Kufa Journal of Engineering
Vol. 9, No. 2, April 2018, PP. 57-73
Received 19 October 2016, accepted 17 July 2017

CONVECTION HEAT TRANSFER IN A CHANNEL OF DIFFERENT CROSS SECTION FILLED WITH POROUS MEDIA

Ahmed A. Mohammad Saleh[1], Suhad A. Rasheed[2] and Rafel B. Smasem[3]

[1] **Asst. Prof., Mechanical Engineering Department, University of Technology, Baghdad, Iraq, Email: Aamsaleh60@yahoo.com**

[2] **Lecturer, Mechanical Engineering Department, University of Technology, Baghdad, Iraq, Email: Sah_jumaily66@yahoo.com**

[3] **MSc. Student, Mechanical Engineering Department , University of Technology, Baghdad, Iraq, Email: rafel_21_4@yahoo.com**

http://dx.doi.org/10.30572/2018/kje/090205

ABSTRACT

A forced convection heat transfer in ducts (circular, triangular, rectangular) cross sections and (1m) length with hydraulic diameter (0.1m) filled with porous media (glass spheres 12 mm diameter) is investigated experimentally at constant heat flux from the wall (1070W/m²) with Reynolds number range of (12461-2500). Comparison was made between three ducts for local temperature distribution and local Nusselt number). The experimental results showed the effect of Reynolds number and cross section on the temperature profile and local Nusselt number,also empirical correlations for average Nusselt number and Peclet number were obtained for three ducts.

KEYWORDS: Force convection, Heat transfer, Porous media, Different cross section duct, Constant heat flux.

1. INTRODUCTION

Various techniques have been proposed as the use of fins and baffles. To enhance heat transfer many engineering applications Another way for improving the heat transfer characteristics in industrial processes is the use of porous media.A porous medium is a material consisting of a solid matrix with an interconnected voids (pores) or non-interconnected. A number of recent studies incorporated porous-media forced convection in duct of various shapes analytically include those Poulikakos and Renken, (1987), Amir and Osama, (2006), Antonio et al., (2013), Yasin et al., (2008) and Falah, (2011). An experimental investigation was conducted by Makkawi, (1995), the forced convection of air in a rectangular duct with large rashig rings hard plastic packing. The experiments found that the value of Nu increases considerably in the packed channel as Re increases and Nu was about 6 times higher in the packed channel than in the empty one. Experiment had been carried out for forced convection of air in square packed duct by Kifah et al., (2012), they studies heat transfer and pressure drop. It was found that Nusselt number increases as Reynold number, heat flux and number of duct surface exposed to heat flux increase. Nusselt number in packed duct is to be (1.2, 1.19) times higher than the empty ducts at heating all surfaces and top & bottom surface of packed duct, respectively. Many empirical relation between Reynold number, Nusselt number and pressure were obtained. Nobuhiro, (1978), presented an experimental investigation of convective heat transfer in a confined rectangular cavity packed with porous media, on the opposing vertical walls of which different temperatures are imposed. Measurements are made for each of two kinds of solid particles using three kinds of fluids, i.e. water, transformer oil and ethyl alcohol. Irfan and Nevin, (2009), investigated experimentally the heat transfer characteristics of mixed convection flow through a horizontal rectangular channel where open-cell metal foams of different pore densities (10, 20 and 30 PPI) were situated. A uniform heat flux was applied at all of the bounding walls of the channel. For each of three values of the uniform heat flux, temperatures were measured on the entire surfaces of the walls. Results for the average and local Nusselt numbers are presented as functions of the Reynolds and Richardson numbers.

Three different aspect ratios (AR) as 0.25, 0.5 and 1 are tested. Based on the experimental data, new empirical correlations have been constructed to link the Nusselt number. The results of all cases were compared to that of the empty channel and the literature.

Omar et al., (2009), conducted an experimental and theoretical study of natural convection heat transfer between two concentric cylinders filled with a porous medium, when of applying uniform heat flux on the inner cylinder and constant outer surface temperature for the outer cylinder. Two types of filling material were used as porous medium, iron and glass beads. This study shows that the heat dissipated ability in the inner cylinder is a function of Rayligh number, and the Nusselt number increase proportionally with the increases in Rayligh number.The experimental results revealed that the nature of the heat distribution depends greatly on Rayligh number, and the use of highly conductive materials as porous medium will completely cancel the convection as a mean of heat transfer. For this purpose, experimental work had been performed to investigate and explore the effect of change cross section shape (circular, rectangular and triangular) has (1m) length with hydraulic diameter (0.1m) filled with porous media (glass spheres 12 mm diameter) of the duct and the effect of Reynolds number range of (12461-2500) and constant heat flux from the wall (1070W/m²) on the force heat transfer.

2. EXPERIMENTAL APPARATUS

An experimental apparatus, which has been designed and employed in this work is shown in Fig. 1 and Fig. 2. Most of these parts were manufactured and carefully prevented any air leakage between the connected sections during operation and fixing. The three ducts (circular, rectangular and triangular) consists of a galvanized iron duct , 0.6 mm thickness with the same hydraulic diameter 10 cm (dh =10 cm) and 1 m long. A small adapter duct was used to connect the duct to air supply. The air blower which is driven by a three phase motor, Fig. 2, (1.1 KW), (2870 rpm), (50)HZ and discharge pipe size is (51 mm). Five levels of flow rate can be obtained by using inlet gate shatter which use to control the inlet blower angle (0°, 40°, 90°, 130°, 180°). The blower discharge is connected by a (102 mm) solid rubber tube of (2m) length before passing through the test section, Fig. 2. The two ends of each duct was connected with two flanges were manufactured and covered by a stainless steel mesh to hold the porous packing. Rubber (3 mm thickness) was used to prevent any air leakage from the flange. Each duct was drilled to insert pitot tube inside it to measure the air velocity at the exist. The temperature of the duct surface was measured by eighteen sensors distributed within six sections along the surface of the duct, as shown in Fig. 3. After that, the duct surface was cleaned carefully by fine grinding paper. All the sensors wires and heater terminals were taken out the test section through the insulation material. Another two sensors were fixed at the inlet and outlet of the rig to measure the temperature of the air. All the sensors were connected to the Lab jack (data equation). Then, the whole test section was well insulated outside with a glass wool layer of (70 mm) thickness, Figs. 4 and 5. The duct was fill with the spheres randomly and then shaken enough to be certain that there are no spaces larger than the diameter of glass spheres, especially near the wall. This process was continued even to fill up all the length of the test rig (1m) by porous media. To produce constant heat flux along the test section, the surface of the duct was heated electrically using an electrical heater supplied with AC-current from a voltage regulator, the properties of heater for each duct are given in Table 1. All heaters were insulated by titanium and warped around the surface of the porous duct.

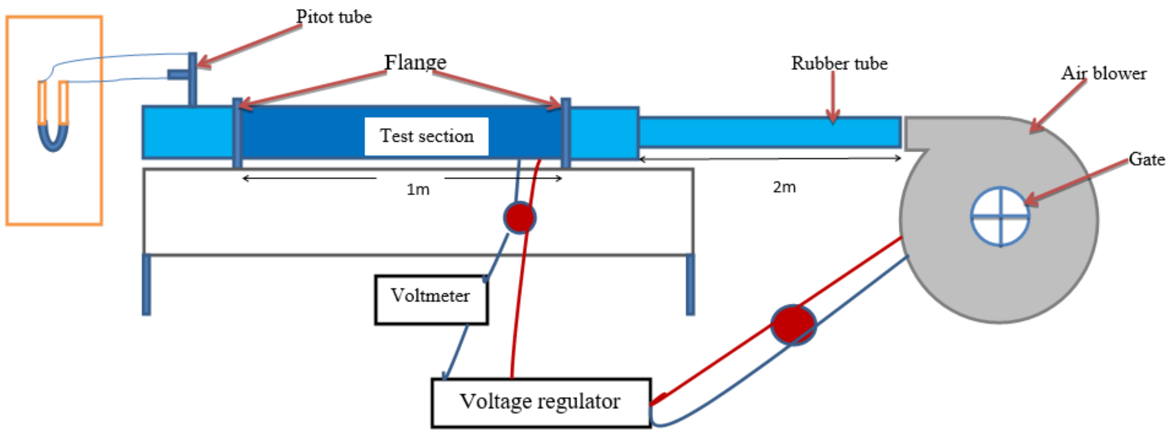


Fig. 1. Schematic diagram of the experimental apparatus.

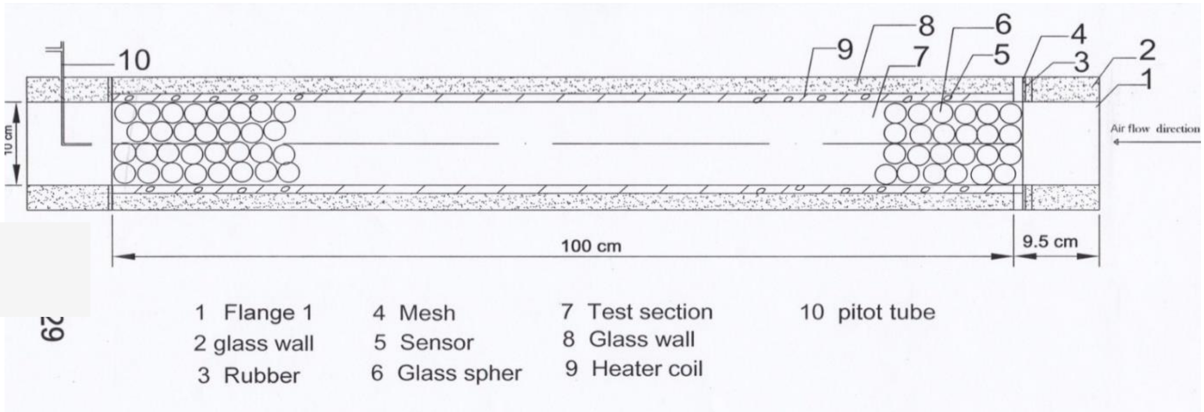


Fig. 2. Details of the cylindrical duct.

Table 1. Properties of heater.

Cross section	Material	Number of heater	Resistance (ohm)	Diameter of coil (mm)	Number of coils
Circular	Stainless steel	4	27.027	8	23
Rectangular	Nickel chrome	2	20.8	0.25	52
Triangular	Nickel chrome	2	18.2	0.25	54

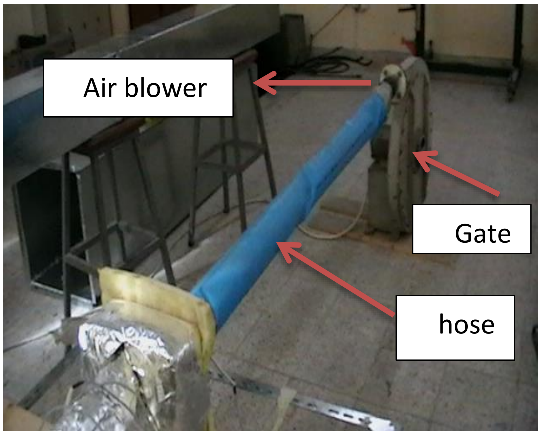


Fig. 3. Air blower and solid rubber hose.

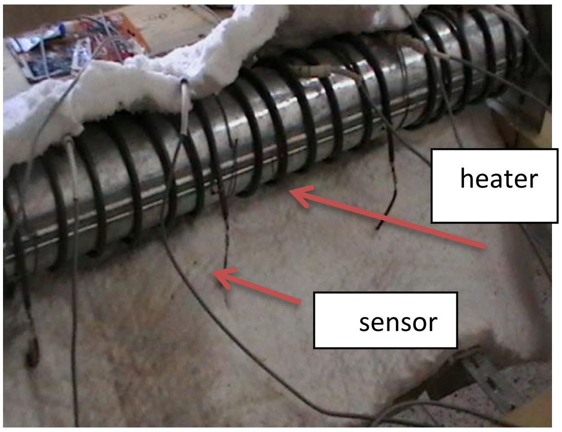


Fig. 4. The distribution of sensor and heater for the circular duct.

Fig. 5. Glass wool layer which covered the whole circular duct.

2. POROUS MEDIA

In the present work, the type of porous media used is glass sphere of (12 mm) diameter. The test section was filled with these glass spheres. The characteristics of the porous medium were:

2.1. Porosity

Porosity (Ɛ) of porous media, depends on the diameter of glass sphere and the diameter of the test section. It is calculate as Mohanned, (2000):

$$\varepsilon = \frac{V_d - V_P}{V_d} \qquad 1$$

Where:

Vd: The volume of duct[$Vd = A \times Length$](m^3), V_P: The volume of glass sphere[$V_P = 4/3 \pi r^3 n$] (m3), A: Area of (test section), m2, r: Radius of glass sphere. n: Number of glass spheres.

[for circular duct n=4750, for rectangular duct n=4900 and for triangular duct n=5300], so porosity for circular duct, rectangular duct and triangular duct is [0.448, 0.6 and 0.616] respectively.

2.2. Density and Thermal conductivity

The average density of the glass spheres was found to be (2562.78 kg/m^3). By using the average density of glass spheres found in the previous item with the density of various kinds of glass given in Kifah, (2012), the thermal conductivity was found to be (0.87 W/m2.C).

3. EXPERIMENTAL PROCEDURE

To carry out the experiments, the following procedure was followed after filling the test section with the porous media: Operating of the blower, controlling the amount of air passing through test section by gates (40°, 90°, 130°, 180°) and place it at the gate of the blower. The electrical heater was switched on, and the heater input power was then adjusted to give the required heat flux. Two hours are required to establish a steady- state condition. The surface temperature was measured by eighteen sensors were measured temperature every forty-five minutes and recorded by lab jack which is connected to the computer until the readings became constant. Repeat procedure after change the inlet blower angles (40°, 90°, 130°, 180°) for the same input power adjusted to give the required heat flux.

4. DATA PROCESSING

1.Heaters current and voltage.

2.Velocity of air passing through the duct.

3.Readings the temperature by a computer special software .

To analyze the heat transfer process from ducts filled with porous media, simplified steps were used as follows, depending on the experimental reading:

4.1. Bulk air temperature:

The bulk air temperature can be calculated by using the following relation Thamir, (2008);

$$\text{Tb} = \left(\frac{Ti+To}{2}\right) + 273.15 \tag{2}$$

4.2. Properties of air

The properties of air which are verbally depending on the bulk temperature, can be calculated by using the following relation Thamir, (2008);

ρf=4.93-0.026Tb+6.4*10-5Tb2–7.5*10-8Tb3+3.36*10-11Tb4 3

μf = 1.3*10-6+5*10-8Tb+1.2*10-12Tb2-3.3*10-13Tb3+12.6*10-16 Tb4 4

Kf=1.43*10-16+9.3*10-5Tb+3.4*10-9Tb2-9.8*10-11Tb3+8.4*10-14Tb4 5

Prf=0.79-1.1*10-4Tb–1.35*10-6Tb2+3.37*10-9Tb3+2.2*10-12Tb4 6

Cpf=1.09-.001Tb+3.78*10-6Tb2-6.27*10-9Tb3+4.14*10-12Tb4 7

4.3. The mean-velocity

At the test section, the local mean-velocity was calculated from the measured dynamic pressure, which is the difference between the total pressure and the static pressure:

$$u = \sqrt{2 \times \frac{p}{\rho_a}} \quad 8$$

4.4. Mass flow rate:

Mass flow rate (ṁ) for the air can be calculated by:

$$\dot{m}=\rho_a * A_c * u \quad 9$$

where:

(A_c) : is the cross- sectional area for the duct (m^2).

4.5. Convective heat transfer :

The convective heat transfer from the air can be calculated as Thamir, (2008):

$$Q= \dot{m}*C_{pf}* (T_o-T_i) \quad 10$$

$$q=Q/As \quad 11$$

where:

(A_S): is duct surface area (m^2).

4.6. The local heat transfer coefficient :

The local heat transfer coefficient can be calculated by using the following relation Mohanned, (2000):

$$h_z = \frac{q}{\Delta tz} \quad 12$$

where

$$\Delta tz = (Ts)z - (Tb)z \qquad 13$$

$$(Tb)z = Ti + \frac{Z}{L}(To - Ti) \qquad 14$$

(Ts)z = average temperature for three points at the cross section of the duct

4.7. Local Nusselt number

The local Nusselt number (Nu_z) was calculated by using the following relation Mohanned, (2000);

$$Nu_Z = \frac{h_Z D}{K_{eff}} \qquad 15$$

where;

$K_{eff} = \varepsilon K_f + (1 - \varepsilon)K_s$ Mohanned, (2000)

K_{eff} = Effective thermal conduction.

K_s= Thermal conductivity of the porous (glass spheres).

k_f = Thermal conductivity of fluid (air).

4.8. Total input power

The total input power supplied to the duct can be calculated as;

$$P_{in} = I*Vo \qquad 16$$

Where:

I: current (A).

Vo: voltage (volt).

4.9. The heat losses

The heat losses from heated duct can be calculated by using the following relations Thamir, (2008);

$$Q_{losses} = 1 - \frac{Q}{P_{in}}. \qquad 17$$

4.10. Reynolds number (Re)

The Reynolds number can be calculated by using the following relation;

$$Re = \frac{\rho_a u_a D_H}{\mu_a} \qquad 18$$

ρ_a : density of the air.

u_a : velocity of the air.

μ_a: dynamic viscosity of air.

D_H = Hydraulic Diameter [$D_h=4(\frac{Aera}{perimeter})$]

4.11. Reynolds number based on particle diameter (Re_p)

The Reynolds number which based on particle diameter was calculated by using the following relation Mohanned, (2000):

$$Re_p = \frac{\rho_a\, u_a\, dp}{\mu_a} \qquad 19$$

4.12. Modified Reynolds number

Modified Reynolds number based on particle diameter was calculated by using the following relation Mohanned, (2000):

$$Re^* = Re_p/(1-\varepsilon) \qquad 20$$

4.13. Average Nusselt number

The average Nusselt number was calculated by using the following relation Mohanned, (2000):

$$Nu_{ave} = \frac{1}{L}\int_{x=0}^{x=l} Nux\, dx \qquad 21$$

By using the Trapezoidal rule to calculate this integration

$$\int_0^1 Nuldl = \frac{\Delta L}{2}\left[Nu^1 + 2Nu^2 + 2Nu^3 + \cdots Nun\right] \qquad 22$$

Where :ΔL *is the distance between two section.*

5. RESULTS AND DISCUSSION

5.1. Temperature Distribution

The direct reading of temperature distribution with axial distance for constant heat flux value of (q"=1070 W/m2) and different Reynolds number values is plotted in Figs. 7 to 10, where Y axis represents the difference between the local wall temperature (Tz) and the inlet air temperature (Ti) to normalize the different inlet temperature taken in different time, and X axis represents the local position on the duct. It is seen that.

The local wall temperature difference (Tz-Ti) gradually increases along the duct and decreasing the Reynolds number at constant heat flux, as shown in Figs. 7 to 9. Increasing Reynolds number yields faster flow through the porous media over heated wall, and therefore thinner thermal boundary layers that require a longer time to develop don't allow heat to transfer into the porous media Figs.10 (a,b,c,d,e) present that the local wall temperature for the triangular duct is more than that for the rectangular duct which is more than circular duct because in the triangular duct, there is a secondary flow formed which made mix convection so the local wall temperature increase Kifah et al., (2012).

In rectangular duct, there is also secondary flow formed which made mix convection, but less than triangular duct, because each angle in triangular duct equals 60°, and angles in rectangular duct equal 90° and the diameter of the glass sphere equals (12 mm). So, the pore space in triangular duct is more than the pore space in rectangular duct, which means the buoyancy effect in triangular duct is more than rectangular duct.

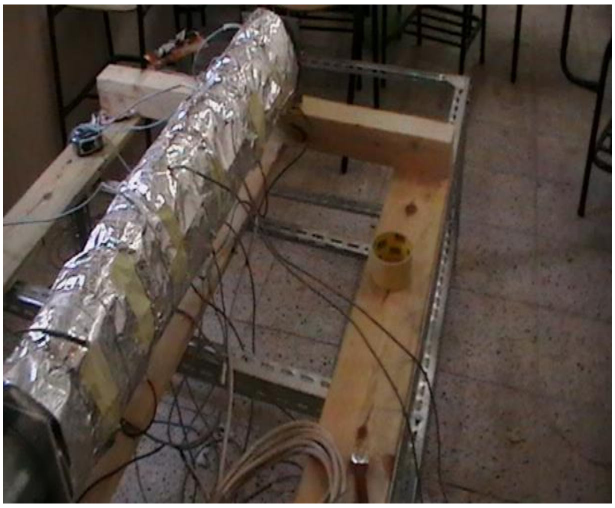


Fig. 6. Circular duct.

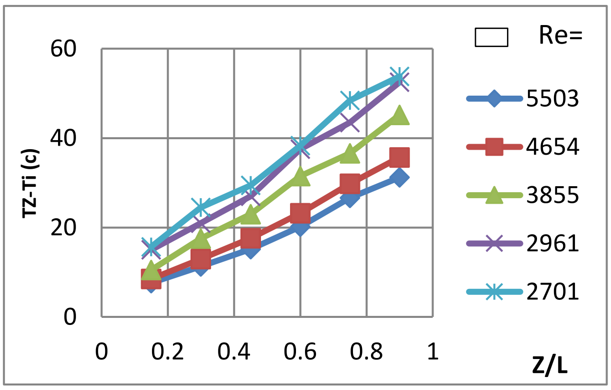


Fig. 7. Temperature differences between local wall temperature and inlet air temperature with Axial Position for rectangular duct.

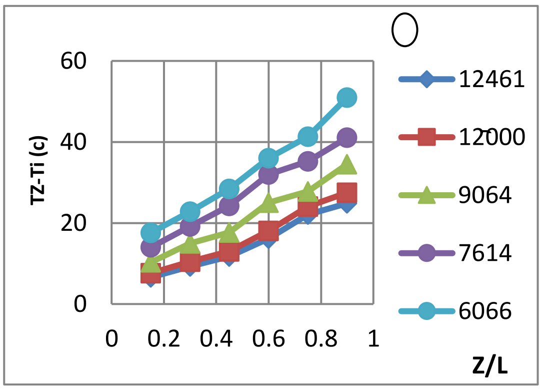


Fig. 8. Temperature differences between local wall temperature and inlet air temperature with Axial Position local for circular duct.

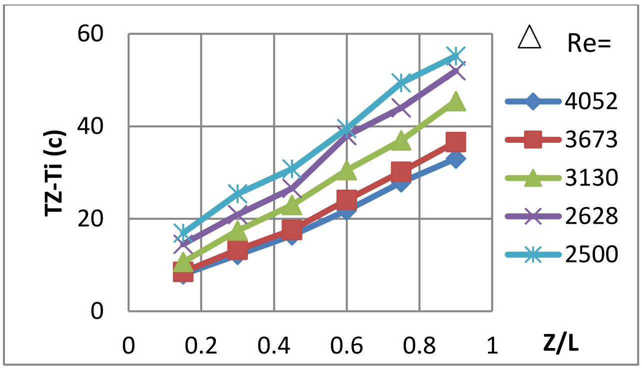
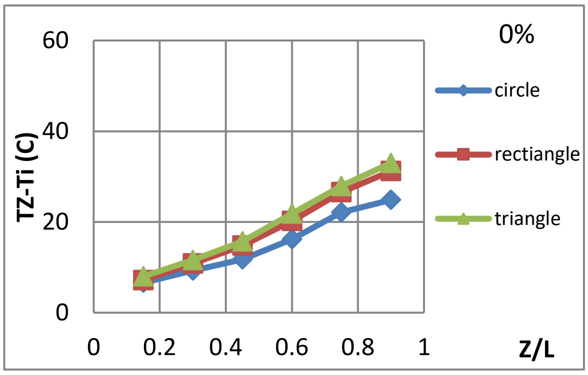


Fig. 9. Temperature differences between local wall temperature and inlet air temperature with Axial Position at (Re =4051.9-2499.7) for triangular duct.



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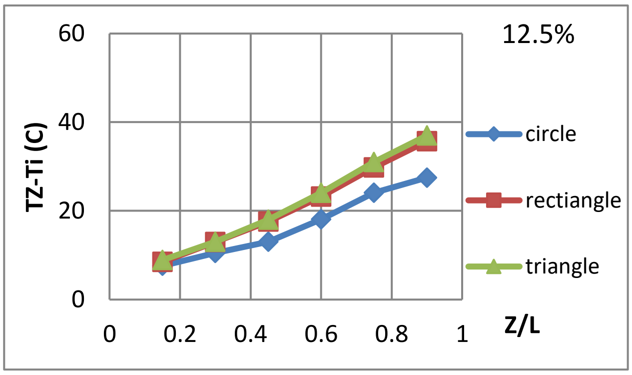

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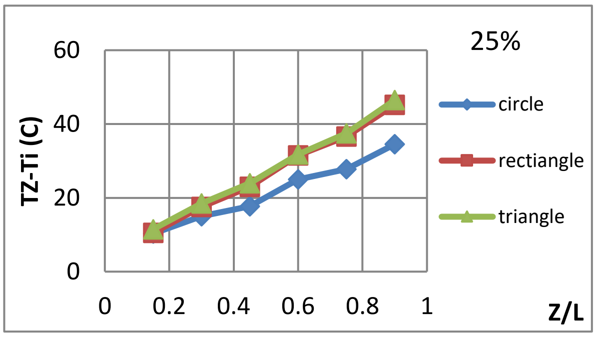

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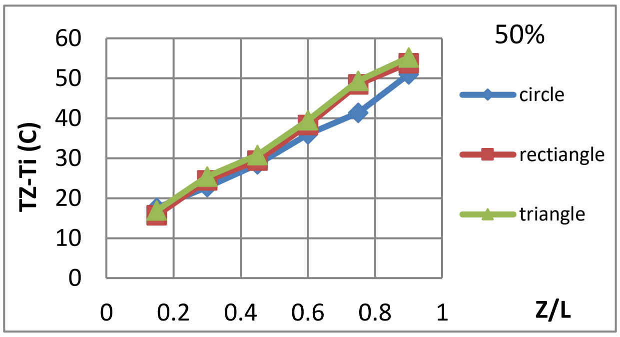

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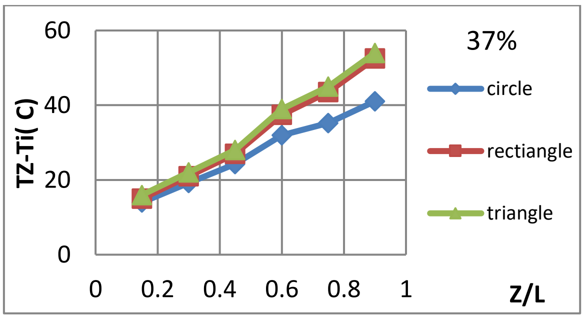


e

Fig. 10 (a, b, c, d and e) Temperature differences between local wall temperature and inlet air temperature with Axial Position for (circular, rectangular, triangular) cross section at the gate closed 0%, 12.5%, 25%, 37% and 50%.

5.2. Local Heat Transfer Coefficient

Figs. 11-13, report the decrease of local heat transfer coefficient which decreases with the axial position along the flow direction because (hz=$\frac{q}{(Ts-Tb)z}$), and increases with the increase in Reynolds number, since the difference between the bulk air and surface temperature (Ts-Tb)z decreases and the thickness of thermal boundary layer also decreases with increasing in Re.

The local heat transfer coefficient in circular duct is much more that for rectangular duct which is more than in triangular duct because of the difference between the bulk air and surface temperature (Ts-Tb)z in triangular duct is much more than in rectangular duct which is more than in circular duct, since the porosity in the circular duct is less than the porosity in the rectangular duct and triangular duct.

5.3. Local Nusselt Number

The local Nusselt number depends on the temperature profile along the duct (circular, rectangular, triangular) was plotted in Figs. 14 – 16.

As seen from these figures, the local Nusselt number begins with high values at the inlet of duct and then decreases along the flow direction and decreases with the Reynolds number, because the difference between the bulk air and surface temperature (Ts-Tb)z increase with the axial position along the duct. Figs.17 (a,b,c,d,e) represent the local Nusselt number distribution at any point on the surface of the duct and the comparison of local Nusselt number with different Reynolds numbers and different cross sections at constant heat flux.

It also can be seen from these figures at the same Reynold numbers, the maximum value for local Nusselt number is found in triangular duct because the porosity in the triangular duct is more that porosity in the rectangular duct and circular duct. Therefore, the effective thermal conductivity (k_{eff}) for triangular duct is less than k_{eff} for rectangular duct and circular duct.

5.4. Average Nusselt Number

The log average Nusselt number has been plotted with the log Peclet number. Figs. 18–20 show that the average Nusselt number increases with increasing Peclet number at constant heat flux. The high value of average Nusselt number due to the low difference between the air and heated ducts temperature and thin thermal boundary layer is formed.

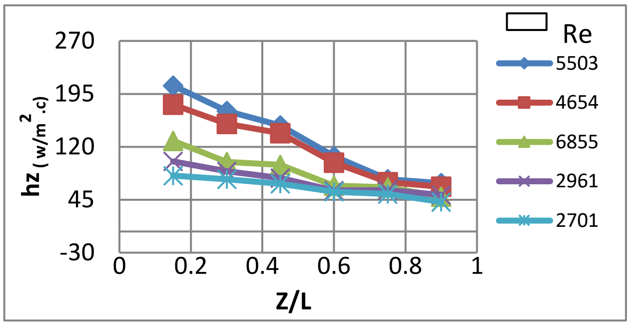


Fig. 11. Local heat transfer coefficient with Axial Position at (Re =5503.4-2701.39) for rectangular duct.

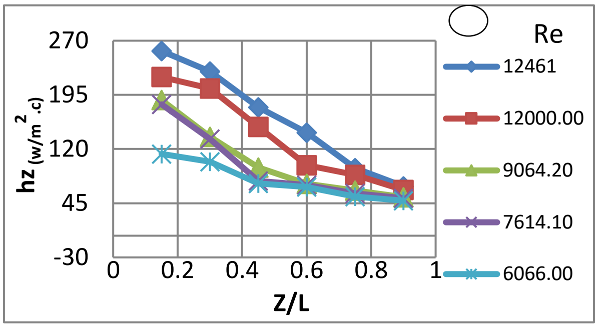


Fig. 12. Local heat transfer coefficient with Axial Position at (Re =12461-6066) for circular duct.

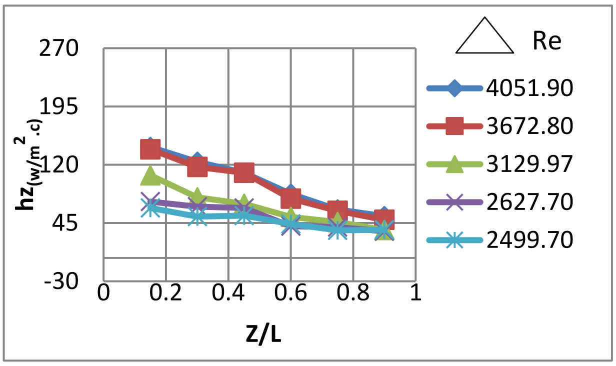


Fig. 13. Local heat transfer coefficient with Axial Position at (Re =4051.9-2499.7) for triangular duct.

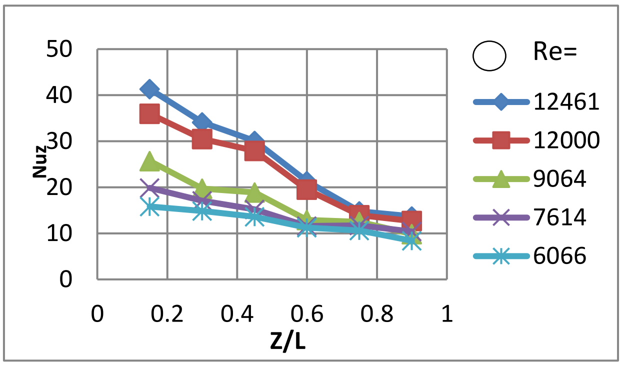


Fig. 14. The local Nusselt number with Axial Position at (Re =12461-6066) for circular duct.

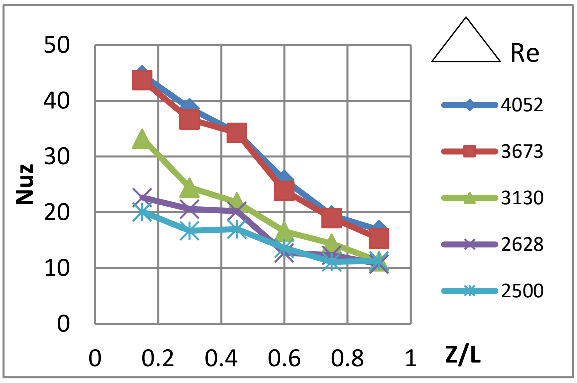


Fig. 15. The local Nusselt number with Axial Position for triangular duct.

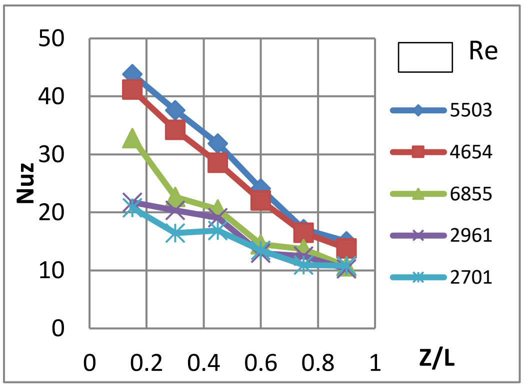
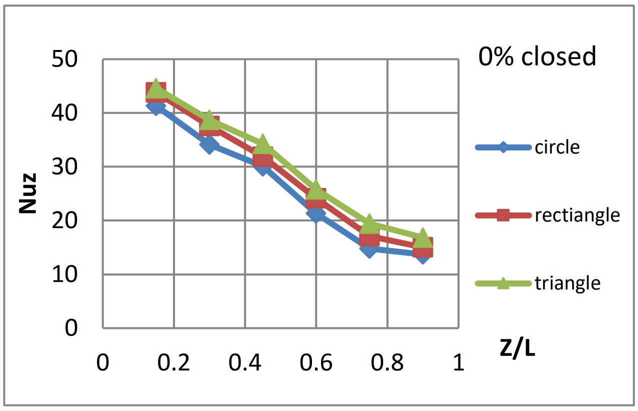
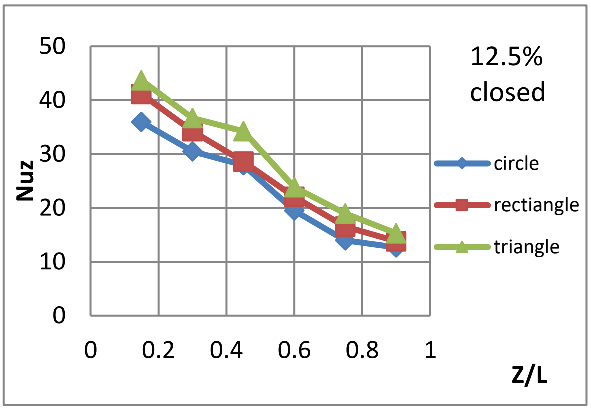


Fig. 16. The local Nusselt number with Axial Position for rectangular duct.



a



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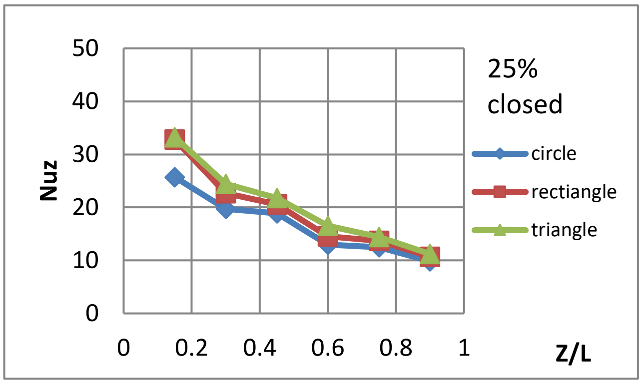

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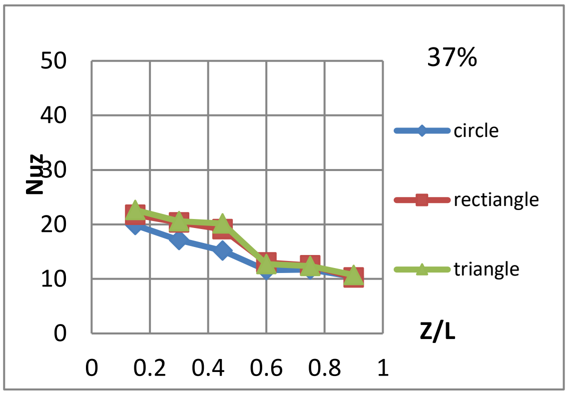

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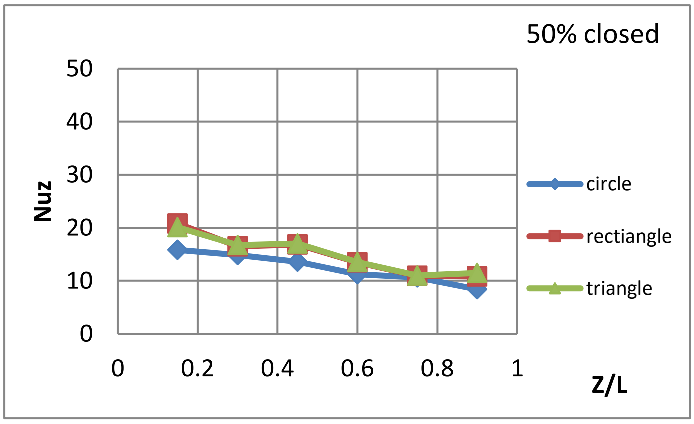

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Fig. 17 (a, b, c, d and e) Local Nusselt Number with Axial Position for (circular, rectangular , triangular) cross section at the gate closed (0%,12.5%,25%,37% and 50%).

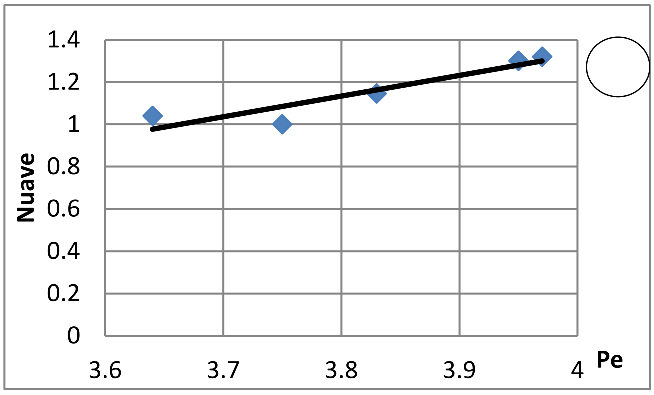

Fig. 18. Average Nusselt Number with Pe for circular duct.

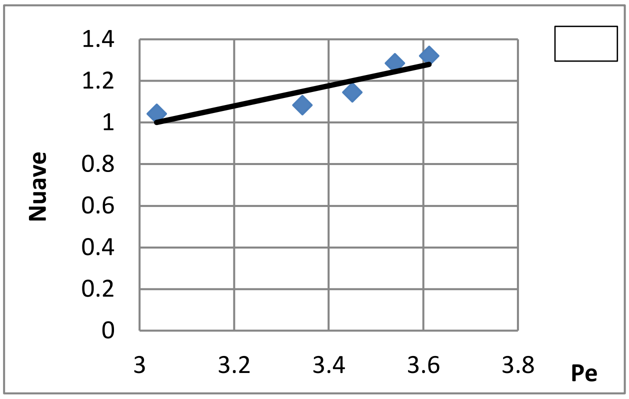

Fig. 19. Average Nusselt Number with Pe for rectangular duct.

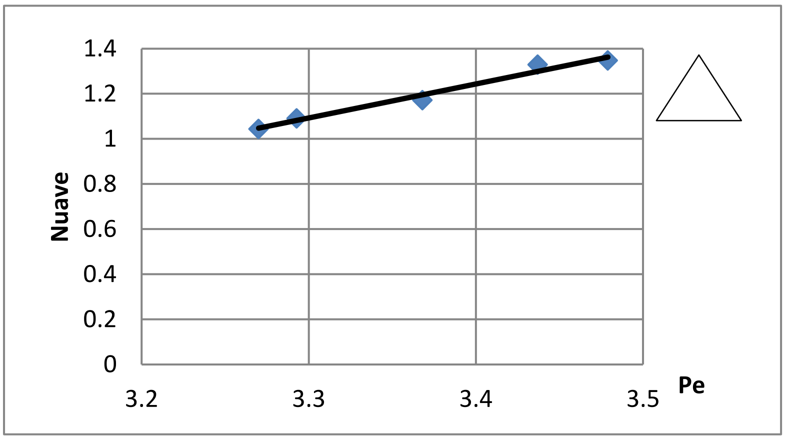

Fig. 20. Average Nusselt Number with Pe for triangular duct.

6. CONCLUSIONS

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

The local surface temperature in the axial direction decreases as Reynolds number increases at constant heat flux. For the constant heat flux, the average Nusselt number increases as the Reynolds number increases. The local surface temperature for the triangular duct is more than in rectangular duct which is more than in cylinder duct.

At the same length and hydraulic diameter,the porosity in the triangular duct is more than porosity in the rectangular duct which is more than in cylinder duct.

The local Nusselt number decreases with increasing the length of the duct. The local Nusselt number for the triangular duct is more than local Nusselt number in the rectangular duct which is more than in cylinder duct. The empirical relations for the three ducts are:

1. For circular cross section duct :

   $Nu = 0.0026\ Pe^{.978}$ for (ε=0.448)

2. For rectangular cross section duct :

   $Nu = 0.339\ Pe^{0.484}$ for (ε=0.6)

3. For triangular cross section duct :

   $Nu = 0.0001\ Pe^{1.5}$ for (ε=0.616)

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