Experimental Investigation of Heat Transfer and Pressure Drop in Square Metal Packed Duct with Different Boundary Heating

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Received on: 26/9/2011 & Accepted on: 5/1/2012

ABSTRACT

This paper presents experimental results of forced convection heat transfer and pressure drop across (12.5 * 12.5 * 100 cm) square packed duct. The pad made of forty- eight metallic wrapping coil unit with (0.98 porosity) and (26 W/m.°C thermal conductivity). The local surface duct temperature and local heat transfer coefficient distribution, Nusselt number, pressure drop and friction factor were measured for heat flux (0.56 to 2.73 kW/m^2) ,Reynolds number (40339 to 54797) and three boundary condition of heat flux imposed on duct surface . It was found that Nusselt number increases as Reynold number, heat flux and number of duct surface exposed to heat flux increases. Nusselt number in packed duct is to be (1.2 , 1.19) times higher than the empty ducts at heating all surface and top & bottom surface of packed duct respectively. Many empirical relation between Reynold number, Nusselt number and pressure drop obtained in this study.

Keywords: metal pad – heat transfer – square packed duct – pressure drop

دراسة عملية لأنتقال الحرارة وفقدان الضغط في وسط مسامي معدني ذو مقطع مربع باختلاف حدود التسخين

الخلاصة

تشتمل الدراسة على النتائج العملية لانتقال الحرارة بالحمل ألقسري وفقدان الضغط خلال مجرى مسامي ذو مقطع مربع بأبعاد (12.5 *12.5 *100سم). الوسط المسامي مكون من ثمانية وأربعون وحدة مكونة من أسلاك معدنية ملفوفة ، معامل التوصيل الحراري لها (26 واط/م .درجة مئوية) ومسامية الحشوة(0.98). توزيع درجات الحرارة الموضعي، معامل انتقال الحرارة الموضعي ،رقم نسلت ، فقدان الضغط ومعامل الاحتكاك تم إيجادها لمدى فيض حراري (0.56 -2.73 كيلوواط/م^2) ورقم رينولدز (40339 -54797) وثلاث حالات من التسخين بفيض حراري ثابت على أسطح المجرى المسامي.

بينت النتائج العملية زيادة رقم نسلت مع زيادة رقم رينولدز والفيض الحراري وعدد أسطح المجرى المسامي المعرضة للتسخين ان رقم نسلت في المجرى المسامي يزداد (1.19 ،1.2) مرة أعلى من المجرى الفارغ عند تسخين كل أسطح المجرى المسامي والسطح الأعلى والأسفل فقط من المجرى المسامي على التوالي .تم أيجاد عدد من المعادلات التجريبية بين رقم رينولدز ورقم نسلت وفقدان الضغط في هذا البحث .

Nomenclature

A_s	Surface area	m^2
d_{eg}	Hydraulic diameter	m
F	Friction factor	----
h_x	local heat transfer coefficient	$W/m^2.\ °C$
h	Average heat transfer coefficient	$W/m^2.\ °C$
H	Duct height	m
I	Electrical current	Amp.
K_f	Air thermal conductivity	W/m. °C
L	Duct length	m
Nu_x	Local Nusselt number	----
Nu	Average Nusselt number	----
ΔP	Pressure drop	Pa
Pr	Prandtl number	----
q_w	Wall heat flux	W/m^2
Q	Convection heat transfer	W
Q_{cond}	Conduction heat loss	W
Q_T	Total input power	W
Re	Reynolds number	----
t_{sx}	Local surface temperature	°C
t_{bx}	Local bulk air temperature	°C
t_{mx}	Local mean film temperature	°C
U_i	Air velocity at duct exit	m/s
$V^{\cdot}$	Volumetric flow rate	m^3/s
V_T	Test duct volume	m^3
V_p	Packing volume	m^3
V	Voltage	volt
W	Duct width	m
W_{pump}	Pumping power	W
ρ	Air density	kg/m^3
ε	Porosity	----
υ	Kinematics viscosity	m^2/s

INTRODUCTION

Heat transfer in porous media has been the subject of numerous investigations due to increasing interest in chemical catalytic reactors, building thermal insulation, heat exchanger, petroleum reservoirs, geothermal operations, packed-sphere beds and absorption towers [1].

According to [2], a porous medium provides a large thermal dispersion and solid-fluid contact area many times greater than the duct surface area, thus greatly enhancing the heat transfer. A number of recent studies incorporated porous-media forced convection in duct of various shapes analytically include those Haji-Sheikh *et al* [3],Hooman *et al* [4] and Haji-Sheikh *et al* [5].

An experimental investigation was conducted by [6] to examine the feasibility of using a high-conductivity porous channel as a heat sink for high-performance forced air cooling in microelectronics, the porous channel (5*5*1 cm) was made of sintered bronze beads with two difference mean diameters dp=0.72 and1.59 mm. The local wall temperatures distribution , inlet and outlet pressures and temperatures , and heat transfer coefficients were measured for heat flux of 0.8,1.6,2.4 and 3.2 W/ cm^2 with air velocity ranging from 0.16 to 5 m/s and inlet air pressure of 1-3 atm ,the results showed that the high conductivity porous channel enhances the local heat transfer rate and reduced maximum wall temperature of the heat sink . For example, the forced air heat transfer coefficient can be increased from 0.01 to 0.5 W/cm.C° by using the porous heat sink of dp=0.72 mm.

Also an experiment had been carried out for forced convection of air in a rectangular duct with large rashig rings hard plastic packing by Makkawi [7] .The experiments were conducted for a constant heat flux ranging from 50 to 550 W/m^2 and Reynolds number ranging from 1000 to 5500 in a large rectangular channel of an aspect ratio W/H=8 , it was found that the value of Nu increases considerably in the packed channel as Re increases, a correlative equation for Nu ($Nu = 0.8334\ Re^{0.6836}$) and Nu was about 6 times higher in the packed channel than in the empty one.

For beds of monosized spheres ,Ribeiro *et al* [8] presented new experimental results on the average bed porosity and pressure drop data for water flowing in the bed was obtained for $3 < Re_p < 379$ and $3 < D/dp < 17$. The results showed that the average bed porosity increases as D/dp decreases and the pressure gradient of the fluid across the bed increases when both the particle Re and the ratio D/dp increase.

Also ,heat transfer enhancement using porous metal foams depends on both the cellular structure of the foam material and the thermal properties of the metal foam . Metal foam thermal conductivity is dependent upon the overall density of the piece and the metal from which the foam is made [9] . For that this paper presents an experimental investigation of forced convection heat transfer and pressure drop across a square packed (W/ H=1) and (W/L= 0.125) with pad of metallic wrapping coil (0.98 porosity) as metal foam . Three condition of constant heat flux supplied to square duct surface and Reynolds number range (40339 to 54797) are investigated to find Nusselt number and heat transfer coefficient in the metallic packed bed.

EXPERIMENTAL A APPARATUS

An experimental apparatus, which has been designed and employed in this paper is diagrammatically in Fig (1) and photographically in Fig (2) .

The air blown by a centrifugal fan, enters (1.5m) length of square duct with cross section (12.5 *12.5 cm) to obtain hydro- dynamically developing entrance then passed through a test section , the test section has the same cross section (12.5 *12.5 cm) and length (100 cm) . The duct walls were made of (1 mm) galvanized steel . The heat transfer section of (80 cm) in length is preceded by entrance section of (10 cm) in length and followed by (10 cm) length an exit section .Two distributor stainless steel are placed at entrance and exit of the test duct to hold the packing in place and to get a better distribution of the inlet air flow . Also two piece of Teflon represented at entrance and exit of the test duct to decrease the heat losses from ends, a schematic of test section is shown in Fig(3) .

The duct was heated electrically by four electrical heaters distributed spreadtly on each side of the duct . To have different case of constant heat flux condition on the packed ducts which are investigated in this paper, the heaters arrangement are:

(1) Constant heat flux is supplied from the top while the other walls are insulated.
(2) Heat flux are supplied from the top and bottom while the side walls are insulated.
(3) Heat flux are supplied from all walls of duct.

Each heater consisted of a (0.5) mm in a diameter nickel-chrome wire electrically isolated by ceramic tubs in a different length , (25) loops of wire spread out and fixed on the surface by thermal defcon adhesive as in Fig(4) . The heaters voltage and current were measured by digital multimeters, and the electric power input was controlled by a varic connected to the heaters . A (0.25 mm) thickness asbestos tape wounded strongly around the heater to confirm it and to decrease the heat loss shown in Fig(5). Then, insulated with two layers of glass wool(100 mm) thickness (Fig(6)) and (4 mm) aluminum sandwich panel which outer surface is reflective to reduce the conduction and radiation heat transfer from the test section (Fig(7)) .Three thermocouple inserted after the first layer of glass wool along the heated section. Also, three thermocouple imposed before the aluminum sandwich panel at the same position of the preceding thermocouple . The heat losses from the side wall test section was estimated to be less than (6%) of the power input to the heater by using average measured temperature drop and thermal conductivity of glass wool.

Twenty-eight K-type probe digital thermocouples of accuracy (± 0.1 °C) are distributed transversely and longitudinally at the one side plate and top plate of the test duct . In transverse direction six thermocouples are placed at each position (10 ,50 and 90 cm) from inlet test duct ,(3,6.25 and 9.5 cm) from bottom duct edge on the side plate and at (3, 6.25 and 9.5 cm) on top plate. Also four thermocouples are located at the mid-width of the side plate at (30,40,60, and 70 cm) from inlet-duct. In the mid-width of top plate, six thermocouples are placed at (20, 30,40,60,70 and 80 cm) from entrance duct. Details of the thermocouples distribution are shown in Fig(8).

All the thermocouples are fixed at a drilling (V) holes along duct wall , threaded through a (10 cm) and covered with asbestos insulation tube to reduce conduction losses to surrounding and to protect the thermocouple from heat flux effects .

Two pressure taps are inserted along the duct side wall to measure the pressure drop caused by the packing. The taping holes are located at (20 and 80 cm) respectively and have (1 mm) diameter and obtained from drilling cleanly with the axis normal to the duct side wall .Copper tubes (6 mm) diameter and (10 cm) length are welded normally at these taps for obtaining " T " joint to connect them with a sensitive U-tube manometer filled with mercury which used to measure experimentally the pressure drop with in the packed bed duct.

The air flow rate is regulated by slide gate connected at the discharge of the centrifugal fan, any excess air flow through a by-pass duct with slide gate is located at the outlet of fan. After passing the packed test duct, the velocity of the outlet heated air is measured by vane anemometer with precision (± 0.01 m/s). The volumetric air flow is obtained by multiply the measured velocity with the duct cross section area to obtain the desired flow rate through the experiment. To check the steady state of air flow rate, air velocity is measured several times during the test .

PACKING MATERIAL

The packed bed was made of metallic wrapping coil (MWC) (1 mm) diameter as shown in Fig(9). Forty-eight of (MWC) were used to pad the test duct to form the packing which arranged regularly inside the duct. The weight of each unit (MWC) was measured by sensitive balance with a precision of (± 0.01 gram), the average weights of the (MWC) unit are found to be (51.64 gram).

A measuring cylinder with a capacity of (250 ±1 mL) was used to find the volume of each unit of (MWC). A known volume of water was placed inside the cylinder then one unit of (MWC) previously weighted was add. By measuring the total volume occupied by the water and by (MWC) unit and subtracting the initial volume of the water ,the volume of (MWC)unit could be obtained .This measurement is repeated for all unit of (MWC) used in this work .The average volume of the (MWC) unit is found to be (6.5 mL) .Density was measured using the volume displacement method ,by dividing the weight of the (MWC) unit to the volume of the water displaced. The average density is found to be (7945 kg/m^3) ,and by comparing this average density with the density of various kind of steel in Ref[10 , 11] ,which indicate that (MWC) is made from Nickel steel (10 Ni) with thermal conductivity (k=26 W/m.K) . The porosity of (MWC) packed bed used can be calculated as follow :

Test duct volume (V_T) =100*12.5*12.5=15625 cm^3 =0.015625 m^3

Number of (MWC) unit used in packing = 48

Total packing volume (V_P) =48* 6.5* 10^{-6} =0.000312 m^3

Porosity (ε) =(V_T- V_P) / (V_T) = (0.015625 – 0.000312) / (0.015625) = 0.98

EXPERIMENTAL PROCEDURE

The procedure to obtain experimental data was repeated for each test run .Three cases of constant heat flux condition supplied on the packed duct are investigated in this paper . For special case of heat flux, the fan was switched on to flow air through the packed , the required mass flow rate was adjusting by slide gate . The variac is adjusted for the required input power to the heater to give the required heat flux. The apparatus is left for (2- 4 hr) to reach thermal steady state condition , this period depend on the a mount of heat flux and mass flow rate. During each test run ,the reading of thermocouples ,air velocity at the exit of packed duct and heater current and voltage are recorded .

The experimental convection heat transfer coefficient from duct surface to air passed through packed duct , Nusselt number and Reynolds number are found as follow:

The total input power supplied to test duct is:

$$Q_T = V * I \qquad \ldots\ldots (1)$$

The convection heat transferred from the duct surface:

$$Q = Q_T - Q_{cond} \qquad \ldots\ldots (2)$$

Where Q_{cond} is the conduction heat loss from the external packed duct surfaces and the duct ends that not exceed (6 %) during all experiment due to good insulation .

The wall heat flux can be calculated by :

$$q_w = Q / A_s \qquad \ldots\ldots (3)$$

Where A_s = duct surface area that exposed to heating condition (m^2)

The local heat transfer coefficient can be obtained as :

$$h_x = q_w \ / \ (t_{sx} - t_{bx} \qquad \ldots\ldots (4)$$

The local Nusselt number (Nu_x) can be determined by :

$$Nu_x = h_x * d_{eg} \ / \qquad \ldots\ldots (5)$$

The hydraulic diameter (d_{eg}) for square or rectangular duct [12]:

d_{eg} = 4 (cross sectional area for flow) / (wetted perimeter)

$$d_{eg} = 4 \ (W * H) / 2 * (W+H) \qquad \ldots\ldots (6)$$

All the properties are evaluated at the film air temperature from Ref [12]:

$$t_{mx} = (t_{sx} + t_{bx}) / 2 \qquad \ldots\ldots (7)$$

Reynolds number (Re) can be calculated as :

$$Re = (U_i * d_{eg}) / \upsilon \qquad \ldots\ldots (8)$$

The average heat transfer coefficient (h) is found as :

$$h = 1/L \int_{X=0}^{X=L} h_x dx \qquad \text{.......... (9)}$$

Also, the average Nusselt number (Nu) can be obtained by integration as :

$$Nu = 1/L \int_{X=0}^{X=L} Nu_x dx \qquad \text{.............. (10)}$$

Another quantity analysis the flow through packed bed is the pressure drop ,which can be expressed as [12] :

$$\Delta P = F \frac{L}{d_{eg}} . \frac{rU_i^{\,2}}{2} \qquad \text{........... (11)}$$

Where F is the friction factor .

The required pumping power to maintain the flow through the pad is determined from [12] :

$$W_{pump} = V^{\cdot} \ \Delta P \qquad \text{.......... (12)}$$

Table (1) lists the experimental condition used in present study to obtain empirical correlation in term of heat transfer and flow variables in metallic packed bed with (0.98 porosity)

$$Nu = C_1 \ Re^{m1} \qquad \text{........ (13)}$$

$$F = C2 \ Re^{m2} \qquad \text{....... (14)}$$

RESULTS AND DISCUSSION

The experiments were carried out under a range of variables parameter described in Table (1) to investigate heat transfer through square cross section metallic packed duct with Reynolds number range between (40339 to 54797) and constant heat flux (2.73 to 0.54 kW/m^2) placing on top surface , top(T) and bottom (B) or all surface in various conditions .

TEMPERATURE DISTRIBUTION

The local top and side packed duct wall temperature distribution along the flow direction are presented in Fig. (10) to Fig. (12) .In general ,the temperature increases as the heat flux increases for same Reynolds number ,while the surface temperature decreases as the Reynolds number increases for same heat flux due to increasing air flow which lead to more turbulence and heat transfer rate . It is seen that all surface temperatures increase sharply at the duct entrance then the increasing become gradual toward the exit duct because of higher heat transfer rate occurred at thermal entrance region then decreases along the axis direction of the duct until reaching thermal fully developed region. The difference between the top and side surface temperature is small at the heating all duct surface, It is observed that this difference increases when duct surfaces imposed to heating condition are decreased .For example, at(Re=54797) and (heat flux =2.73 kW/m^2) ,the difference are (7.1 ,55.3 and 54.6 °C) at all surface heating , top (T)and bottom (B)heating and top heating respectively .

LOCAL HEAT TRANSFER COEFFICIENT (H_X)

Fig (13) to Fig (15) show the values of the local heat transfer coefficient in metallic packed bed plotted vs the packed duct length at different Reynolds number and heat flux . It can be seen that (h_X) have a high value at the entrance of packed duct due to high temperature difference between the air passes through the duct and duct surface temperature ,then (h_X) will be decreased along the axial direction of duct .Increasing Reynold number will increasing the flow velocity and causes a reduction in the thickness of the thermal boundary layer and thermal resistance to heat transfer.

Also , It's show that the largest local heat transfer coefficient for the same Reynolds number with constant heat flux occurred when heating the duct from all surface, top (T) and bottom (B) surface and top surface only respectively ,this is due to the large surface area of contact between the flow and duct surface.

LOCAL NUSSELT NUMBER (NU_X)

The variation of local Nusselt number (Nu_X) with packed duct length is shown in Figs(16 to 18). It is evidenced that the local Nusselt number has the same trend as local heat transfer coefficient shown in Figs(13 to 15). It is clear that (Nu_X) values increase as Reynolds number and heat flux increased for the same imposed surface heat flux on the packed duct .Figures show that (Nu_X) values decrease with the axial direction of the duct due to increase thermal boundary layer thickness and increase the difference of (surface to bulk air temperature) . For the same heat flux and (Re) number ,Nusselt number increase with increasing number of heating surface of the packed duct , heat is transferred mainly by molecular conduction near the heat transfer surface depending on the temperature of the duct surface ,in other words, the (Nu_X) for heating all surface duct is greater than that for heating top (T) and bottom (B)surface of the duct and the last is greater than that for heating only top surface at the same experimental conditions.

CORRELATION OF AVERAGE PACKED DUCT SURFACE TO FLUID NUSSELT NUMBER

Equation represents the convection heat transfer through packed bed can be expressed in terms of the average Nusselt number (Nu) and Reynolds number (Re) in the form:

$$Nu = C_1 \; Re^{m1} \qquad \ldots\ldots (13)$$

The values of Nusselt number for metallic packed bed at heating all surface of duct, top &bottom surface only and top surface only of the present work is plotted in Fig (19). All the plots are straight line appear increasing (Nu) with increasing (Re) number. It can be seen that inserting constant heat flux on all surface gives the highest values of Nusselt number , due to the high temperature difference between the air bulk temp. and surface temperature at all side of duct, then the higher values obtained when heating top &bottom surface only which are larger than heating top surface only.

Table(2) shows percentage of increment in (Nu) number from the results of top heating only . The resulting correlation of average Nusselt number and Reynolds number represented by the values of constants (C1) and (m1) are shown in Table (3).

COMPARISON WITH FORCED CONVECTION HEAT TRANSFER IN EMPTY SQUARE DUCT

The experimental Nusselt number results for metallic packed bed compared with the results obtained from equation [12] to estimate the benefit of using (MWC pad) in the empty duct :

$$Nu = 0.023\, Re^{0.8}\, Pr^{0.4} \qquad \ldots\ldots\ldots (15)$$

This relation gives Nusselt number for fully developed turbulent flow in smooth tube and in noncircular tubes used by replacing the diameter (d_{eg}) in finding Reynolds number by hydraulic diameter ($d_{eg} = 4\, A_c / P$) , this relation is plotted in Fig(19) with the experimental results for different boundary conditions .

It can be seen that as (Re) increases (Nu) increases . At heating all surface ducts Nusselt number is increase (1.20) times higher than the empty duct. It also shows Nusselt number is increase (1.11) times higher than the empty duct at heating top&bottom surfaces only. This is due to high thermal conductivity of metallic pad and turbulence caused by the presence of pad which tend to prevent the build up of thermal boundary layer of fluid next to the heating surface.

When heating impose only on top surface of packed duct, the empty duct is higher than the packed duct (1.23) times because all surface of empty duct is exposed to constant heat flux according to Eq.(15),high temperature difference between air and surface temperature will obtained to increase heat transfer from surface duct to air.

PRESSURE DROP

The pressure drop of air flow through the metallic bed is studied in this study because the pressure drop is important in determining the energy requirements to pump a fluid any given bed.

Fig(20) illustrates the pressure drop vs Reynolds number for different boundary condition of heating. It is shown that heat flux imposed on duct surfaces has no effect on pressure drop and the pressure drop increase as Reynolds number increase due to increasing turbulent effect occurred in the metallic pad.

The correlation obtained from experimental pressure drop results is:

$$\Delta P = 0.0067\, Re^{0.8072} \qquad \ldots\ldots\ldots (16)$$

In order to compare friction factor (F) of metallic packed bed (0.98 porosity) used in this study Eq.(16) used to find friction factor for fully developed turbulent flow in smooth circular tube [12]:

$$F_{empty}= 0.184\, Re^{-0.2} \quad \ldots\ldots\ (17)$$

Experimental friction factor (F) and friction factor in empty duct were plotted in Fig(21) , it is shown that the packed duct friction factor is greater than the empty duct at (86%) maximum and (81%) minimum difference respectively. The friction factor in the present study can be correlated by the following equation:

$$F =1650\, Re^{-1.0932} \quad \ldots\ldots\ (18)$$

CONCLUSIONS

An experimental study of heat in metal packed bed is carried out to analyze the influence of different heating boundary conditions imposed on duct surface, heat flux, and air flow rate on the heat transfer coefficient and pressure drop of air passes across bed formed by metallic wrapping coil (MWC) with (0.98 porosity).

The following conclusions can be drawn for the conditions under study:

1- Top and side surfaces duct temperature increase as the heat flux and Reynolds number decrease.
2- The temperature difference between top and side surfaces have larger value at heating only top surface and decrease to lowest value at heating all duct surfaces.
3- Local heat transfer coefficient decrease along the axial direction of duct, increase with increasing Reynolds number, heat flux and number of duct surface exposed to constant heat flux.
4- Nusselt number increase with increasing Reynold number and increment percentage of Nusselt number between all surface duct heating and top surface duct heating only are (22.44% , 30,68% , 40.97%) at Reynold number (54799 , 48222 , 40339) respectively and (16.94% , 24.115% , 33.05%) the increment percentage of Nusselt number between top & bottom surface duct heating and top surface duct heating only at the same (Re) range.
5- Nusselt number in metal packed bed increase at (1.2 , 1.1) times higher than empty duct when packed duct heating at all surface and top & bottom surface respectively, but empty duct increase at (1.23) times higher than packed duct heated at top surface only.
6- Pressure drop across the packed duct increase as Reynolds number increase.
7- The following empirical relation obtained in this study :

   $Nu = 0.0194\, Re^{0.8172}$ ……. Were all surface heated

   $Nu = 0.0023\, Re^{1.0064}$ …….. Were top &bottom surface heated

   $Nu = 9E\text{-}07\, Re^{1.7084}$ ……. Were top surface heated

   $\Delta P =0.0067\, Re^{0.8072}$

   $F =1650\, Re^{-1.0932}$

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Table (1) Experimental boundary condition

Case	I (Amp.)	V (Volt)	Vel. (m/s)	Heating condition
1	2	154	8	All
				Top&Bottom
				Top
2	1.5	100		All
				Top&Bottom
				Top
3	1	64		All
				Top&Bottom
				Top
4	2	154	7.1	All
				Top&Bottom
				Top
5	1.5	100		All
				Top&Bottom
				Top
6	1	64		All
				Top&Bottom
				Top
7	2	154	6	All
				Top&Bottom
				Top
9	1.5	100		All
				Top&Bottom
				Top
10	1	64		All
				Top&Bottom
				Top

Table (2) Increment percentage of Nusselt number

Re	Nu (Top only) (1)	Nu (Top&Bottom) (2)	Nu (All surface) (3)	Incre. Percentage between 2&1	Incre. Percentage between 3&1
54797	112.184	135.148	144.648	16.94%	22.44%
48222	90.177	118.835	130.301	24.115%	30.68%
40339	66.475	99.295	112.616	33.05%	40.77%

Table (3) Empirical constant for Eqs, (13)

Heating condition	C1	m1
All surface heated	0.0194	0.8172
Top &Bottom surface heated	0.0023	1.0064
Top surface heated	9E-07	1.7084

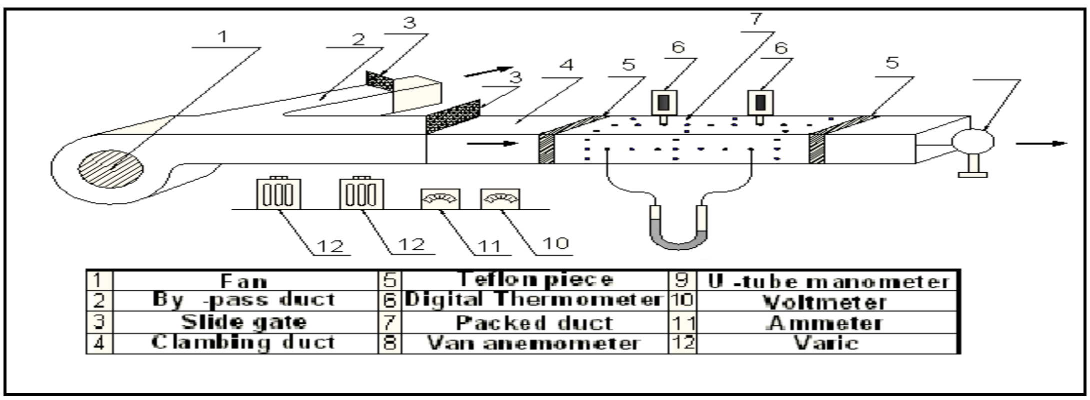


Figure (1) Schematic of setup

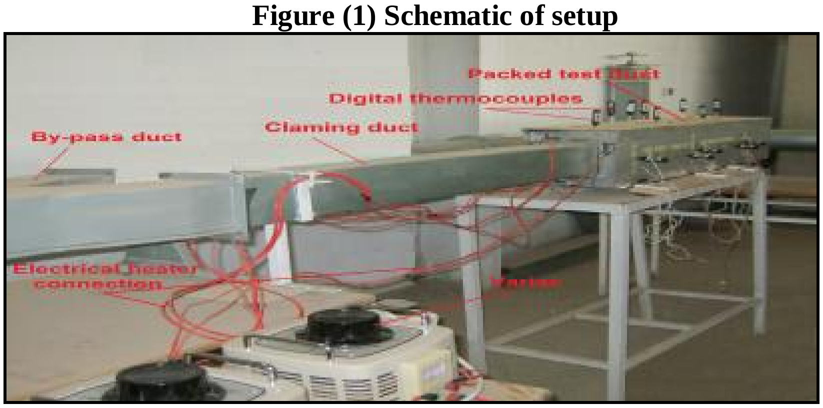


Figure (2) Photographic of setup

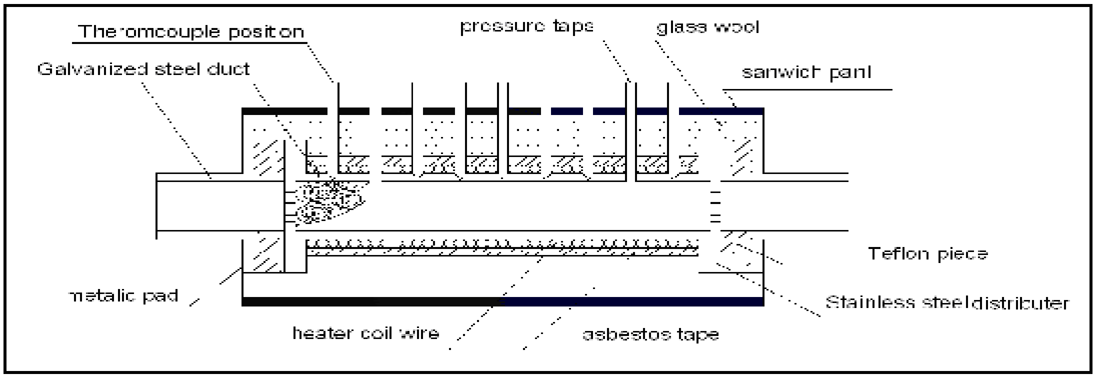


Figure (3) Schematic of test duct

Figure (4) Heating arrangement on test duct

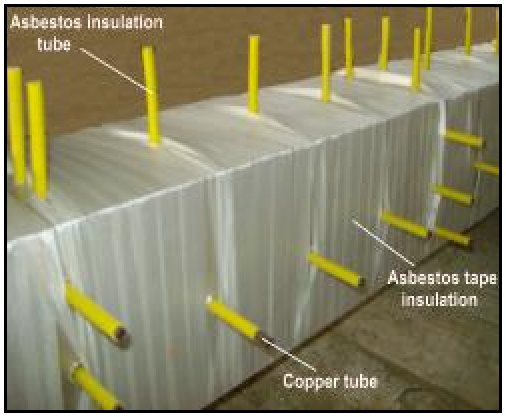


Figure (5)Test duct insulation with asbestos tape

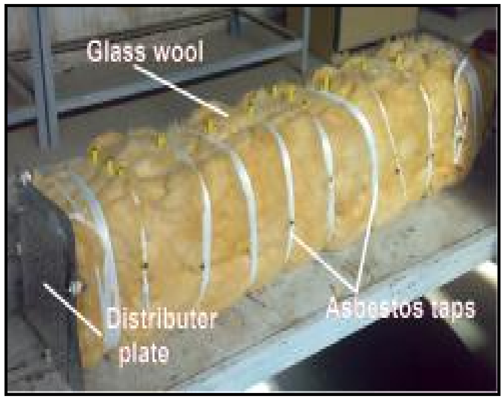


Figure (6) Test duct insulation with glass wool

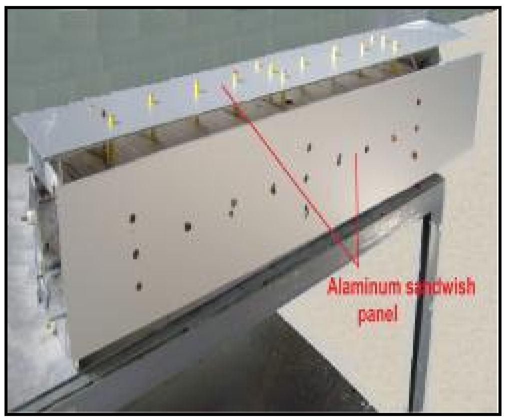


Figure (7) Test duct covered with aluminum sandwich panel

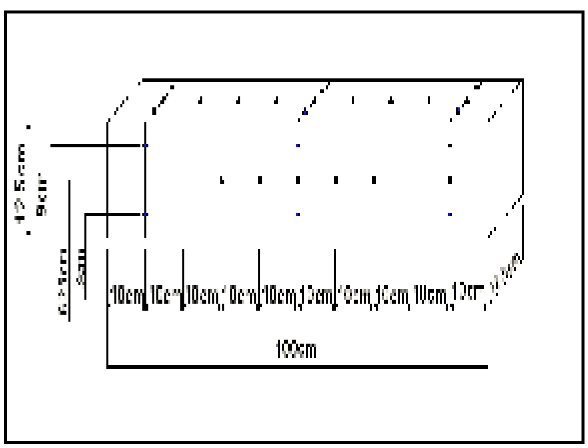
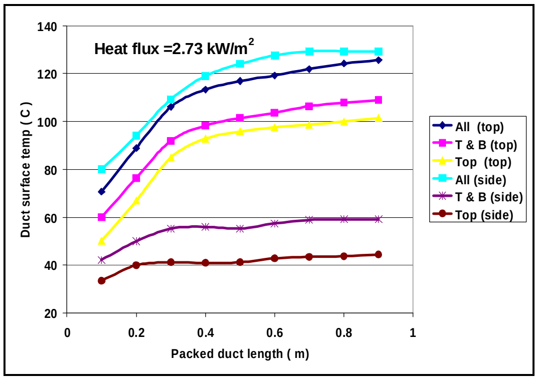
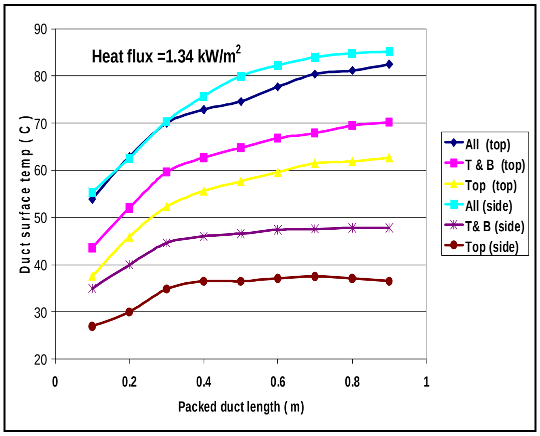


Figure (8) The position of thermocouples on test duct

Figure (9) Metallic wrapping coil



(a)



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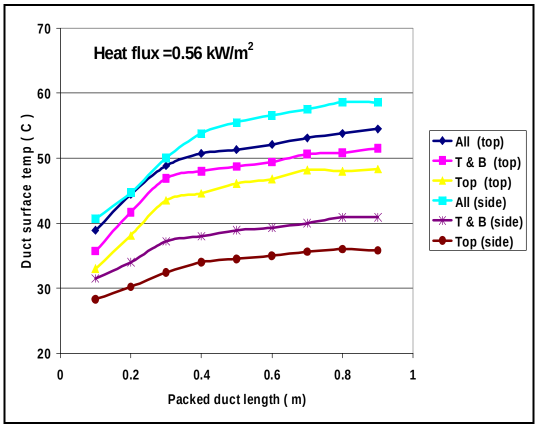
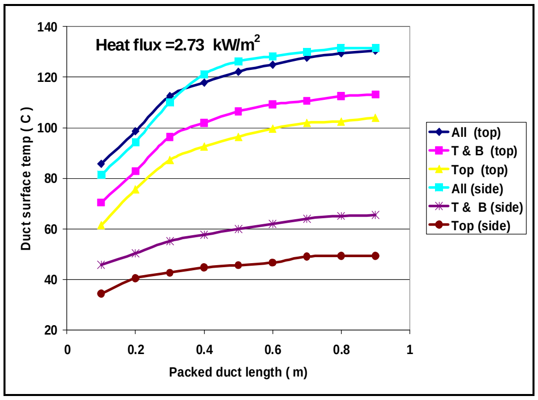
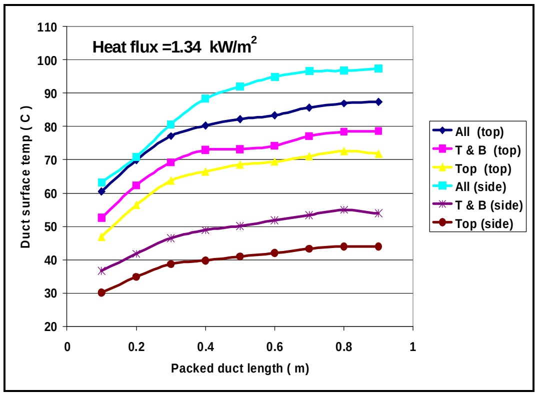


(c)

Figure (10) Duct surface temperature with duct length for different heat flux (Re=54797)





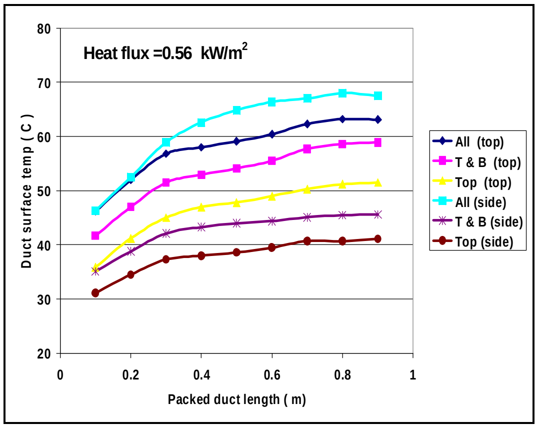
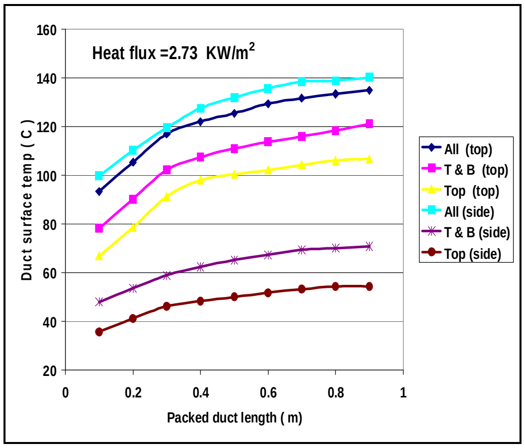
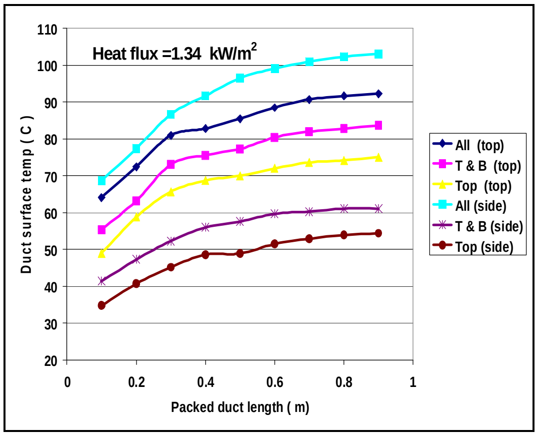


Figure (11) Duct surface temperature with duct length for different heat flux (Re=48222)





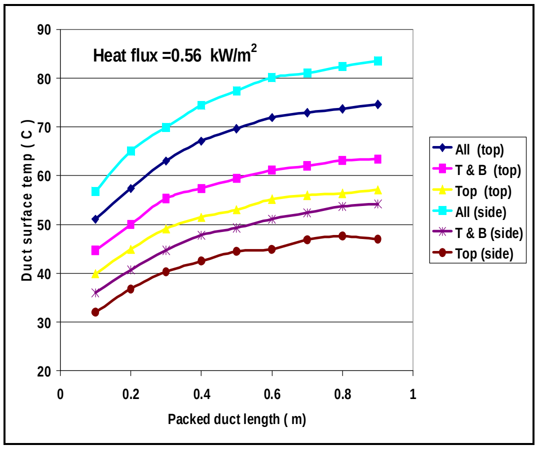
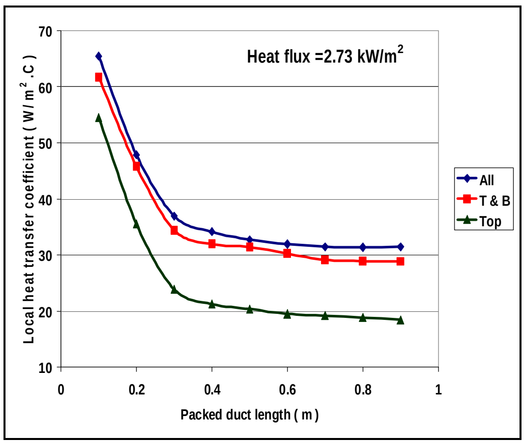
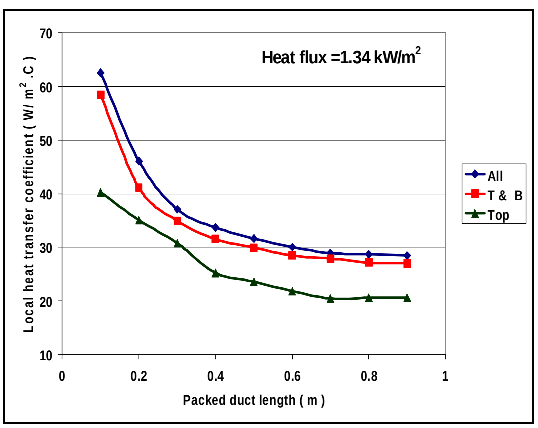


Figure (12) Duct surface temperature with duct length for different heat flux (Re=40339)





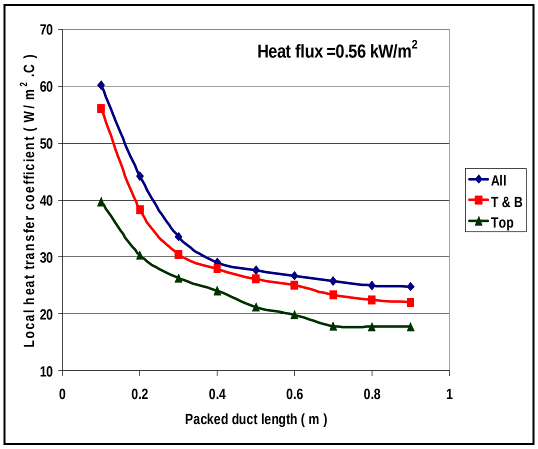
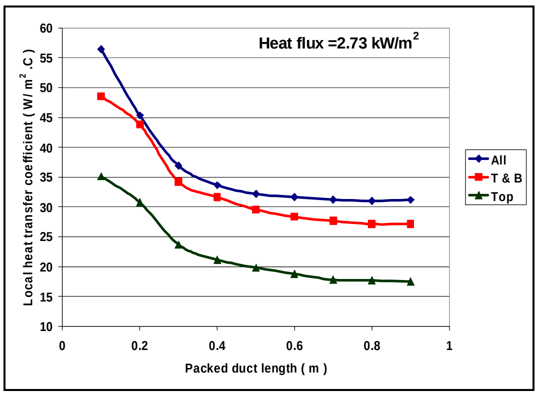
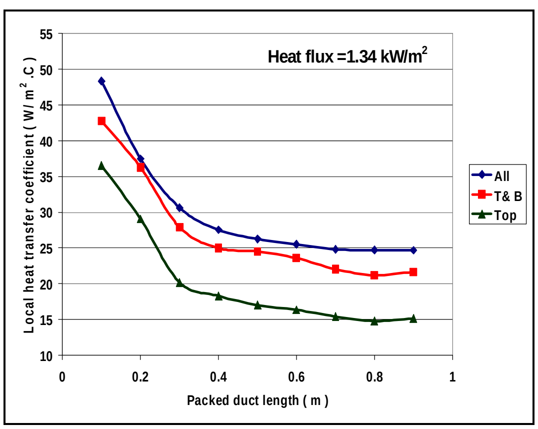


Figure (13) Local heat transfer coefficient with duct length for different heat flux (Re=54797)





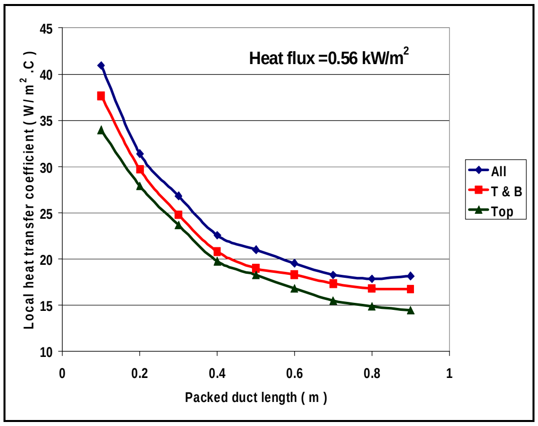
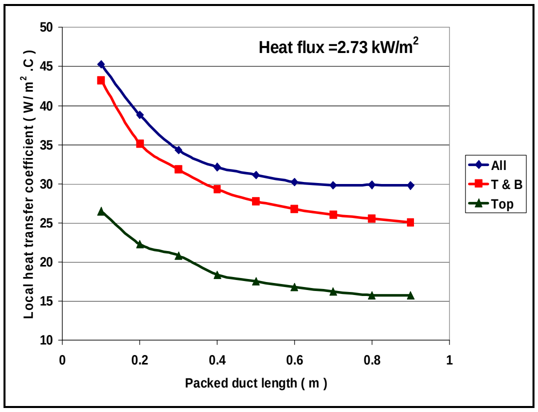
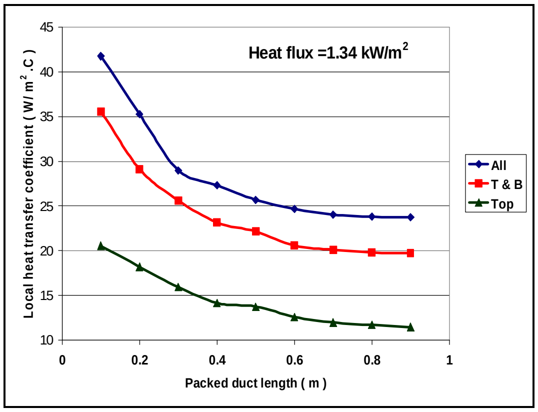


Figure (14) Local heat transfer coefficient with duct length for different heat flux (Re=48222)





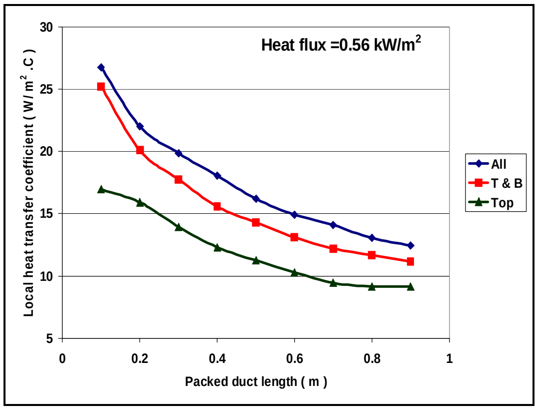
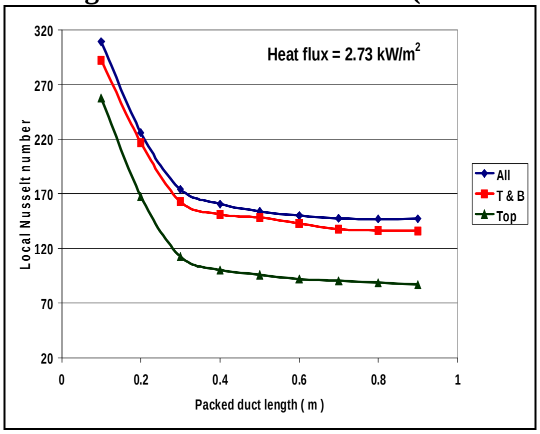
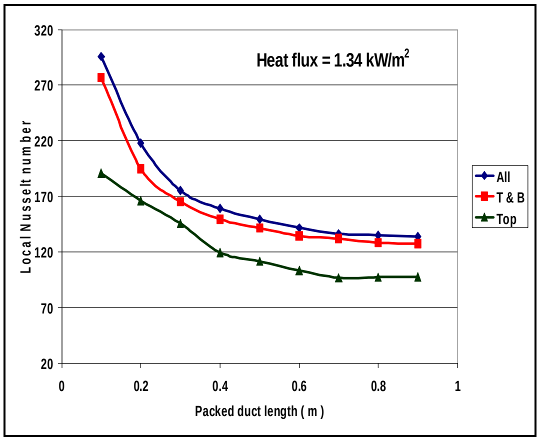


Figure (15) Local heat transfer coefficient with duct length for different heat flux (Re=40339)





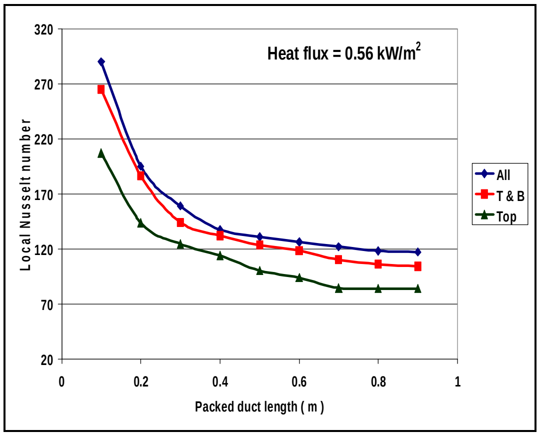
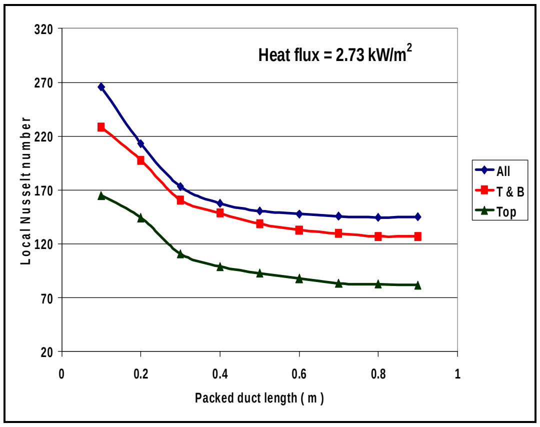
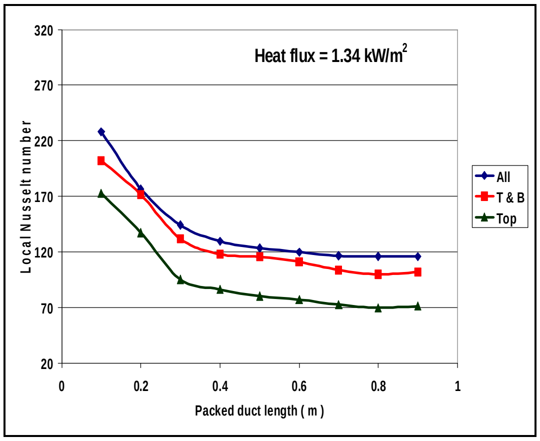


Figure (16) Local Nusselt number with duct length for different heat flux (Re=54797)





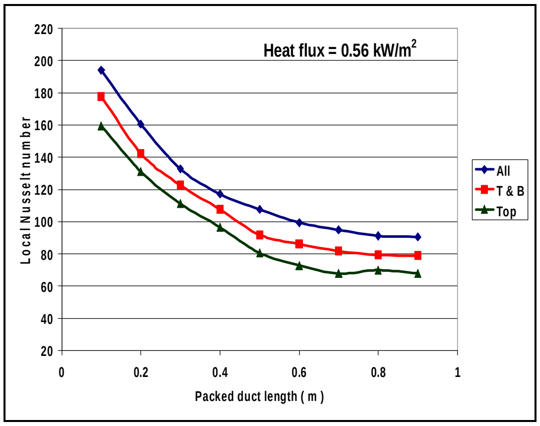
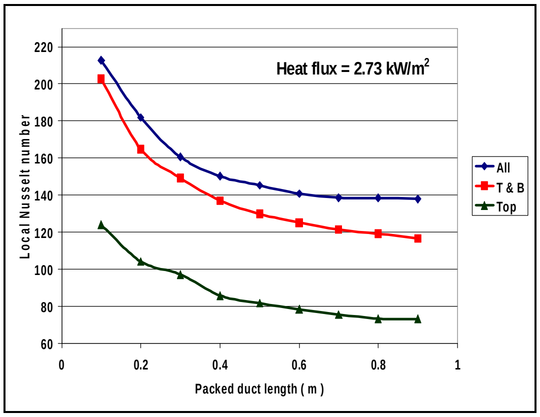
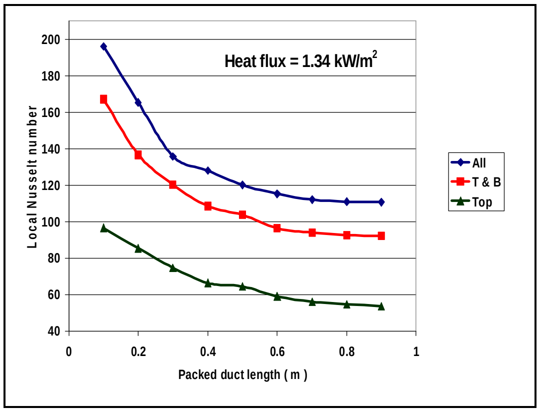


Figure (17) Local Nusselt number with duct length for different heat flux (Re=48222)





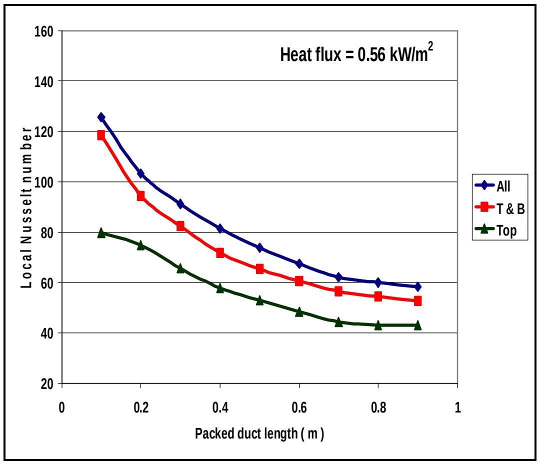


Figure (18) Local Nusselt number with duct length for different heat flux (Re=40339)

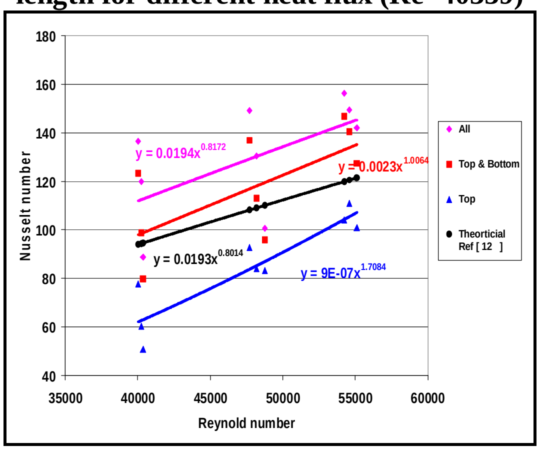


Figure (19) Average Nusselt number with Reynold number in packed &empty duct

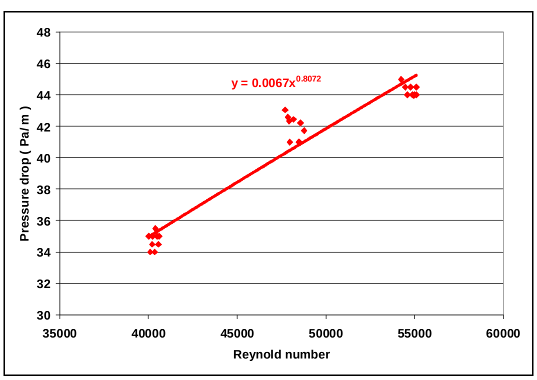


Figure (20) pressure drop vs Reynolds number for packed bed

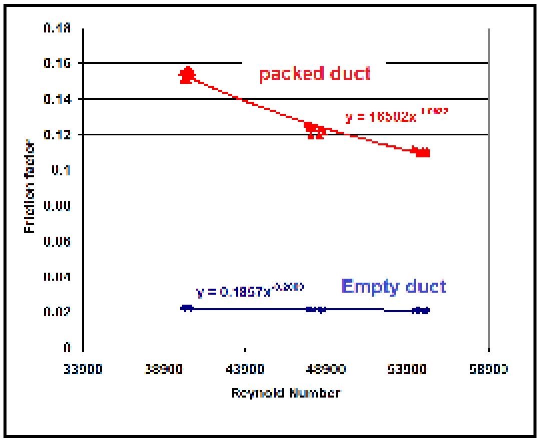


Figure (21) Friction factor vs Reynolds number for packed bed & empty duct