Heat transfer rate	W
Re	Reynolds number	
Re_D	Reynolds number depending on hydraulic diameter	-
Re_{dp}	Reynolds number depending on bead diameter	-
T_{1a}	Inlet air temperature	^0C
T_{2a}	Outlet air temperature	^0C
T_{WX}	Local duct surface temperature	^0C
T_{ax}	Local air mean temperature	^0C
U	Air velocity	m/s
W	Duct width	cm
X	Duct length from duct enterence	cm
Pe	Peclet number	-
ε	Porosity	-
ϑ	Kinematic viscosity of air	m^2/s
α_m	Effective thermal diffusivity of pad	m^2/s

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