Experimental Investigation of Convection Heat Transfer Enhancement in Horizontal Channel Provided with Metal Foam Blocks

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ABSTRACT

Convection heat transfer in a horizontal channel provided with metal foam blocks of two numbers of pores per unit of length (10 and 40 PPI) and partially heated at a constant heat flux is experimentally investigated with air as the working fluid. A series of experiments have been carried out under steady state condition. The experimental investigations cover the Reynolds number range from 638 to 2168, heat fluxes varied from 453 to 4462 W/m^2, and Darcy number 1.77x10^{-5}, 3.95x10^{-6}. The measured data were collected and analyzed. Results show that the wall temperatures at each heated section are affected by the imposed heat flux variation, Darcy number, and Reynolds number variation. The variations of the local heat transfer coefficient and the mean Nusselt number are presented and analyzed. The mean Nusselt number enhancement was found to be more than 80% for all the studied cases.

Key words: convection heat transfer, metal foam, constant heat flux.

دراسة عملية لتحسين انتقال الحرارة بالحمل في قناة أفقية مزودة بكتل من رغوة معدنية

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الخلاصة

أنتقال الحرارة بالحمل في قناة أفقية مزودة بكتل ذات رغوة معدنية لعددين من المسامات لكل وحدة طول (10 و 40 ثقب لكل بوصة) وبتسخين جزئي بثبوت الفيض الحراري قد تم دراسته عملياً مع أستخدام الهواء كمائع مشغل. تم أجراء عدد من التجارب العملية وعند الوصول لحالة الاستقرار تم جمع البيانات وتحليلها. شملت الدراسة العملية لمدى رقم رينولدز من 638 الى 2168 وفيض حراري متغير من 453 واط/م^2 الى 4462 واط/م^2 ورقم دارسي 1.77x10^{-5}،3.95x10^{-6} . أظهرت النتائج تأثر درجات حرارة الجدار في كل جزء مسخن بتغير الفيض الحراري المسلط وتغير رقم رينولدز ودارسي. تغير معامل أنتقال الحرارة الموضعي و متوسط رقم نسلت قد تم أظهارها وتحليلها. وجد أن التحسن في متوسط رقم نسلت أكثر من 80% لجميع الحالات التي تم دراستها.

الكلمات الرئيسية: أنتقال الحرارة بالحمل، رغوه معدنية، فيظ حراري ثابت.

1. INTRODUCTION

Convection heat transfer in a horizontal channel that provided with heated metal foam blocks of different PPI has a considerable technological interest. This is due to the wide range of applications such as electronic cooling, heat exchangers, nuclear power generation, filtration, and separation. A porous medium is considered as an effective enhancement method of heat transfer due to their intense mixing of the flow and their large surface area to volume ratio. Because of the random structures of porous mediums, they are different in their engineering, physical and thermal properties. Metal foams are class of porous materials with unique properties that are used in several structural and heat transfer applications. Metal foams are being produced as open-cell (functional) and closed-cell foams (structural). Open-cell metal foam consists of pores that are open to their neighboring pores and allow the fluid to pass through them. The closed-cell metal foams have a thin layer of metal dividing the individual pores. **Huang and Vafai, 1994** performed a numerical study of forced convection in a channel with four porous blocks. The Brinkman-Forchheimer-extended Darcy model was used to simulate the flow in the porous medium and the Navier-Stokess equation in the fluid region. They showed that a significant heat transfer augmentation can be achieved by adding the porous blocks. **Hadim, 1994** studied numerically the laminar forced convection in a fully or partially filled porous channel contains with discrete heat sources flush-mounted on the bottom wall. In the both cases, partial and fully porous channel, results show that the Nusselt number increased when the Darcy number was decreased. Results also show that the heat transfer in the both cases was almost the same increase (especially at low Darcy number), while the pressure drop was much lower in the partially filled channel. **Rachedi and Chick, 2001 investigated** numerically the electronic cooling enhancement by insertion of foam materials. This technique based on inserting the porous or foam material between the components on a horizontal board. The Darcy -Brinkman-Forchheimer model was used to describe the fluid motion in the porous media. Results indicated that for a high thermal conductivity porous substrate, substantial enhancement is obtained compared to the fluid case even if the permeability is low. In the mixed convection case, inserting the foam between the blocks lead to a remarkable reduction in temperature of 50%.**Chikh et al., 1998** numerically studied force convection heat transfer in a parallel plate channel, with equally spaced porous blocks attached on the partially heated lower plate. The Darcy-Brinkman-Forchheimer equation was used to simulate the flow in the porous regions. Results show that for porous blocks with low permeability, recirculation zones appear between the blocks and prevent the fluid from going through the next blocks. The local Nusselt number was increased with a decreased in the wall temperature up to 90% by insertion of porous blocks. **Guerroudj and Kahalerras, 2010** studied numerically mixed convection in a channel provided with porous blocks of various shapes that subjected to constant heat flux from the lower plate. The considered shapes vary from the rectangular shape (γ= 90º) to the triangular shape (γ= 50.2º). Results indicated that when the mixed convection (Gr/Re^2) increased, the global Nusselt number increased, especially at small values of permeability for triangular shape. At small values of the Reynolds number, Darcy number, thermal conductivity ratio and porous blocks height, the triangular shape lead to high rate of heat transfer. The highest pressure drop was obtained with the rectangular shape due to its volume which the highest in comparison to the others shapes.

Kurtabas and Celik, **2009** investigated experimentally the mixed convective heat transfer in rectangular channel where the channel filled with open-cells aluminium foams with different number of pores per unit of length (PPI) with constant porosity (ε=0.93). The channel was heated from the top and bottom by uniform heat flux. Results indicated that the average Nusselt number increased relative to the pore densities. Results also showed that at high values of the Reynolds number and Grashof number, the local Nusselt number increased rapidly.

To the best of the authors' knowledge, there is no existing experimental study on the convection heat transfer in a horizontal channel that provided with metal foam blocks of different number of pores per unit of length (PPI) and partially heated at constant heat flux. So the present study was proposed to cover this shortage in the understanding of the fluid flow and heat transfer characteristics in metal foam with convection effects.

In the present work, convection heat transfer in a horizontal channel that provided with metal foam (copper foam) blocks of different (PPI) and subjected to a constant heat flux, is experimentally examined with air as the working fluid. The main objective is to study the influence of convection heat transfer on the flow field and the associated heat transfer process in such system. The influence of heat flux, Darcy number, and Reynolds number variation on wall temperatures and Nusselt number are investigated and analysed.

2. EXPERIMENTAL APPARATUS AND PROCEDURE

2.1 Experimental Apparatus

The experimental investigation is carried out in the apparatus shown in **Fig.1**. The apparatus has been constructed in the Heat Transfer Lab, at the Mechanical Engineering Department, University of Baghdad to achieve the requirements of the present study and it consists of the following major assemblies:

1- The Wind Tunnel

A driven type, low speed wind tunnel with solid wood walls was used in the present work. The general arrangement of this tunnel is shown photographically in **Fig.2** .The wind tunnel has the following parts and features;

- A conical inlet section of 500 mm length. The conical section inlet diameter is 70 mm and its outlet is designed as a rectangular section (100 mm x 50 mm).
- A calming (entry) section of 150 mm length.
- The test section is made up from Perspex material (10 mm thickness) from three sides (left, right and top) while the bottom side is made up of solid wood. The left and right sides dimensions are (250 mm x 100mm), and the top and bottom sides dimensions are (250 mm x 50 mm) as sketched in **Fig.3**.
- Exit section of 1500 mm length.
- A blower that driven by a one phase A.C motor with a speed of 2800 RPM.
- The valve by which the flow rate of air can be controlled and bypass.

2- Heating System

Three heaters are used in the present study. Each heater is made from the zigzag coil with a 0.5mm diameter and length 1050 mm made from nickel-chrome wire, putting on a mica sheet of 0.4mm thick to ensure electrical insulation. The zigzag coils as shown in **Fig.3a** are covered from the top and bottom with mica sheets. Bottom side of the each heater was insulated by a Teflon plate (16 mm thickness) and the top side served as the heating surface unit that covered by copper plate (1.5mm thickness) to ensure a uniform distribution of the supplied heat flux. The Teflon was chosen because of its low thermal conductivity in order to reduce the heat loss from the heaters bottom. Then the test section is well insulated from bottom with a glass wool and kaowool layer of 20, 60 mm thickness respectively. The heaters are supplied with the AC-current in series from a two transformer (Variac), to control the incoming current according to the heat flux desired. Thermal grease is used between the heated section and the metal foam blocks to achieve better contact between them.

3- The Thermocouples Network

The temperature distribution of the enclosure test section was measured by twenty four K-type thermocouples. There are fifteen thermocouple used to measure the air bulk temperature distribution inside the test section and distributed within five sections along the test section as sketched in **Fig.4a**. Three thermocouples are positioned on each heater to measure the temperature distribution along the copper plate surface. The mean value of the measured air temperature is used as the air temperature at that section.

The thermocouples were fixed on the copper plate surface by drilling (V) holes of 1 mm in diameter and 0.5 mm deep. Then the thermocouples junctions were fixed by the solder. The excess solder was removed and the copper plate surface was cleaned carefully by fine grinding paper. All the thermocouples wires and heater terminals were taken out the test section through the Perspex material and the insulation material, respectively.

In the case of adding full length rectangular metal foam blocks, another seven K-type thermocouples are added for each block as shown in **Fig.4b**. Four thermocouples are used to measure the metal foam wall temperature distribution by soldering them with metal foam wall. The other three thermocouples are used to measure the air bulk temperature that flow through the metal foam block.

The outer covering shield is removed from each thermocouple in order to reduce the space it takes inside the metal foam block. The thermocouples are fixed on the metal foam surface by the solder.

Finally, the temperature distribution on the lower surface of the insulation shield was measured by employing two K-type thermocouples distributed in an equal pitch, to calculate the heat transfer lost during the experiment, which is found to be approximately 3% during the whole range of the imposed heat flux.

2.2 Measuring Systems

1- Temperature Measuring System

The thermocouples were connected to a data acquisition system, consisting of NI CompactDAQ chassis with three NI 9213 16-channel thermocouple input model, with LabVIEW 2009 software to record the temperature measurements.

2- Pressure Drop Measuring System

The measurement of pressure drop is interesting consideration to study in the metal foam as it related to the characteristics of the metal foam blocks.

3- Pressure Taps Positions

For pressure drop measurements, four pressure taps of inner diameter 3 mm were placed on four locations long the flow direction on the top side with a uniform spacing of 75 mm as sketched in **Fig.4a**. The pressure drop in the test section is measured by micromanometer.

4- Electrical Power Measurement

The constant heat flux is supplied by using electrical circuit of alternating current that includes:

A) Voltage regulator.
B) Transformer.
C) Voltmeter.
D) Clamp meter.

2.3 Experimental Procedure

The experiments are repeated with varying some parameters to study their effect on the temperature distribution, pressure drop and Nusselt number. These parameters are:

1. Porosity (90.302%) for 10 PPI and (89.81%) for 40 PPI.
2. Air flow rate range (3, 7, and 9 m^3/hr.).
3. Heat flux range (453, 1862, 3007, and 4462 W/m^2).

The experiments has been started by operating the blower and adjusting the required air flow rate by regulating valve, then the electrical power switched on and the heaters input voltage is adjusted by the transformer (Variac) to give the required voltage and current to calculate the electrical power in accordance to the heat flux required. The apparatus is left at least for two hours to reach the steady state condition. The thermocouples readings are recorded every half an hour by the data acquisition system until the reading became almost constant (temperature did not vary by more than 0.5˚C in 30 min), a final reading is recorded at the steady state condition. During each test run, the following readings were recorded:

a- Thermocouples reading in ˚C.
b- Micro manometer reading (pressure drop) in Pa with the pressure of the metal foam.
c- The heater voltage in volts.
d- The heater current in amperes.

3. HEAT TRANSFER CALCULATIONS

The net heat flux that supplied at the copper plate is determined from recording the electrical power that supplied to the heater and applying the following equation;

$$P_o = I \times V_h \tag{1}$$

$$q = \frac{P_o}{3 \times A_h} \tag{2}$$

where P_o is the electrical power that consumed by the heaters, I is the current that flow through the heaters, V_h is the voltage across the heaters, and A_h surface area of each heated section.

The Reynolds Number and Darcy number are prescribed, following as;

$$Re = \frac{u_{in} D_h}{v} \tag{3}$$

$$Da = \frac{K}{D_h^2} \tag{4}$$

The local friction coefficient is given by:

$$f = (-\frac{dp}{dx}) \frac{D_h}{2\rho u^2} \tag{5}$$

The local heat transfer coefficient at the heated wall can be defined as:

$$h = \frac{q}{T_w - T_b} \tag{6}$$

Hence, the local and the mean Nusselt number can be calculated as, **Nield and Bejan, 2006**:

$$\mathrm{Nu} = \frac{\mathrm{h\, D_h}}{\mathrm{k}} = \frac{\mathrm{q\, D_h}}{\mathrm{k\,(T_w - T_b)}} \tag{7}$$

$$Nu_m = \frac{1}{W} \int_{X_i}^{X_i + W} Nu\, dX \tag{8}$$

where Xi is the position of the block i from the channel entrance.

4. POROSITY AND DENSITY OF METAL FOAM

In general, porosity (ε) is the important property of metal foams, which depends on the volume of pores and the total volume of the sample that contains the copper foams. Measuring the volume of the copper foam sample was done after taken three readings for volume of three different samples for the same (PPI) and dimensions, and takes the average of these readings. Then porosity can be found from the equation;

$$\varepsilon = (1 - \frac{\rho_{cal.}}{\rho_{th.}}) \text{ X } 100\ \% \tag{9}$$

where $\rho_{th.}$,theoretical density of pure copper (8933 kg/m^3), **Incropera et al. 2007**.

For the copper foams the average porosity was obtained at (90.302%) for 10 PPI and (89.81%) for 40 PPI. The average density of the copper foams was measured (866.301 kg/m³) for 10 PPI and (909.575 kg/m³) for 40 PPI.

5. PERMEABILITY OF THE METAL FOAM

The permeability can be found from a Darcy- Forchheimer relation:

$$\frac{\Delta P}{L} = \underbrace{\frac{\mu}{K} u}_{\text{Darcy term}} + \underbrace{\frac{\rho\, C}{\sqrt{K}} u^2}_{\text{Forchheimer term}} \quad (10)$$

$$\frac{\Delta P}{L} = A\, u + B\, u^2 \quad (11)$$

By comparing Eq. (10) with Eq. (11), permeability and inertial coefficient for each metal foam block can be written as:

$$K = \frac{\mu}{A} \quad (12)$$

$$C = \frac{B\sqrt{K}}{\rho} \quad (13)$$

The permeability (K) and inertial coefficient (C) of each metal foam block are experimentally determined by measuring the pressure drop across the test section at different air flow rate. To find permeability and inertial coefficient for each metal foam block, first A and B constants in Eq. (11) must be calculate by making a curve fitting (second order polynomial) for each curve in **Fig.5.** A and B values for each curve are listed in **Table 1**. **Table 2** lists the permeability and inertial coefficient for each metal foam block.

6. RESULTS AND DISCUSSION

A series of experiments have been carried out with a heat flux range from q = 453 W/m^2 to 4462 W/m^2 and Re= 638 to 2168. The temperature distribution along the three heaters surfaces is measured and presented. The influence of heat flux, Darcy number, and Reynolds number variation on the local heat transfer coefficient and the mean Nusselt number is discussed and analyzed.

6.1 Wall Temperature Distribution of the Heated Section

Figs. 6 and **7** show the effect of the imposed heat flux variation and Reynolds number variation on the distribution of the wall temperature at each heated section for the case of without metal foam blocks (fluid case) at q=453,1862, and 3007 W/m^2 and Re=638,1594,and 2168. As expected, **Fig.6** shows that the wall temperature is increased at each heated section as the heat flux increased for the same Reynolds number value. When the imposed heat flux is increased (with a constant Reynolds number), the buoyancy effect increases and causes a faster growth in the thermal boundary layer along the surface at each heated section.

Fig.7 shows the influence of Reynolds number variation on the wall temperature distribution at each heated section for q = 1862 W/m^2. It is obvious that the increasing of Reynolds number reduces the wall temperature at each heated section as heat flux is kept constant. When the Reynolds number is increased the thermal boundary layer retreat along the heated wall at each heated section.

Another observation can be made from **Fig.7** that the difference between the temperature distribution values for different Reynolds number at the first heated section is maximum while this difference is minimum at the third heated section. This can be attributed to the fact that the thermal boundary layer thickness over the third heated section is greater than that over the first heated section and as a result the Reynolds number variation has a smaller effect on the temperature variation at the third heated section.

A general trend can be seen from **Figs.8-10** that the wall temperature values at the third heated section are higher than that at the first heated section due to the largest thickness of the thermal boundary layer over the third heated section. **Fig.8** shows the influence of the imposed heat flux variation on the the distribution of the wall temperature at each heated section for Re=1594,and Da=1.77 $x10^{-5}$. It can be seen that wall temperature is increased at each heated section as the heat flux increased for the same Reynolds number value. When the imposed heat flux is increased, the buoyancy effect increases and causes a faster growth in the thermal boundary layer along the surface at each heated section.

Fig.9 shows the effect of Reynolds number variation on the wall temperature distribution at each heated section for q = 1862 W/m^2 ,and Da=3.95 $x10^{-6}$. It is obvious that the increasing of Reynolds number reduces the wall temperature at each heated section for the same heat flux value. When the Reynolds number is increased the thermal boundary layer retreat along the heated wall at each heated section.

Fig.10 shows the effect of insert metal foam blocks on the wall temperature distribution at each heated section for q=1862 W/m^2, Re=1594, and Da=1.77 $x10^{-5}$, 3.95 $x10^{-6}$. It can be seen that the wall temperature in the case of insert metal foam blocks is much lower than that in the case of without metal foam blocks (fluid case). The presence of the metal foam blocks caused part of heat to transfer from the heated section by mean of conduction through the metal foam blocks and the other part by mean of convection to the incoming fluid that passed over the heated section. The conducted heat through the metal foam block is then transferred to the incoming fluid by means of convection due to the high mixing that provided by the metal foam. The above mechanism that works in the case of metal foam presence increased the heat transfer from the heated section to the incoming fluid and increases the bulk temperature of air which means that air will gain more heat from the hot wall which leads to decrease the wall temperature at each heated section. The wall temperature at each heated section reduces more with a Darcy number decrease because of increasing in mixing which leads to increase in convected heat by air.

6.2 Local Heat Transfer Coefficient

A general behavior can be seen from the distribution of the local heat transfer coefficient in **Figs. 11-13** that the local heat transfer coefficient in all cases decreases with increase in the axial distance at each heated section. Since the heat transfer coefficient is based on the temperature difference with respect to the bulk temperature, this trend is expected as the largest temperature difference between the heated wall and the incoming cold fluid occurs at the leading edge of the heated section especially at higher Reynolds number. Therefore, the highest heat transfer rate occurs at the leading edge of the heated section especially at first heated section which is nearest to the inlet.

Fig.11 shows the influence of the imposed heat flux variation on the distribution of the local heat transfer coefficient at each heated section for Re =2168 and Da=1.77 $x10^{-5}$. It can be seen that the local heat transfer coefficient is increased at each heated section as the heat flux increased for the same Reynolds number value. This can be attributed to the fact that for higher heat fluxes the buoyancy effect increases and the thermal boundary layer growth is more rapidly and causes a smaller temperature difference between the fluid bulk temperature and the heated wall temperature at each heated section.

Fig.12 shows the influence of Reynolds number variation on the distribution of the local heat transfer coefficient at each heated section for q = 1862 W/m^2 and Da= 3.95 $x10^{-6}$. It can be seen that the local heat transfer coefficient is increased as the Reynolds number increased for the same heat flux value. When the Reynolds number is increased, a reduction in the thermal boundary layer thickness occurs with the domination of the incoming cold-fluid effect and this will cause a larger fluid mixing and higher local heat transfer coefficient values especially at first heated section which is nearest to the inlet.

Fig.13 shows the effect of Darcy number on the local heat transfer coefficient at each heated section for q= 1862 W/m^2 and Re=1594. It can be seen that the local heat transfer coefficient in the case of insert metal foam blocks is much higher than that in the case of without metal foam blocks (fluid case). **Fig.13** also shows that the local heat transfer coefficient for lower Darcy Number (Da= 3.95 $x10^{-6}$) is higher than that for higher Darcy Number (Da= 1.77 $x10^{-5}$). This can be attributed to the heat transfer enhancement that caused by the higher fluid mixing in the lower Darcy number metal foam block.

6.3 Mean Nusselt Number

A general behavior can be seen from **Figs.14-16** that the maximum mean Nusselt number value is located at the first block. This trend is expected as the largest temperature difference between the heated wall and the incoming cold fluid occurs at the first block. Therefore, the highest heat transfer rate occurs at the first heated section.

The variation of the mean Nusselt number with imposed heat flux along the heated wall at each block is presented in **Fig.14** for Re=1594 and Da= 1.77 $x10^{-5}$. It can be noticed from **Fig.14** that the mean Nusselt number is increased as the heat flux increased at each block number for the same Reynolds number value.

Fig.15 shows the influence of Reynolds number variation on the mean Nusselt Number at each block for q = 1862 W/m^2 and Da= 3.95 $x10^{-6}$. It can be seen that the mean Nusselt number is increased at each block as the Reynolds number increased for the same heat flux value. This can be attributed to the higher fluid mixing that associated with the domination of the incoming cold-fluid effect.

Fig.16 shows the effect of Darcy number on the mean Nusselt number at each block for q= 1862 W/m^2 and Re=1594. It can be seen that the mean Nusselt number in case insert of metal foam blocks enhanced the heat transfer from the heated sections compared with the pure fluid case. **Fig.16** also indicate that the mean Nusselt number for Da= 3.95 x10^{-6} is higher than that of Da= 1.77 x10^{-5} due to the higher fluid mixing in the lower Darcy number. The adding of the metal foam blocks caused a remarkable enhancement in the mean Nusselt number from that in the fluid case as in case of q=1862 W/m^2 and Re=1594 (95% first block, 93% second block, and 93% third block) for Da=3.95 x10^{-6} , (91% first block,89% second block,and 85% third block) for Da=1.77 x10^{-5} .

6.4 Temperature Distribution through the Metal Foam Blocks

Fig.17 shows the influence of Darcy number on the temperature distribution, through the metal foam blocks at each heated section. It can be seen that the temperature distribution is decreased at each heated section as the Darcy number decreased for the same Reynolds number and heat flux. This can be attributed to the fact that for higher surface area of the metal foam especially for lower Darcy number (3.95 x 10^{-6}).

6.5 Local Friction Coefficient

Fig.18 shows the effect of the local friction coefficient on the Reynolds number for different Darcy number. This figure shows that the local friction coefficient is affected by the Darcy number of the metal foam. It shows that the local friction coefficient increases with Darcy number decreasing because of the increase in the resistance to the axial flow. So the local friction coefficient also decreases with increase in Reynolds number.

7. COMPARISON WITH PREVIOUS EXPERIMENTAL RESULTS

Hadim, 1994 conducted a numerical study on the force convective heat transfer in a channel filled partially porous with discrete heat source. This work is the closest previous published work that found in the literature using the same setup with convection heat transfer except that it is a theoretical study and different properties such as Pr=10 and ε=0.97. Additionally, thermal conductivity ratio (R_k= $k_{eff.}$ / k_f) was set equal to 1 which means that the thermal conductivity of the porous medium is equal to the thermal conductivity of the fluid. But in our present work the thermal conductivity of the porous blocks material has great influence of the overall heat transfer and fluid flow characteristics.

It can be seen from **Fig.19** that the average Nusselt number increases when the Reynolds number is increased and when the Darcy number increased. The behavior shown in **Fig. 19** agrees with the present work results that shown in **Fig.15** (the mean Nusselt number increases as Reynolds number is increased for the same Darcy number value at each block) and in **Fig.16** (the mean Nusselt number is increased as Darcy number decreased for the same Reynolds number value at each heated section).

8. CONCLUSIONS

The main conclusions of the present work are:

1- The wall temperature at each heated section increases as the imposed heat flux is increased of two cases without (fluid case) and with metal foam blocks.

2- The wall temperature at each heated section decreases as the Reynolds number (the same heat flux value) is increased of two cases without (fluid case) and with metal foam blocks.
3- The local heat transfer coefficient and the mean Nusselt number increased with the increased of the imposed heat flux and Reynolds number.
4- The local heat transfer coefficient and the mean Nusselt number is increased with the decreased of the Darcy number.
5- The enhancement in the mean Nusselt number for all the studied cases is over 80 % from the fluid case.

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NOMENCLATURE

C= inertia coefficient.
Da= Darcy number.
D_h = hydraulic diameter, m.
h = local heat transfer coefficient, W/m^2.K.
k= thermal conductivity of fluid, W/m .K.
$k_{eff.}$= effective thermal conductivity , W/m .K.
L =thickness of metal foam block in flow direction, m.
p = pressure, pa.
PPI= pores per inch.
q = heat flux, W/m^2.

Re= Reynolds number.
T = temperature, °C.
u_{in}= inlet velocity, m/s.
W= width of the copper foam block, m.
K = permeability, m^2.
Nu = Nusselt number.
Pr = Prandtl Number.
v = kinematic viscosity, m^2/s.
ε = porosity.
μ = dynamic viscosity, kg m/s.
ρ = density of air, $\mathrm{kg/m^3}$.

Subscript Meaning

b = bulk.
m = mean.
w = wall.

Table 1. The values of A, and B constants.

Pores per inch (PPI)	Porosity (ε) (%)	A	B
10	90.302	234.89	728
40	89.81	1052	1444

Table 2. Measured permeability and inertial coefficient of the metal foam.

Pores per inch (PPI)	Porosity (ε) (%)	Permeability † (K) (m^2)	Inertial coefficient (C)	Darcy number†† (Da)
10	90.302	7.859×10^{-8}	0.17	1.77×10^{-5}
40	89.81	1.75×10^{-8}	0.159	3.95×10^{-6}

† Calculated experimentally with the use of Eq. (12).
†† Calculated from Eq. (4).

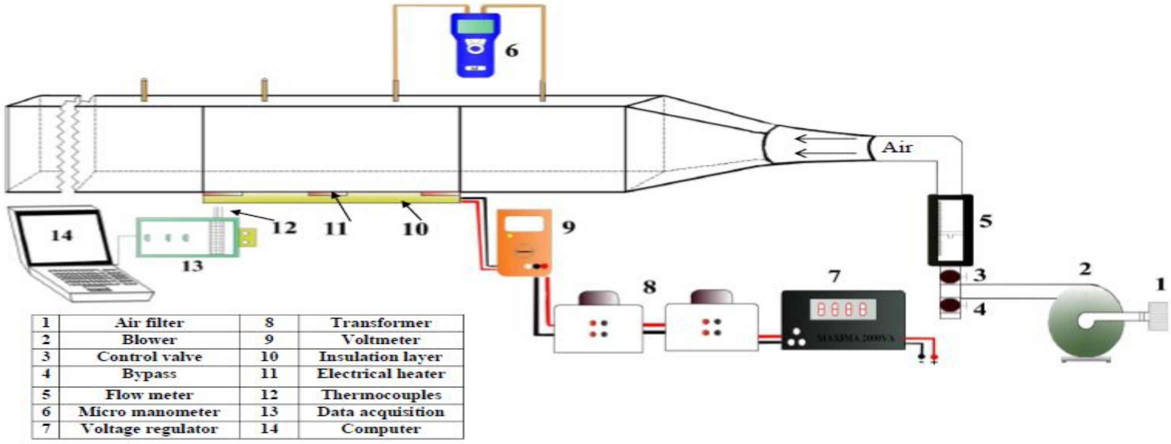
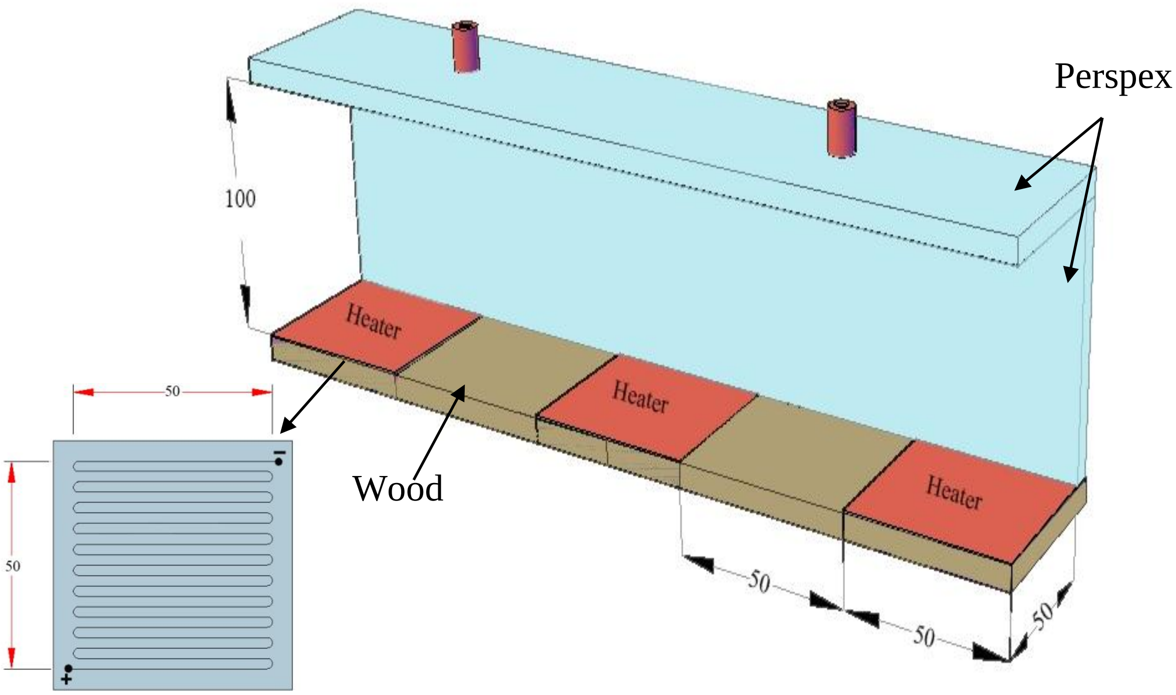
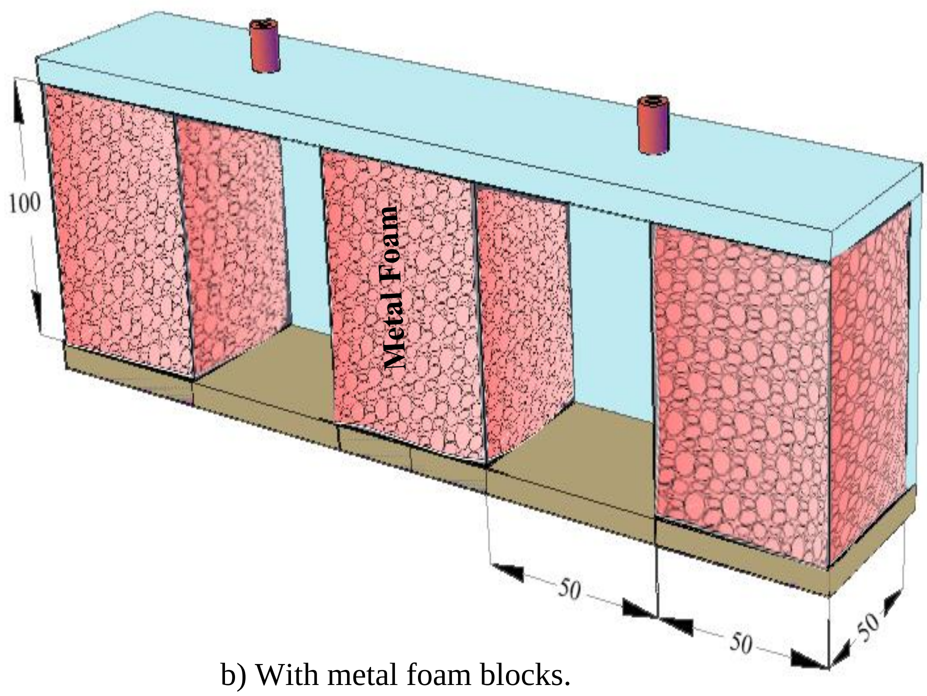


Figure 1. Schematic of the experimental apparatus.

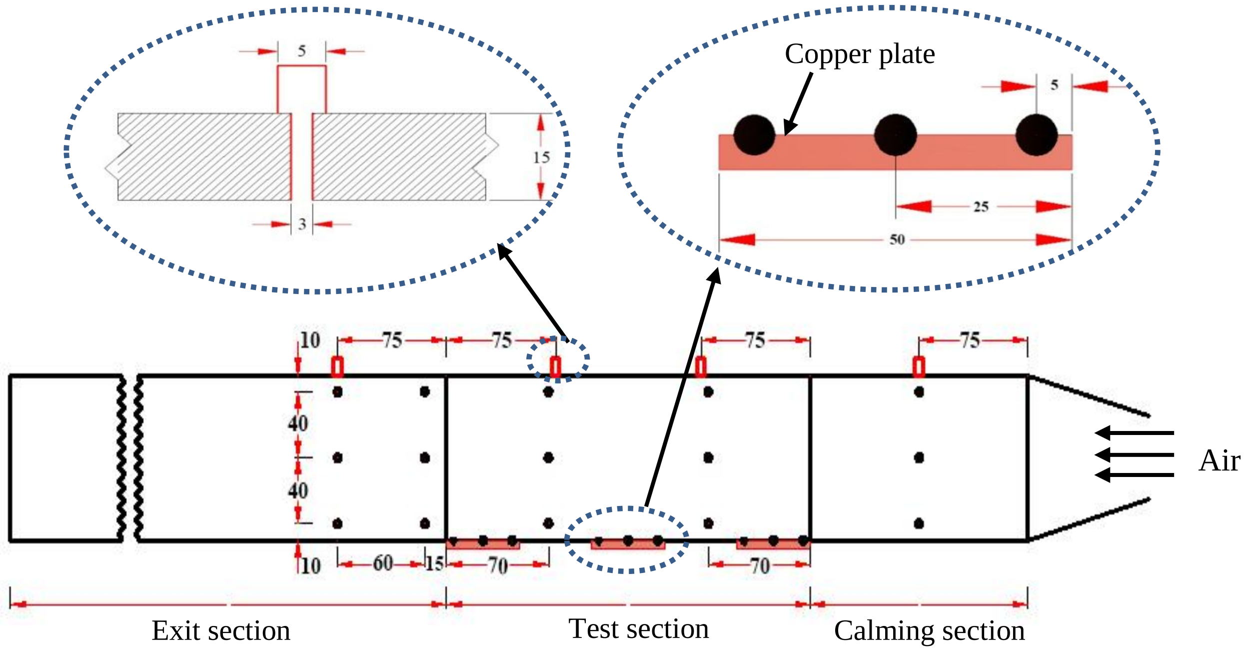


a) Without metal foam blocks.

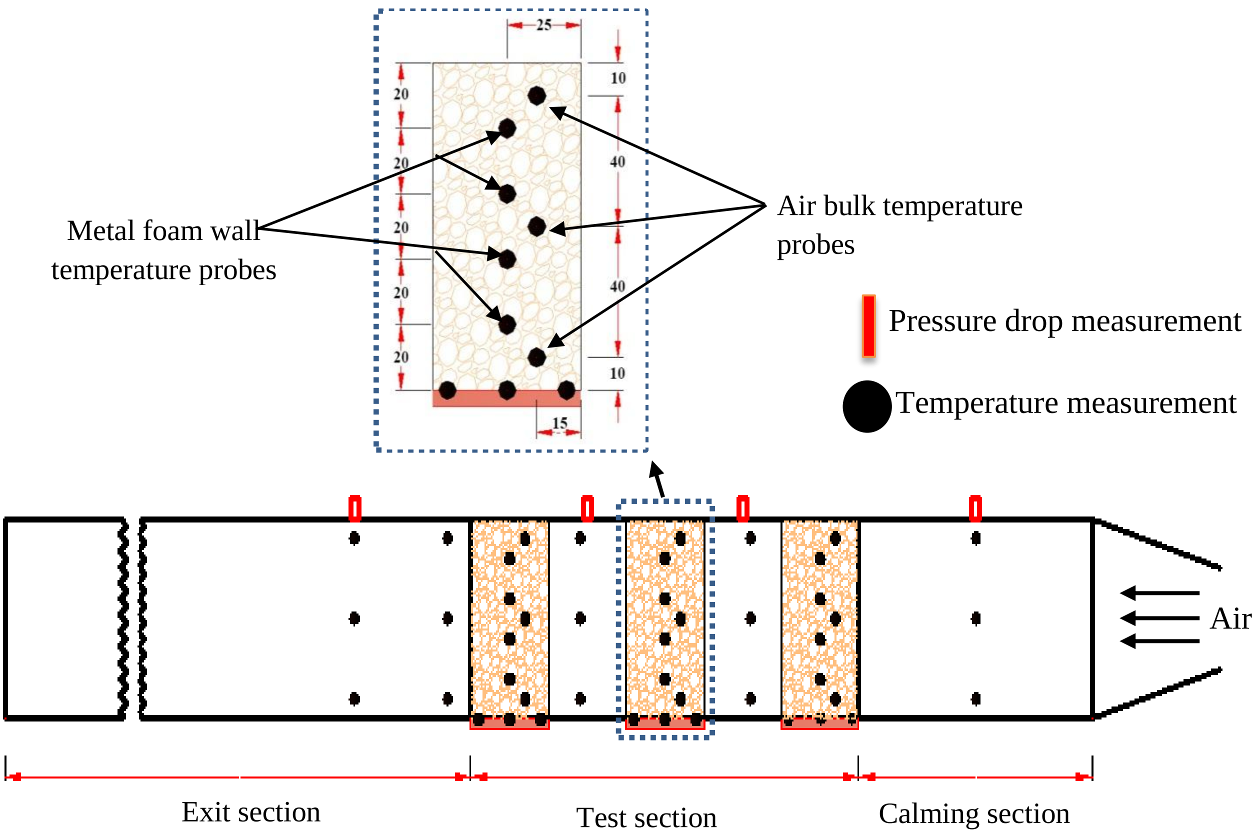


b) With metal foam blocks.

Figure 3. Schematic of the test section (All dimensions are in mm).



a) Without metal foam blocks.



b) With metal foam full rectangular block.

Figure 4. Thermocouples distribution and pressure taps. (All dimensions in mm).

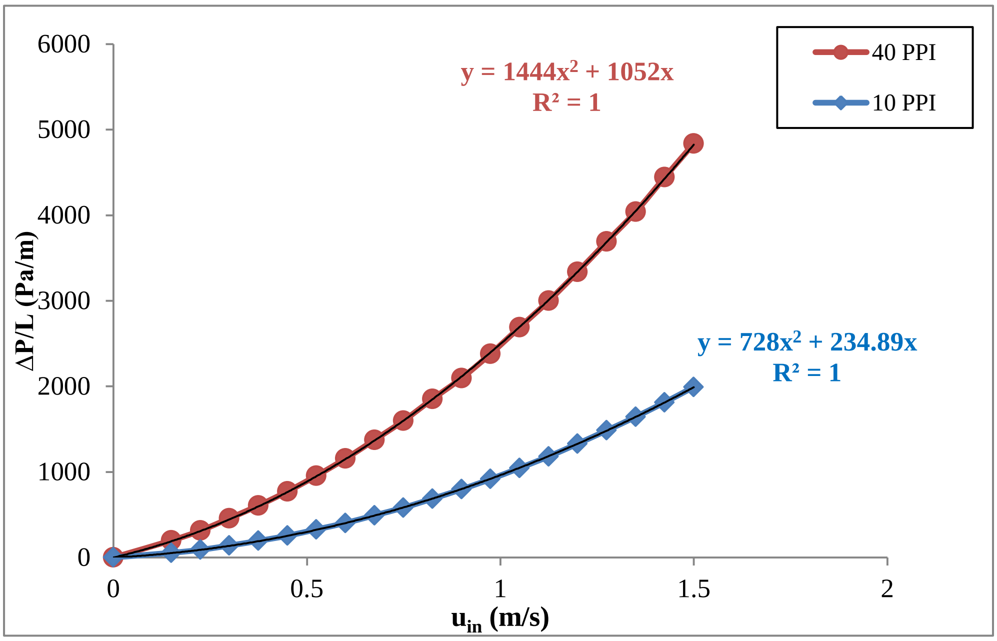


Figure 5. Variation of pressure drop across the metal foam with the velocity at the inlet.

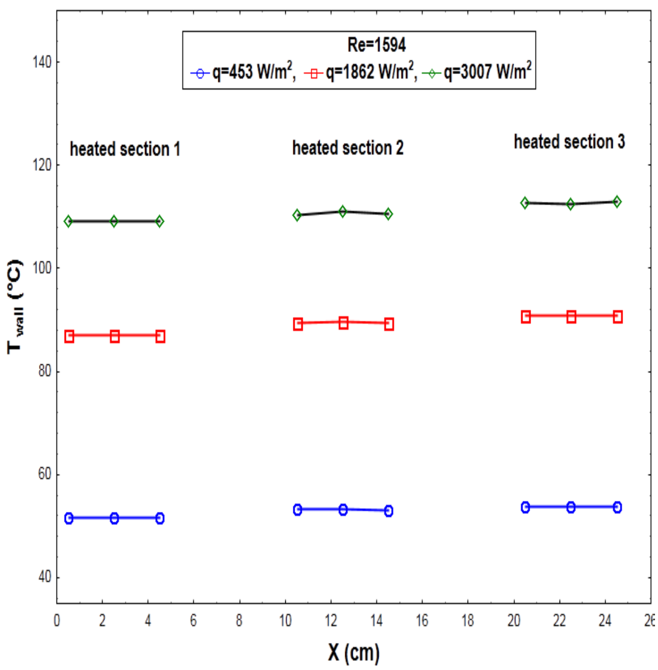


Figure 6. Variation of the wall temperature (fluid case) with the axial distance for different heat fluxes.

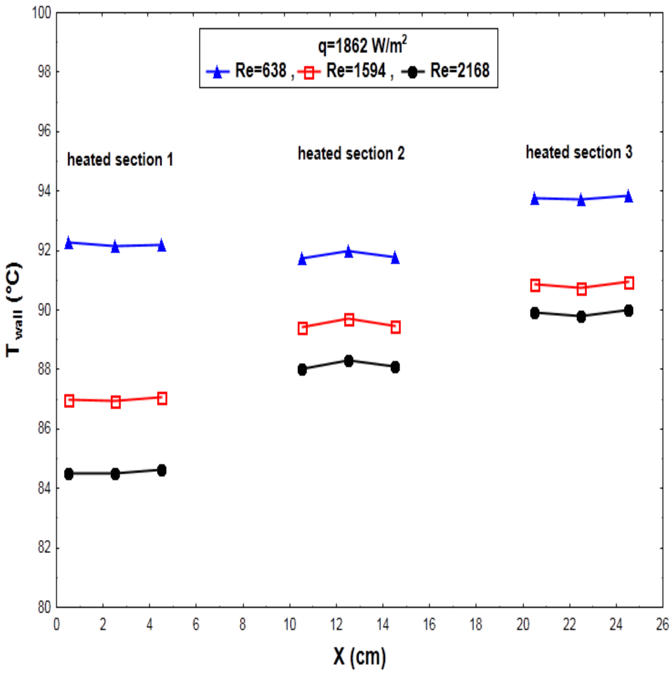


Figure 7. Variation of the wall temperature (fluid case) with the axial distance for different Reynolds numbers.

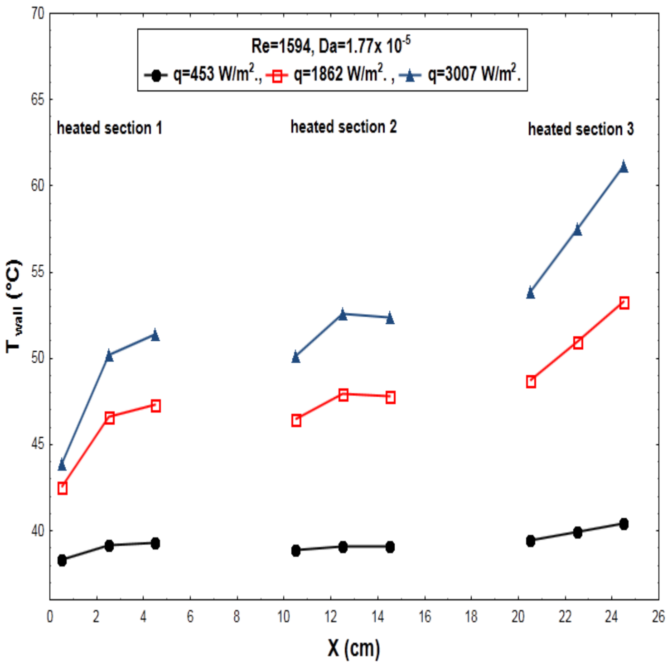


Figure 8. Variation of the wall temperature with the axial distance for different heat fluxes.

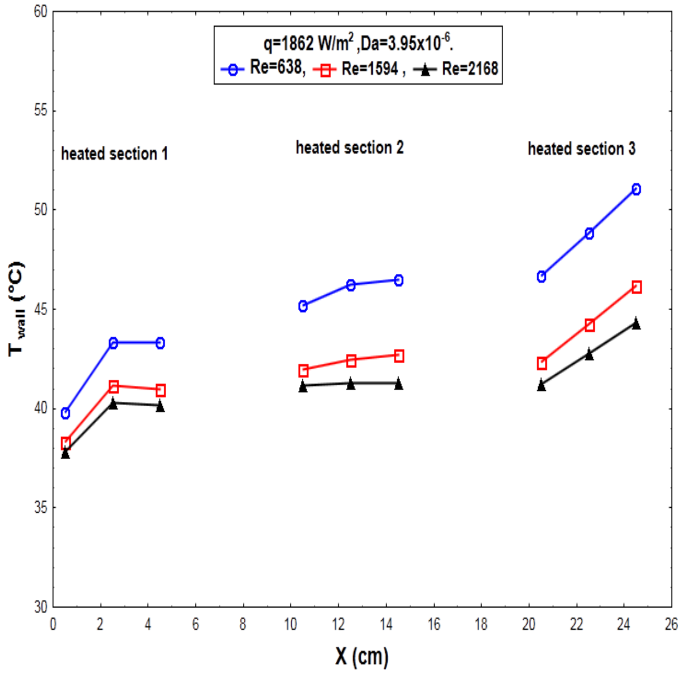


Figure 9. Variation of the wall temperature with the axial distance for different Reynolds numbers.

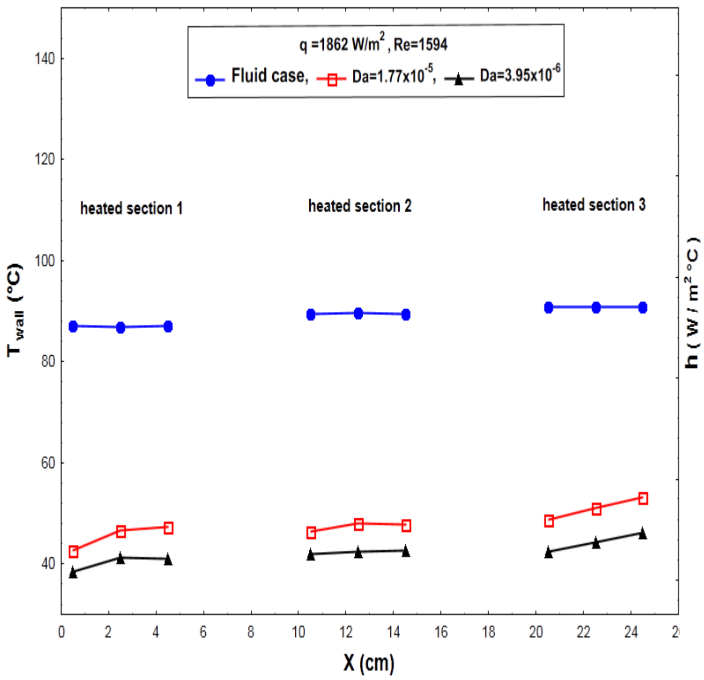


Figure 10. Variation of the wall Temperature with the axial distance for different Darcy number.

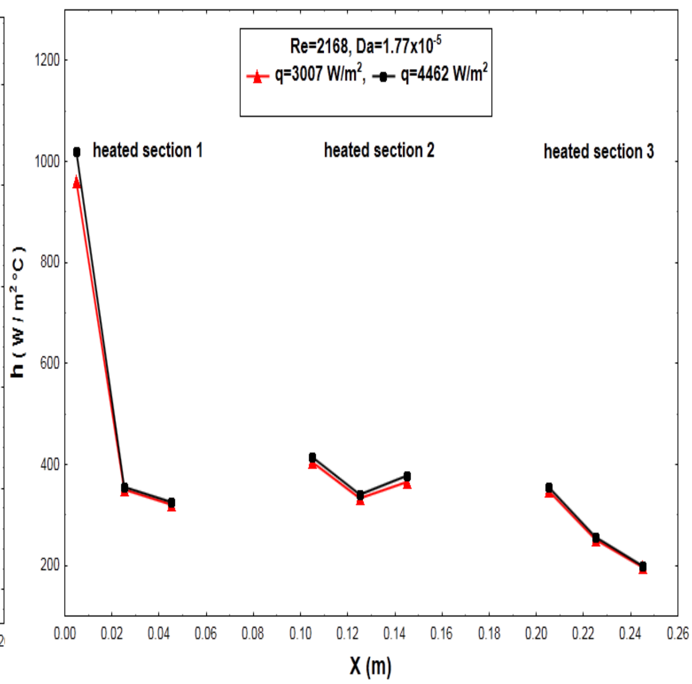


Figure 11. Local heat transfer coefficient with the axial distance for different heat fluxes.

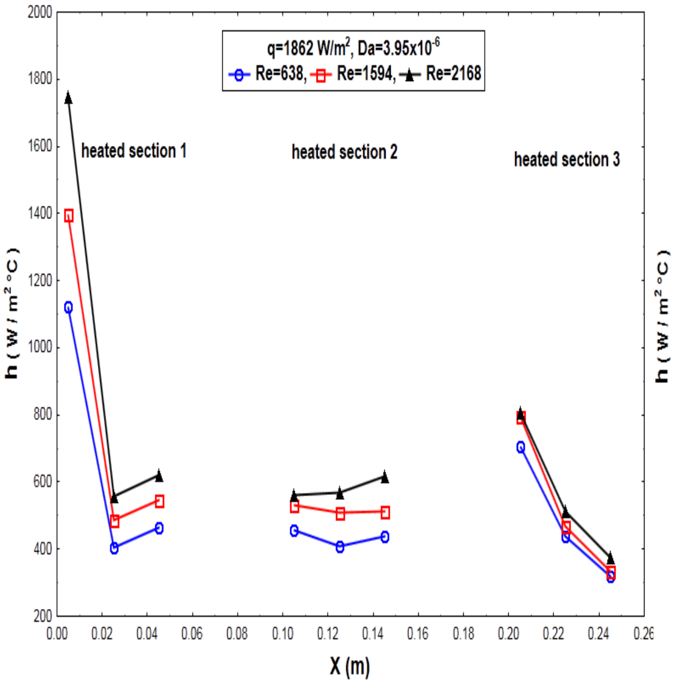


Figure 12. Local heat transfer coefficient with the axial distance for different Reynolds numbers.

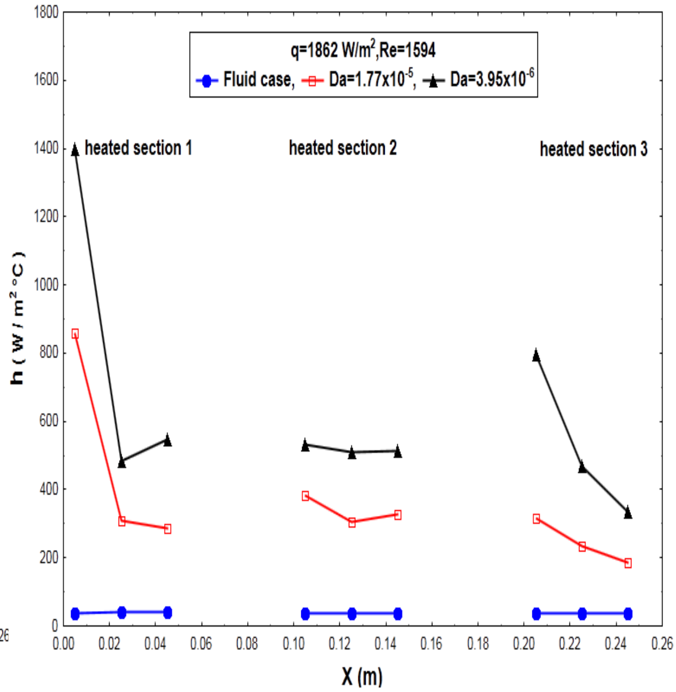


Figure 13. Local heat transfer coefficient with the axial distance for different Darcy number.

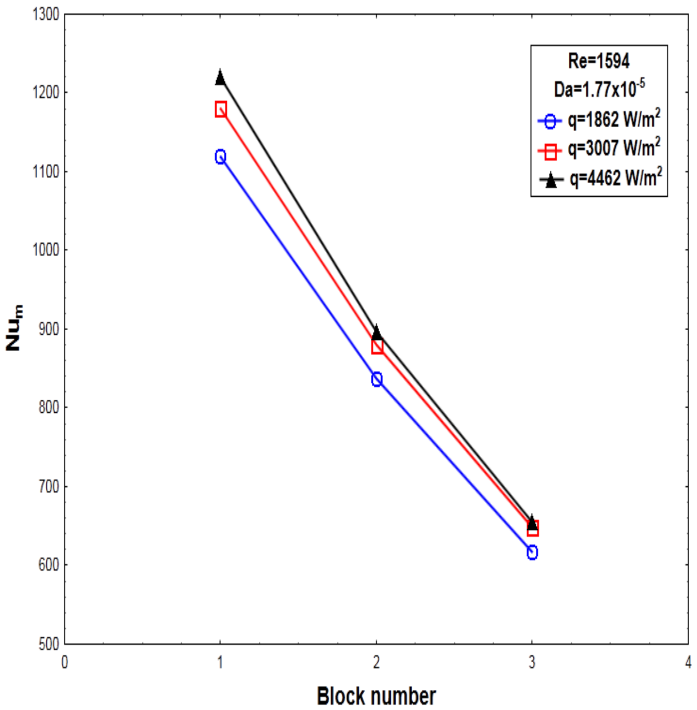


Figure 14. Mean Nusselt number with block number for different heat fluxes.

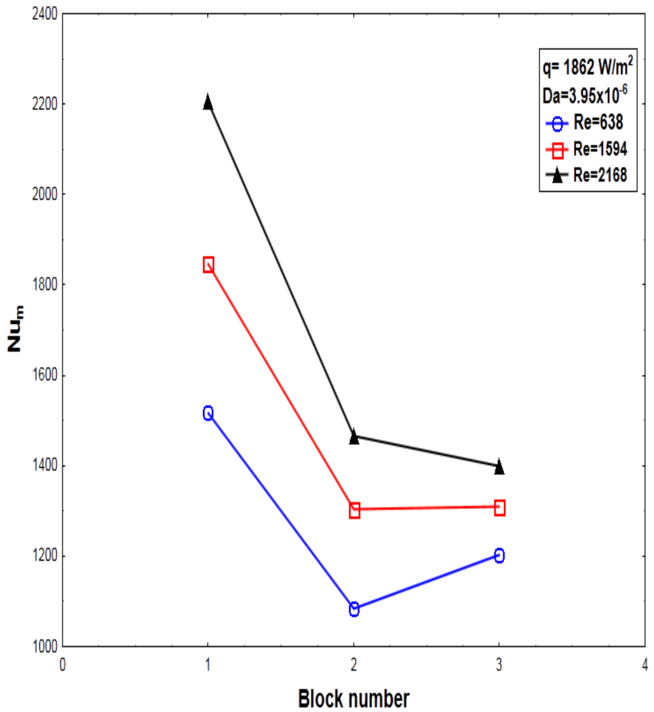


Figure 15. Mean Nusselt number with block number for different Reynolds numbers.

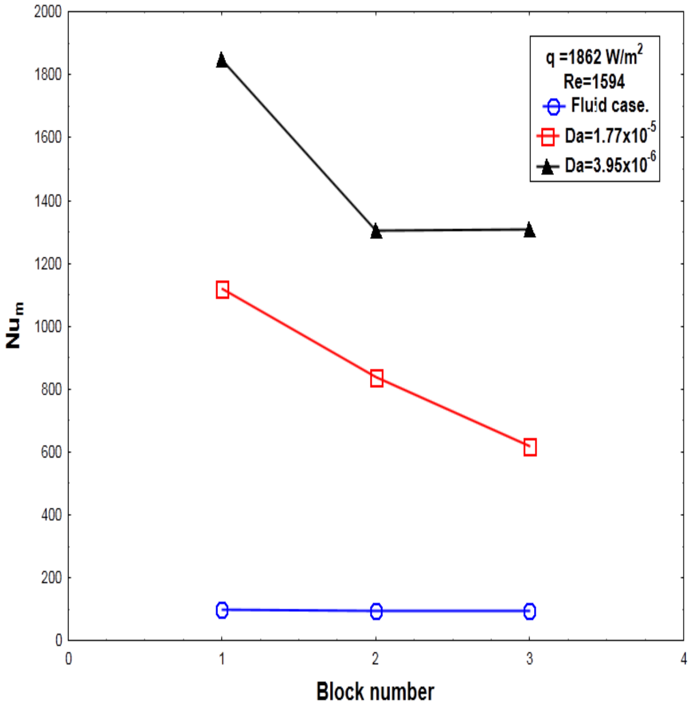


Figure 16. Mean Nusselt number with block number for different Darcy number.

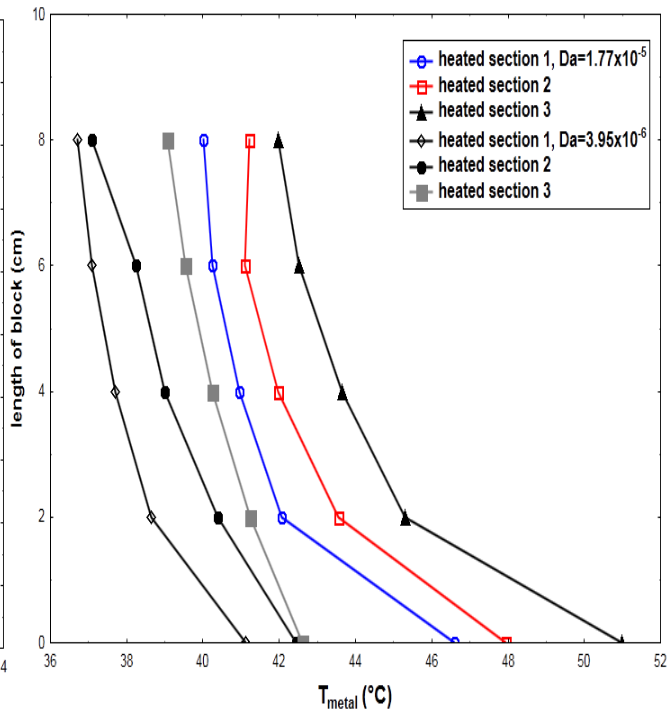


Figure 17. Variation of the metal foam temperature with the length of block for different Darcy Number.

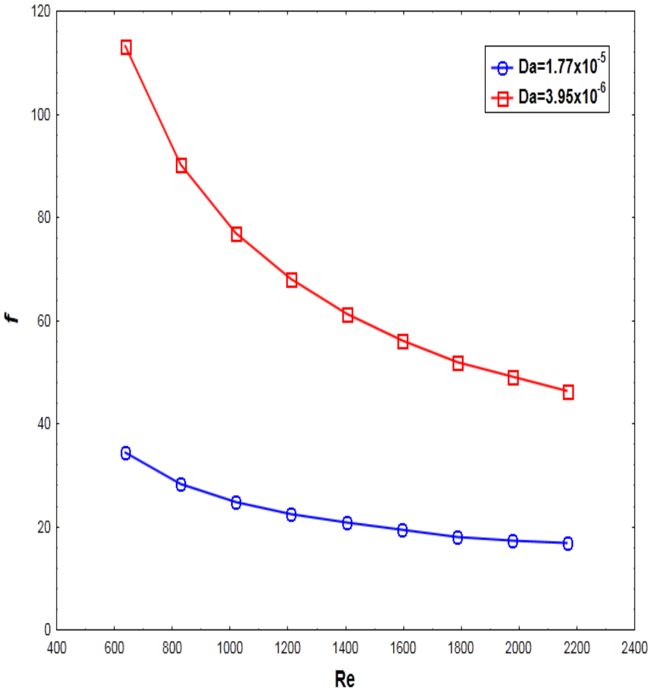


Figure 18. Local friction coefficient with the Reynolds numbers for different Darcy number.

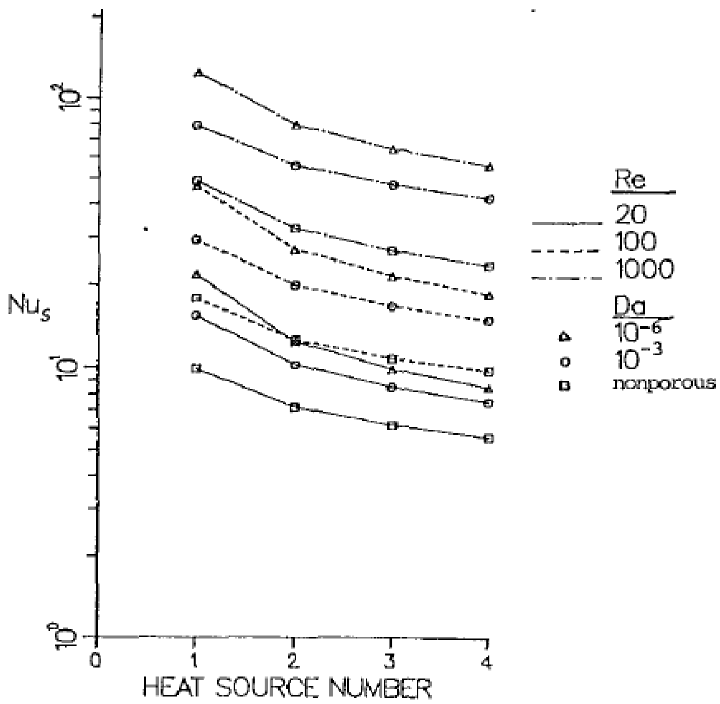


Figure 19. Variation of the average Nusselt Number over each heated section with Darcy and Reynolds numbers, **Hadim, 1994.**