EXPERIMENTAL AND NUMERICAL STUDY OF CONVECTION HEAT TRANSFER IN PLATE CHANNELS FILLED WITH PLASTIC PARTICLES [+]

Layth Taleb Fadhl [*]

Abstract :

Experimental and numerical study has been conducted to examine forced convection for air flowing through rectangular duct (10 * 12*100 cm) for vertical and horizontal position; the duct was heated at constant heat flux ranging from (0.2 to 0.54kW/m^2). The porous materials were packed by plastic sphere with diameter (4cm). The tests were done for Reynolds number ranged (6913-11433).The results showed that Nusselt number increased with increasing Reynolds number and heat flux for both position. Nusselt number in vertical position is higher at (14%) percent than horizontal and highest at (1.1, 0.91) times than empty duct at vertical and horizontal position respectively. Many experimental correlations have obtained.

Key word: forced convection – rectangular packed duct – spherical pad

دراسة عملية ونظرية لانتقال الحرارة بالحمل القسري خلال مجرى صفائحي محشو بكرات بلاستيكية

ليث طالب فضل

المستخلص:

اجريت دراسة عملية و نظرية لانتقال الحرارة بالحمل القسري لهواء مار خلال مجرى صفائحي ذو مقطع مستطيل بابعاد 10 (* 12 * 100 سم) للوضعين العمودي والافقي , سخن المجرى بفيض حراري ثابت(0.2− 0.54 كيلو واط/متر^2) , الحشوة المسامية مكونة من كرات بلاستيكية بقطر (4 سم) .تم اجراء الاختبارات لمدى رينولدز يتراوح بين (11433− 6913)و بينت النتائج زيادة عدد نسلت بزيادة رينولد و الفيض الحراري المسلط للوضعين العمودي والافقي .رقم نسلت في الوضع العمودي يكون اعلى بنسبة (14%)عن الوضع الافقي ويكون اعلى ب(1.1 , 0,91) مرة عن المجرى الفارغ للوضعين العمودي والافقي على التوالي. تم الحصول على عدد من المعادلات التجريبية .

الكلمات الدالة: الحمل القسري – مجرى مسامي مستطيل – حشوة كروية

[+] Received on 2/4/2014 , Accepted on 16/3/2015
[*] Assistant Lecturer / Institute of Technology / Baghdad / Middle Technical University

A_S	Duct surface	m^2
Cp	Specific heat at constant pressure	$kJ/kg.^0C$
D_{eq}	Equivalent diameter	m
D_P	Particle diameter	m
h_x	Local heat transfer coefficient	$W/m^2.^0C$
H	Duct height	m
I	Electrical current	Amp.
K_e	Effective thermal conductivity	$W/m.^0C$
K_s	Pad thermal conductivity	$W/m.^0C$
K_f	Fluid thermal conductivity	$W/m.^0C$
L	Duct length	m
$\dot{m}$	Air mass flow rate	Kg/s
Nu_x	Local Nusselt number	
Nu_{av}	Average Nusselt number	
P	pressure	N/m^2
Pe	Peclet Number	
q_w	Duct heat flux	W/m^2
Q	Heat energy	W
R	Electrical resistance	ohm
Re	Reynolds number	
Re_{Σ}	Particle Reynolds number	
t_{sx}	Local duct surface temperature.	^0C
t_{bx}	Local air bulk temperature.	^0C
T_{in}	Inlet air temperature.	^0C
T_{out}	Out let air temperature.	^0C
u_i	Inlet uniform velocity	m/s
V_T	Total duct volume	m^3
V_{pad}	Pad volume	m^3
w	Duct width	m
x	Refers to axial length	
y	Refers to vertical length	
ρ	Air density	Kg/m^3
Σ	Porosity	
ϑ	Kinematic viscosity	m^2/s
μ	Dynamic viscosity	Kg/m.s

1- Introduction :

Packed beds due to their high surface area –to- volume ratio are widely used in a variety of industries, such as catalytic reactors, absorption tower, packed bed regenerators, high temperature gas- cooled nuclear reactors and heat accumulators [1]. There are a number of factors affecting the transmission of heat transfer through particles packed bed, such as thermal conductivity of particles and fluid, porosity of packed bed, particles shape, particle distribution and thermodynamic properties of fluid

Influence of the tube and particle diameter and shape, as well as their ratio, on the radial heat transport in packed beds have been studied to obtain overall heat transfer coefficient for glass spheres, alumina cylinder and alumina rashing rings from experimental study by borkink *et al*.[2]

Also ,Demirel *et al*.[3] carried out experiments in a horizontal rectangular duct with a (160 cm) long packed section and heated from the top wall only by uniform heat flux while the bottom and all the side walls were insulated , the packing of polyvinyl chloride rashing rings (hollow cylinder) with the equivalent diameter of(3.85 and 3.26 cm) . And expanded pole styrene spheres with diameter of (4.8, 3.8 and 2.9 cm) were used in the duct with the equivalent diameter (D_{eq}= 21.7 cm).

Two kinds of working fluid (water and air)experimentally investigated by jiang *et al*.[4]to study convection heat transfer in sintered porous plate channels with the effects of fluid velocity ,particle diameter, type of porous media (sintered and non - sintered), and fluid properties on heat transfer enhancement , it is shown that local heat transfer coefficients in the sintered porous plate channels increase up to 15 times for water and 30 times for air .

Mueller *et al*.[5]developed a novel approach numerically pack spheres in cylinders with($D/Dp \geq 2$) , the packing algorithm used a simple sequential techniques that was based on a dimensionless packing parameter which includes both axial and radial variables in order to determine a spheres sequential placement with in a cylindrical packing structure . Hilal*et al*.[6] presented experimental results of forced convection heat transfer and pressure drop across (12.5 *12.5cm) square packed duct , the pad made of metallic wrapping coil unit with(0.98 porosity) , Reynolds number (40339 to 54797) and three boundary conditions of heat flux imposed on duct surface . it was found that Nusselt number in packed duct is arranging between 1.2 to 1.9 times higher than the empty duct at heating either all surface or top bottom surface of packed duct respectively .

From the above survey, it is seen that the experimental researches have been made for one rectangle packed inclination angle horizontal or vertical. Hence, this research is to examine the convection of heat transfer experimentally and numerically in a rectangular packed duct. So, the influence of Reynolds number, heat flux and duct orientation (horizontal& vertical) will be detected. In order to, determine the local and average Nusselt number for different Reynolds and Peclet number.

2- Experimental System :

The system in Ref [6] was used in this research, to investigate convection heat transfer of air in a packed duct. The system is illustrated photographically in Fig.(1) and schematically in Fig.(2) . it consists of an open air flow circuit , test section and measuring devices . Air is induced to the packet duct by a centrifugal fan and the inlet temperature is measured by digital thermometer with a precision of (∓ 0.1℃) . Two constructions of hydro-dynamically developing

entrance used in this work, (100cm) length with cross section (12*10cm) used in horizontal orientation and (90^0 elbow) fitting to vary the direction of flow.
Then (50cm) length with cross section (12*10cm) duct is used in vertical position before entering the test duct. Galvanized steel walls packed duct, with a length of (100cm) and cross section (12*10cm) ,packed and heated by uniform heat flux . The heating element consisted of a resistance heater made of a (0.5mm) diameter nickel-chrome wire isolated by ceramic spheres, electric power input is adjustable by avarice controlled the heater voltage and current.
The packed duct (100cm) length is divided into three parts (10 cm) long clamming section, (80 cm) test duct and (10 cm) exit section. The measuring parameters in this experiment are the duct wall temperatures, the inlet and outlet air temperatures, the flow rate and pressure drop through pad.
….The local temperature of the test duct was measured with eighteen K-type thermocouples , nine thermocouples was inserted into the top wall (0.5mm deep) along the axially centerline. Nine more thermocouple were inserted at the center line of the right wall of test duct .The inlet and outlet pressures were measured using mercury manometer connected to pressure tap at (20 to 80cm) from the packed duct entrance .(160) plastic spheres were packed randomly in order to be the pad of the test duct as shown in Fig. (3-A), with (4cm) diameter, (2.7gram) weight and (0.36 W/m.℃)thermal conductivity .Two layers of glass wool and sandwich panel were added to reduce conduction and radiation heat loss from the packed duct.
Air enters the packed duct at uniform velocity (u_i), its temperature at the inlet section (T_{in}) and (T_{out}) at the exit section. When thermal steady state reached, the amount of heat transferred from electric heater into air can be calculated as follow:

$$Q = \dot{m}(Cp_o T_{out} - Cp_i T_{in}) \quad \ldots\ldots (1)$$

….The energy supplied into electrical heater is determined by:

$$Q = I^2 . R \quad \ldots\ldots (2)$$

….All air properties (density, dynamic viscosity, and thermal conductivity) are taken at the average temperature (T_{av})

$$T_{av} = \frac{T_{in} + T_{out}}{2} \quad \ldots\ldots (3)$$

….Heat balance between Eq. (1) &Eq (2) was made to find the difference between them . Which did not exceed (4.5%) in this experiment.

….Duct heat flux

$$q_w = \frac{Q}{A_s} \quad \ldots\ldots (4)$$

A_s=Duct surface area (m^2)

$$q_w = \frac{Q}{2(H * L) + 2(W * L)} \quad \ldots\ldots (5)$$

Local heat transfer coefficient can be calculated from:-

$$h_x = \frac{q_w}{(\Delta T)_x} \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots..(6)$$

Where: $(\Delta T)_x$ represents the difference between local duct surface temp.(t_{sx}) and local air temperature (t_{bx}) at length (x) from duct entrance .

$$t_{bx} = T_{in} + \frac{x}{L}(T_{out} - T_{in}) \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots..(7)$$

$$h_x = \frac{q_w}{t_{sx} - t_{bx}} \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots..(8)$$

The local Nusselt number (Nu_x) is defined according to equivalent duct diameter $\left(D_{eq}\right)$ and effective thermal conductivity (K_e)

$$Nu_x = \frac{h_x . D_{eq}}{K_e} \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots..(9)$$

The effective thermal conductivity was computed from:

$$K_e = (1 - \Sigma)K_s + \Sigma K_f \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots..(10)$$

The porosity (Σ) of plastic packed bed used in this paper can be obtained from total volume (0.012 m^3), number of plastic spheres (160 unit) and (0,00535 m^3) volume pad ,the porosity is defined as:

$$\Sigma = \frac{V_T - V_{pad}}{V_T} = 0.5541 \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots..(11)$$

Comparison of experimental porosity from Eq (11) and Ref [7]:

Σ=0.3754+4.744 D_P= 0.56516 ………………………………………………… (12)

In practical calculations we applied (Σ= 0.5541). Then the deviation of (1.95%) between experimental data and the ones found in Ref [7] was determined.

Reynolds number (*Re*) and particle Reynolds number (Re_{Σ}) are defined as:

$$Re = \frac{u . D_{eq}}{\vartheta} \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots..(13)$$

$$Re_{\Sigma} = \frac{u . D_p}{\vartheta(1 - \Sigma)} \quad \dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots\dots(14)$$

To illustrate the combined effect of forced convection and thermal conductivity, Peclet number (Pe) can be calculated as follow:

$$Pe = \frac{u.D_P}{\propto_e} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(15)$$

3. Model Equations :

For modeling the heat transport in the experimental device, the problem was simplified by considering the followings: the flow is assumed to be two dimensional (symmetrical about centerline), the heat generated by the viscous effects negligible and there is a local thermal equilibrium between the particle and air [8] .With these assumptions, the energy equation is defined as follows:

$$\rho.Cp.u\frac{dT}{dx} = K_e\frac{d^2T}{dy^2} \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(16)$$

Where (y) refer to vertical length (duct height) and (x) refer to axial length (duct length).

With the boundary condition

$$\frac{dT}{dy} = \frac{q}{Ke} \quad at \quad y = 0 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots.(17)$$

$$T = T_i \qquad at \quad x = 0$$

The momentum equation used takes into account the boundary effects and inertial forces of air through particle pad [9]:

$$\frac{dp}{dx} = \frac{\mu}{\Sigma(y)}.\frac{d^2u}{dy^2} - \frac{\mu.u}{K(y)} - A(y)\rho u^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(18)$$

The permeability $K(y)$ of packed bed is:

$$K(y) = \frac{Dp^2\Sigma\,(y)^3}{175\{1-\Sigma(y)\}^2} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots..(19)$$

$A(y)$ Is known as the forchheimer constant:

$$A(y) = \frac{1.75\{1-\Sigma(y)\}}{\Sigma(y)^3D_P} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots(20)$$

The boundary condition on velocity and pressure are:

$$u = o \qquad at \qquad y = 0$$

$$\rightarrow \quad \frac{du}{dy} = 0 \qquad at \qquad y = \frac{w}{2} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (21)$$

$$p = p_o \qquad at \qquad x = x_o$$

Finite differences method with central differencing scheme shown in fig (3-B) is used for the numerical solution of the problem. After knowing the distribution of velocity and temperature inside the packed duct, air bulk temperature (T_b) can be found from:

$$Tb_{(x)} = \frac{\int uTdA}{\int udA} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (22)$$

Numerical local heat transfer coefficient (h_x) can be found from:

$$h_x = \frac{q_w}{T_{sx} - T_{bx}} \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (23)$$

Which is similar to Eq (8) in experimental calculation. Then local Nusselt numbers are obtained from Eq (9).
The average Nusselt numbers is determined as follows:

$$Nu_{av} = \frac{1}{L}\int_{x=0}^{x=L} Nu_x dx \qquad \ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots\ldots (24)$$

4. Result and Discussion :

Experiments were conducted with the heat flux range of (0.2) to (0.54 kW/m^2) and Reynolds number which is based on equivalent diameter range of (11433) to (6913)in two packed duct orientations horizontal and vertical position.
Fig (4) illustrates wall temperature distribution along the test duct measured at different heat flux and Reynolds number in horizontal position. General shape of the curves shows that temperature increases toward the end of the duct, the reason of this is that the air is gaining heat from the pad which in its turn is gaining heat from the inner surface duct. At the same axial position, temperature increases with increasing heat flux and decreasing Reynolds number.
Fig (5) shows similar plots for vertical position, which have similar trend as the one appears in Fig (5), Duct surface temperature at vertical position has lower value than horizontal position because the buoyancy force driven flow has the same direction of external forces driving the forced convection .as a result of this force the transfer of heat from the surface gets better and thus surface temperature is less in the case of vertical duct situation.
Fig (6) shows local Nusselt number in packed duct which is evaluated by equation (9) for vertical and horizontal position. it is seen that increasing Reynolds number , yield to increase turbulence and decrease thermal boundary layer thickness due to appearance of porous pad which causes high local Nusselt number for constant heat flux . Also it can be seen that vertical packed inclination gives the highest values of local Nusselt number.
Fig (6) also compares (Nu_x) calculated from the numerical analysis for air flow in packed duct with that obtained from experimental work .It is shown that the numerical value of local Nusselt

number is higher than in experiment at all conditions with maximum difference between them (27% - 20%) for horizontal and vertical at (Re=11433) (Q=0.36 kW/m^2) and illustrates increasing numerical (Nu_x) with increasing Reynolds number and heat flux.
The values of average Nusselt number are plotted in fig (7) for vertical and horizontal position, (Nu_{av}) in vertical is higher than horizontal position at (14%) at high difference and at (7%) at lower difference and resulting correlation are:

$$Nu_{av} = 0.453Re^{0.531} \text{ ---------------------------- } vertical\ position$$

$$Nu_{av} = 0.104Re^{0.683} \text{ ---------------------------- } horizontal\ position$$

The relation between average Nusselt number and particle Reynolds number which included the effect of porosity and particle diameter are plotted in Fig (8) , the experimental correlation are obtained by using a least square analysis for (Re= 4183 to 2535) :-

$$Nu_{av} = 0.786{Re_{\Sigma}}^{0.529} \text{ ---------------------------- } vertical\ position$$

$$Nu_{av} = 0.194{Re_{\Sigma}}^{0.691} \text{ ---------------------------- } horizontal\ position$$

Fig (9) shows the experimental data in term of Nu_{av} versus (Pe) ranging from (351) to (212) which taken into account the effect of forced convection and thermal conductivity which can be expressed in the following correlations.

$$Nu_{av} = 0.887Pe^{0.531} \text{ ---------------------------- } vertical\ position$$

$$Nu_{av} = 0.194\ Pe^{0.689} \text{ ---------------------------- } horizontal\ position$$

Fig (7) illustrate the (Nu_{av}) for two positions and the (Nu_{av}) for empty duct which was obtained from equation [10]:

$$Nu = 0.023\ Re^{0.8}{P_r}^{0.4}$$

It is shown that particle pad enhances the heat transfer through duct up to (1.1 ,0.91) times than empty duct at vertical and horizontal position respectively because of increasing turbulence , increasing contact surface area between air & pad and province of thermal boundary layer to form which leads to increase heat transfer.
The experimental average Nusselt number (Nu_{av}) obtained in this paper for (Re= 6913 to 11433) , (D_i /D_p) is equal to (2.7) and using air as a working fluid in rectangular duct are compared with the result obtained in Ref[11] for (Re=479 to 1694), (D_i /D_p =4.8) and water flow through circular tube . It is seen that all plots in Fig.(10) have the same trend (Nu_{av} increases as *Re* increases) . but ,the result from Ref [11] is higher (0.2 , 0.36) times than the present result at vertical and horizontal position respectively due to the high porosity and small particles sphere which leads to increase contact surface area between water and pad.

5. Conclusions :

In the present paper, the effects of rectangular packed duct orientation on forced convection heat transfer are investigated. Results show that:

1. Surface duct temperature increase with increasing heat flux and decreasing Reynolds number
2. Surfac duct temperature at vertical position is lower than horizontal.
3. Local Nusselt number at vertical position is higher than horizontal at all tests.
4. Numerical local Nusselt number is higher than experimental at maximum percent (27%, 20%) for horizontal and vertical position respectively.
5. Average Nusselt number at vertical position is higher than horizontal at (14%, 7%) at higher and lower differences.

6. References :

1. Yans J. , Wang J. , Bu s. , Zeng M. ,Wang Q. , and Nakayama A. ,"Experimental analysis of forced convective heat transfer in novel structured packed beds of particles", Chemical Engineering science , Vol.71 , pp. 126-137 , 2012.
2. Borkink J. G. and westerterpK .R. "Influence of Tube and Particle Diameter on Heat Transport in Packed Bed", AIchE Journal, Vol. 38, No.5, pp703-714, 1992
3. Demirel Y., Sharma R.M and Al-Ali H-H, "On the effective heat transfer Parameters in a packed bed," International Journal of Heat and Mass Transfer, 43, pp 327-332, 2000.
4. Jiang p.x , Li M. , Lu T. and Ren Z. , " Experimental research on convection heat transfer in sintered porous plate channels ", International Journal of Heat and Mass Transfer , 47 , pp. 2085-2096 , 2004.
5. Mueller G., "Numerically packing Spheres in Cylinders", powder Technology,159 ,pp. 105-110 , 2005.
6. Hilal K., Fadhl L. and Faraj S. , "Experimental Investigation of Heat Transfer and pressure Drop In Square Metal Packed Duct with Different Boundary Heating ", Eng. &Tech. Journal Vol. 30 No.6,pp 1082-1107 , 2012
7. Alkier R. and Grueter T., " Mass Transfer in packed bed Electrochemical Cells having Both Uniform and Mixed particle Sizes", J Electro Chemical Society Vol, 127, pp. 1085-1091, 1980
8. Pavel B. And Mohammad A., "An experimental and numerical study on heat transfer enhancement for gas heat exchangers filled with porous media", International Journal of heat and mass transfer, 47, pp. 4939-4952, 2004.
9. Cheng P.; Hsu C.T. and chowdhuryA. "Forced Convection in the entrance Region of packed channel with a symmetric heating, " Transactions of the ASMI, 110, PP 946 - 954, 1988.
10. Cengely.A. "Heat transfer: A Practical approach", McGraw-Hull, 1997
11. Hilal K.H "Fluid flow and Heat transfer characteristics in a vertical tube packed bed media", PH.D Thesis submitted to University of Technology/ Mechanical engineering department, 2004.

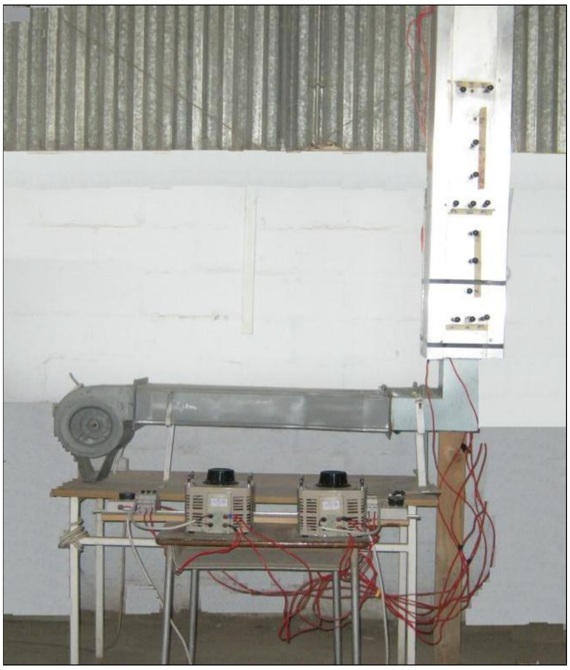

Figure (1-A)

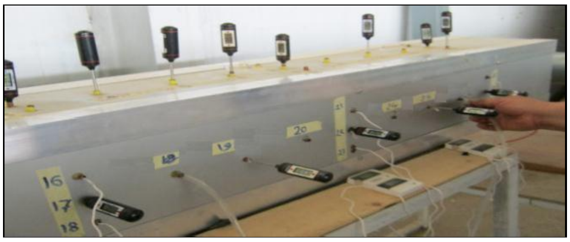

Figure (1-B)

Figure (1): photographic view of experimental apparatus

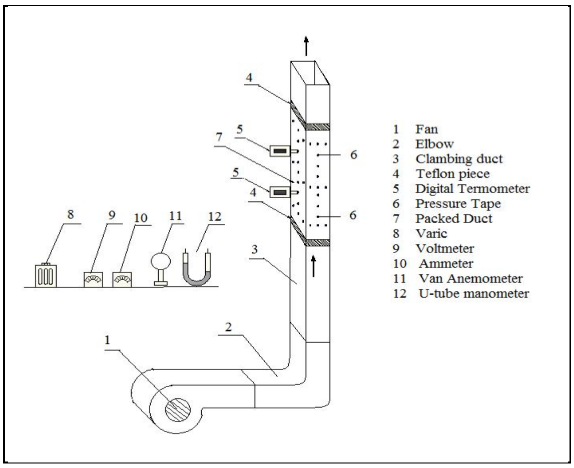

Figure (2): Experimental setup

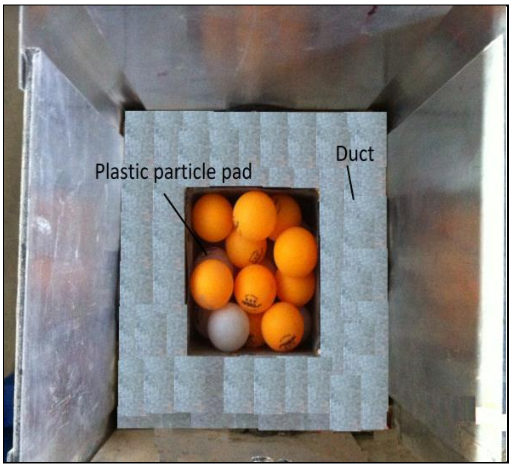

Figure (3-A): porous pad manufactured from plastic sphere

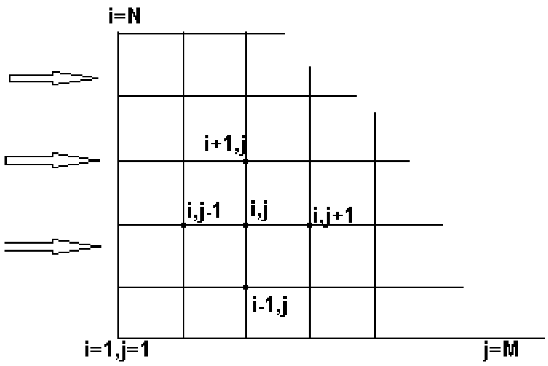

Figure (3-B): Coordinate system and Grid Distribution

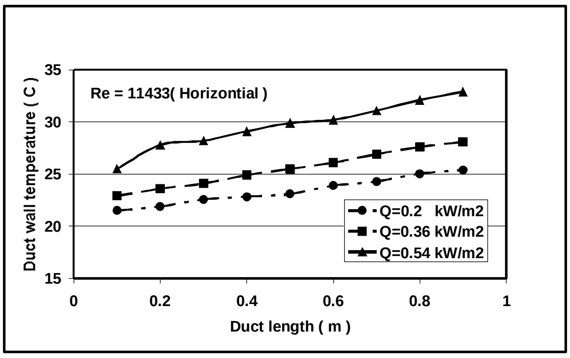

Figure (4-A)

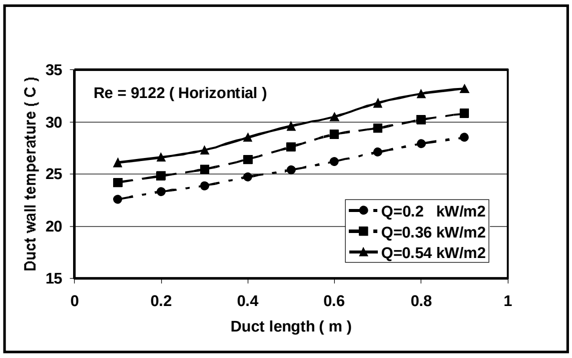

Figure (4-B)

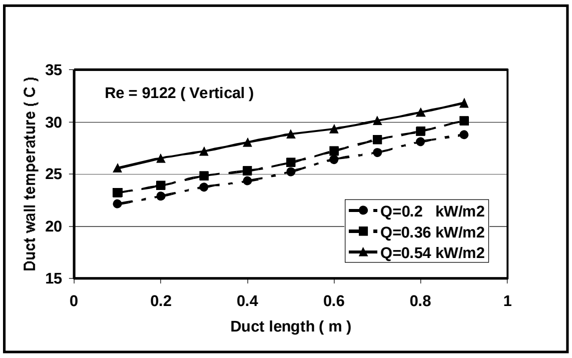

Figure (5-B)

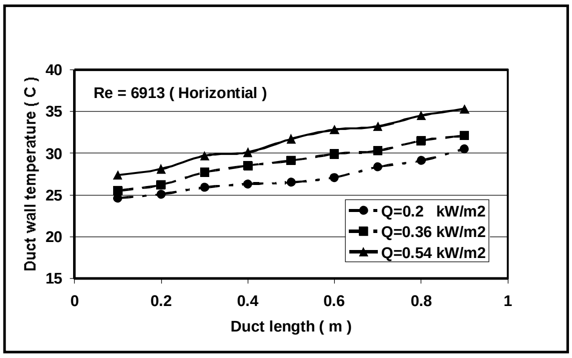

Figure (4-C)

Figure (4): Surface temperature recorded along duct length at horizontal position

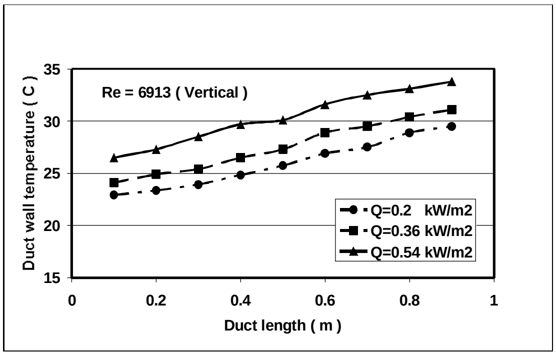

Figure (5-C)

Figure (5): Surface temperature recorded along duct length at vertical position

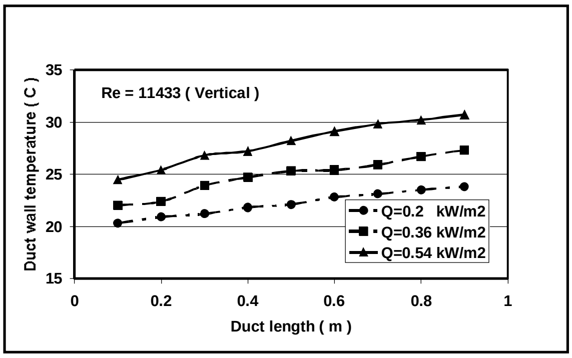

Figure (5-A)

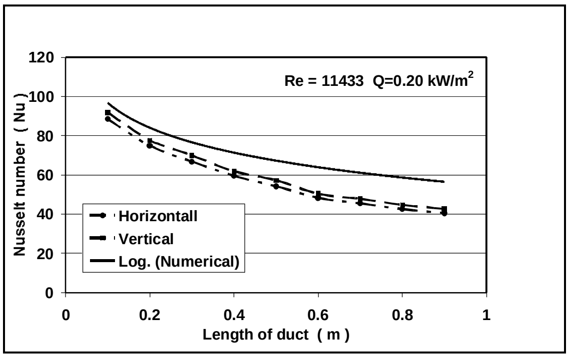

Figure (6-A)

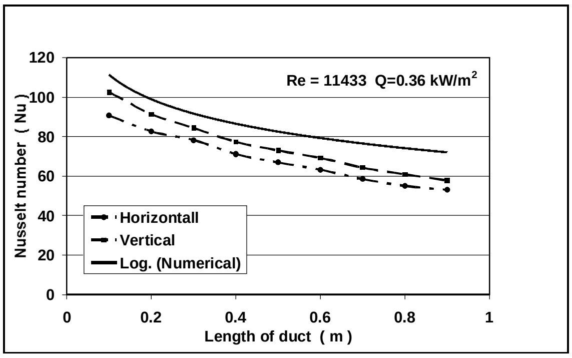

Figure (6-B)

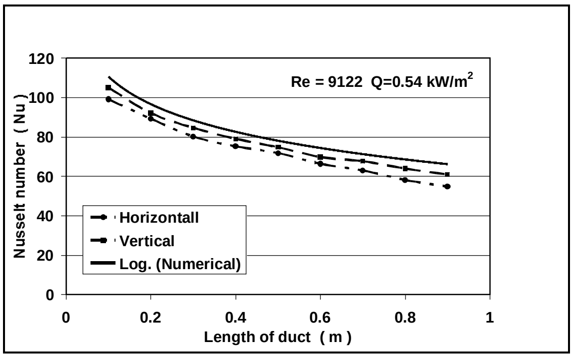

Figure (6-F)

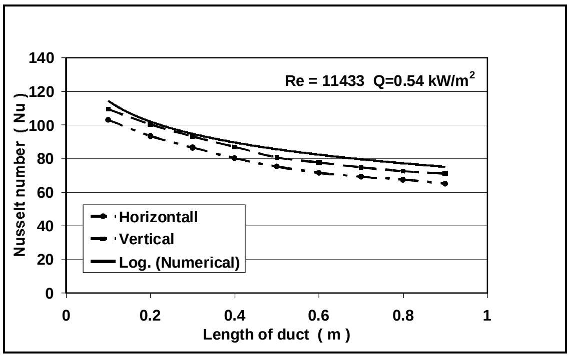

Figure (6-C)

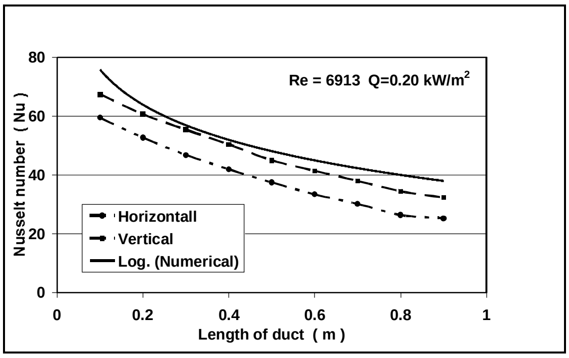

Figure (6-G)

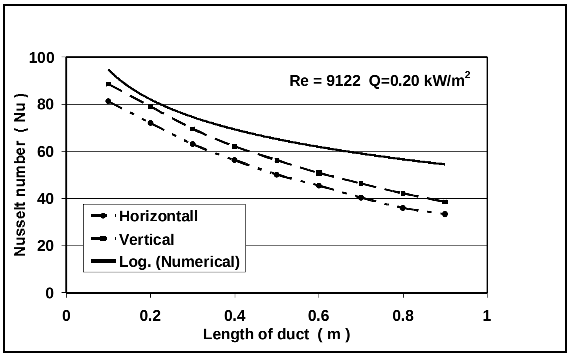

Figure (6-D)

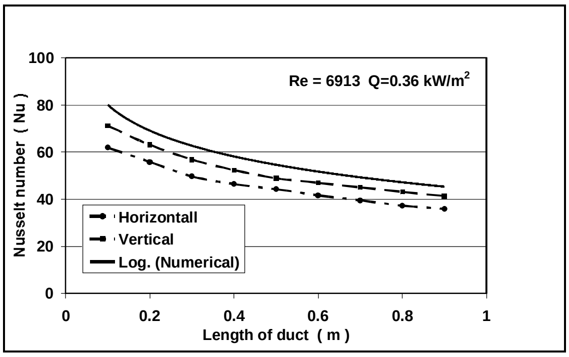

Figure (6-H)

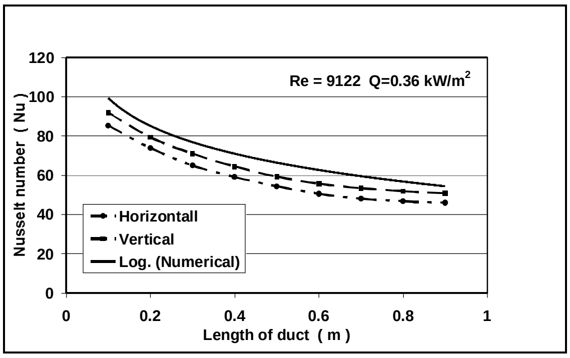

Figure (6-E)

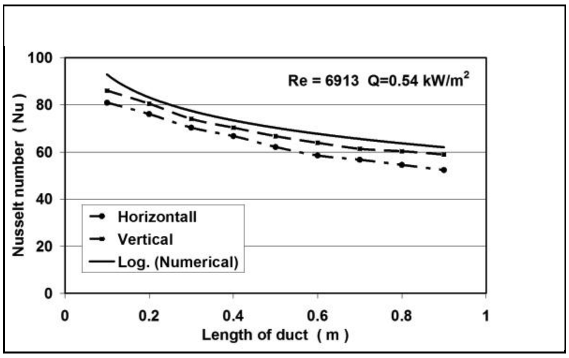

Figure (6-I)

Figure (6): Numerical and experimental Nusselt number at horizontal and vertical position

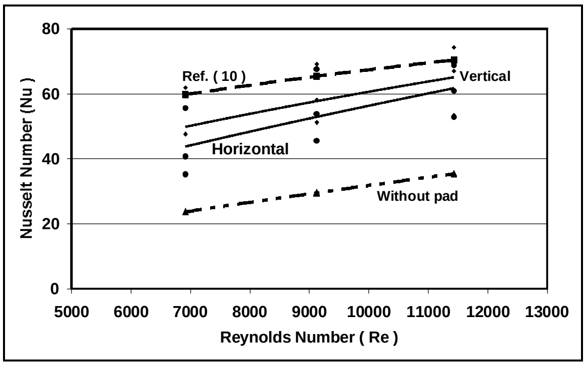

Figure (7): Average Nusselt number with Reynolds number for horizontal and vertical position

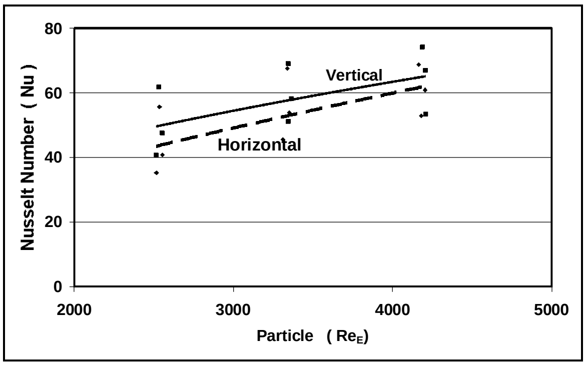

Figure (8): Average Nusselt number against particle Reynolds number for horizontal and vertical position

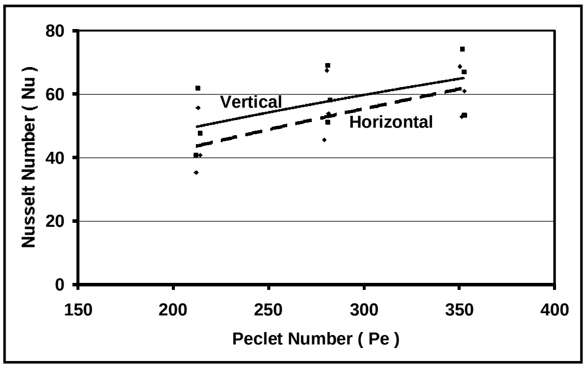

Figure (9): The variation of experimental (Nu_{av}) with Peclet number

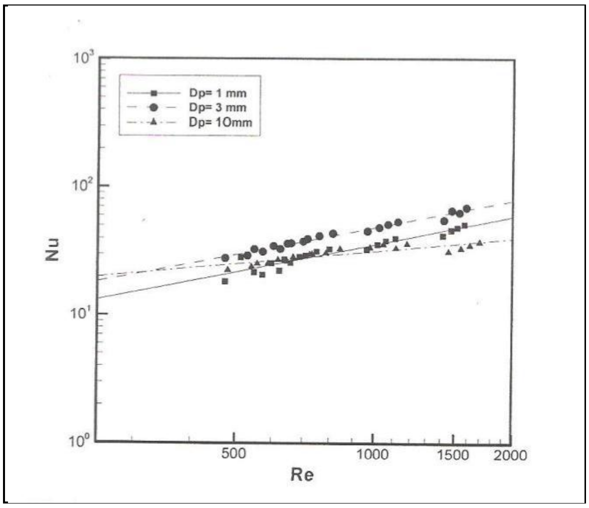

Figure (10): Average Nusselt number with Reynolds number from Ref [11]