Experimental Study of Mixed Convection Heat Transfer in a Vertical Concentric Annulus Subjected to an Asymmetric Heating

Sarmad A. Abdal Hussein
Assist. Lecturer
Mechanical Engineering Dept.
Engineering College
Baghdad University\Iraq

Abstract

The experimental study had been conducted for a mixed convection heat transfer of water flow through a vertical concentric annulus with asymmetric uniformly heated inner and outer cylinders, and packed with a saturated metallic porous media. The experimental investigation includes a range of Ra from 47903 to 122419 for outer cylinder and for inner cylinder Ra range from 23500 to 62300 and Reynolds number range, which is based on the particles diameter of Red = 4.79, 7.91 and 9.58. Heat flux varied from 752.4 W/m2 to 1990.4 W/m2 on the inner surface, and heat flux varied from 1530.8 W/m2 to 3912.2 W/m2 on the outer surface. Under a steady state condition, the measured data were collected and analyzed. The results show that the temperature variations along the inner and outer surfaces of cylinders are affected by the imposed heat flux variation and Reynolds number variation. The local Nusselt number variation with the axial distance and mean Nusselt number are presented and analyzed. An empirical correlation has been suggested to calculate the mean Nusselt number for the geometry and boundary conditions under investigation. Also a comparison was done between the present experimental work and the results of a previous work for the mean local Nusselt number was made and gave good agreement .

Keywords: Mixed Convection, Metallic Porous Media, Concentric Annulus, Asymmetric Heating.

1-Introduction

The convection heat transfer in porous media packed channel is the subject of intense study of the past two decades because of its wide applications including geothermal energy engineering , transport groundwater contamination, disposal of nuclear waste , reactors chemical engineering, insulation building and pipes, and storage of grain and coal, and so on. The work was dedicated to the studies either natural or forced heat transfer in the porous media. There are only a few of the leaves on the numerical and theoretical studies of mixed convection packed Channel. [1] studied experimentally the flow and heat transfer characteristics in a pipe, filled with steel balls as porous medium and the working fluid is water under constant heat flux condition. In the past [2] studied analytically and numerically in the various flow and heat transfer arrangements. [3] experimentally and theoretically studied the local and average heat transfer in mixed convection with a simultaneously developing upward air flow in a vertical concentric cylindrical annulus with radius ratio of 0.41. A comparison has been made between the experimental and theoretical results for local Nusselt number variation with dimensionless axial distance. [4] studied experimentally and theoretically the convection in a vertical concentric annulus of two cylinders filled with porous medium. [5] carried out a set of experiments for forced convection of Freon-l13for (Pr = 8.06) in a channel packed with small glass spheres (3, 5 and 6 mm in diameter) and with chrome steel spheres (6.35 mm in diameter), respectively. They found that the Nusselt number increases as the particle diameter is decreased for small range of the Reynolds number. [6] studied experimentally the enhancement in the heat transfer from a heated aluminium plate placed in a vertical channel and filled with an aluminum metal foam. The experiments have been conducted for different foam thicknesses. They showed that increasing the foam thickness lead to enhancement of heat transfer by 2 to 4 times over an empty channel for the same Reynolds number. [7] reported experimentally the results for natural convection in a horizontal porous cavity of aspect ratio = 5 and heated below. They supported these results by the effects of fluid flow parameters (Rayleigh and Prandtl numbers), porous matrix structure parameters (Darcy and Forchheimer numbers). [8] investigated numerically the effect of thermal asymmetry on laminar forced convection heat transfer in a plane porous channel with Darcy dissipation. They found that the thermal asymmetry may lead to a reversal of the heat flux at a certain position along the flow at least at one of the channel wall by depending on Darcy, Peclet and Reynolds number for asymmetric temperature field and different heat fluxes across the channel boundaries. [9] used R-113 loop in the experiments of mixed convection heat transfer in a vertical packed channel with asymmetric heating opposing walls. A chrome steel bead of 6.35 mm in diameter as porous media was used. The experiments were performed in the range $2 < Pe < 2200$ and $700 < Ra < 1500$. They obtained a correlation equation for Nusselt number in terms of the Peclet number Pe and Rayleigh number Ra of experimental data. The researchers found that the following three thermal systems exist: natural thermal regime: $105 < Ra/Pe$, mixed convection regime: $1 < Ra/Pe < 105$, and forced convection regime: $Ra/Pe < 1$. [10] investigated experimentally the heat transfer enhancement from the heated wall of a vertical rectangular duct under forced flow conditions, by including porous consists of a pile of metal plates pierced that are used to discuss the distinctive features of the model porous medium to conduct hydrodynamic and heat transfer behavior. It is noted that the largest increase in the average Nusselt number is 4.52 times that for clear flow which is observed with a porous material of porosity of 0.85. Also they explored and discussed the influence for wide range of Reynolds number (5-21) on the flow and heat transfer characteristics of the pipe filled with porous medium.

Most of the reviewed studies relied on natural, forced and mixed convection heat transfer in porous media. In the present experimental work, mixed convection heat transfer which produced for laminar water flow and metallic porous media packed in a vertical saturated concentric annulus with asymmetric heating on the inner and outer cylinders.

The main goal of the present work is to investigate the behavior of water flow field, and process which associated with heat transfer in this system. Mixed convection heat transfer with metallic porous media packed in concentric annulus with asymmetric heating on the inner and outer cylinders of constant heat flux, is experimentally examined. Variation on inner and outer wall temperatures and local Nusselt number are investigated, analysed and presented the general relationship that describe the overall process .

2- EXPERIMENTAL APPARATUS AND PROCEDURE

2.1Experimental Apparatus.

The experimental apparatus, which has been designed and constructed to investigate mixed convection heat transfer in an annulus where the inner and outer cylinders is subjected to a constant heat flux, is shown diagrammatically in Figs.1 and 2. The apparatus consists essentially of cylindrical concentric annulus, test section as a part of an open water loop, the test section is consisting of two stainless steel cylinders that form a concentric annulus of length L=63 cm. The inner cylinder inside and outside diameters are (Di=40 mm, Do=41mm) and the outer cylinder inside and outside diameters are (Di=80 mm, Do=82 mm). The heat source in the inner cylinder consists of nickel-chrome wire surrounded by hollow ceramic discs inside stainless steel tube. The stainless steel tube is fitted with thermal cement to isolate this coil from the stainless steel tube. The length of the heater is (63 cm), while its heating active length is (40cm), the diameter is (1.8 cm), the maximum total power is 500 W, resistance of heat source (16.133Ω) and the maximum applied voltage (180-220) Volt. The outer cylinder is covered by a 3 mm thickness asbestos layer, then a (1) mm in diameter nickel-chrome wire electrically isolated by ceramic beads with diameter (6.5) mm, and then twenty layers of asbestos of thickness (25) mm to reduce the heat loss to minimum value. The active heating length is (40 cm). The cylindrical channel is insulated by three layers of glass fibers sheath to reduce the heat loss to the surrounding. To ensure a gap between two concentric annulus cylinders equal to (2 cm), proper flanges made from Teflon are fitted in the both ends of the cylindrical channel to resist the high temperature caused by the effect of the heat source in the top and bottom of the annular channel. The Teflon flanges inner and outer diameters (Di=39 mm, Do=83 mm). The wall thickness of each Teflon flange is 1.5 cm. Four holes were drilled in each Teflon flange as shown in Fig. 3 but with different purposes. The holes in the lower flange are used to distribute the incoming water from the liquid splitter and guide it uniformly to enter the test section, while the holes in the upper flange are used as an exit passage for the thermocouples extension wires that used to measure inner cylinder wall temperatures. The concentric annulus is packed with stainless steel beads, essentially spherical in shape with an average particles diameters of dp=6 mm. A metallic O-Ring with holes (less than 6 mm in diameter) is placed after an entrance length 13.5 cm from the lower end to achieve uniform inlet water velocity profile to the porous packing.The length of the porous packing was designed to be 40 cm as shown in Fig. 3, and the water was then derived out of the test section by a tube that mounted at 9.5 cm far from the test section upper end to the drain. The temperatures of the outer and inner surfaces of the two cylinders heat source are measured by using (18 type - K) thermocouples installed at nine spaces (13.5 to 53.5) cm, the spacing between thermocouple and other is 5 cm arranged along the outer and inner heated wall respectively . The bulk water temperature distribution is measured by inserting fourth (type - K) thermocouples inside the test section through holes on the outer surface of the concentric annulus as shown in Fig. 3.These holes were filled with a relatively high temperature adhesive material to support the thermocouples extension wires and to prevent water infiltration.The temperature variation on the outer surface of the insulation shield was measured using (4 K-type) thermocouples distributed with an equal pitch, to calculate the heat lost during the experiment by referring to the temperature difference between the heater wall and the ambient. The heat lost is found to be approximately 5% during the whole range of the imposed heat flux. The thermocouples wires by leads through a selector switches, connected in parallel to the digital electronic thermometer (type IDC-420042), was used to record the temperature measurements.

2.1.1Water supply

As shown in **Fig. 1**, the cold water inlet which is supplied to the test section from cold water tank fixed at level higher than original test section, valve, filter, flow meter (0.1- 0.8) LPM and liquid splitter. When the water exit from the test section, it flows through a flexible tube to the drain reservoir placed at the test section exit .

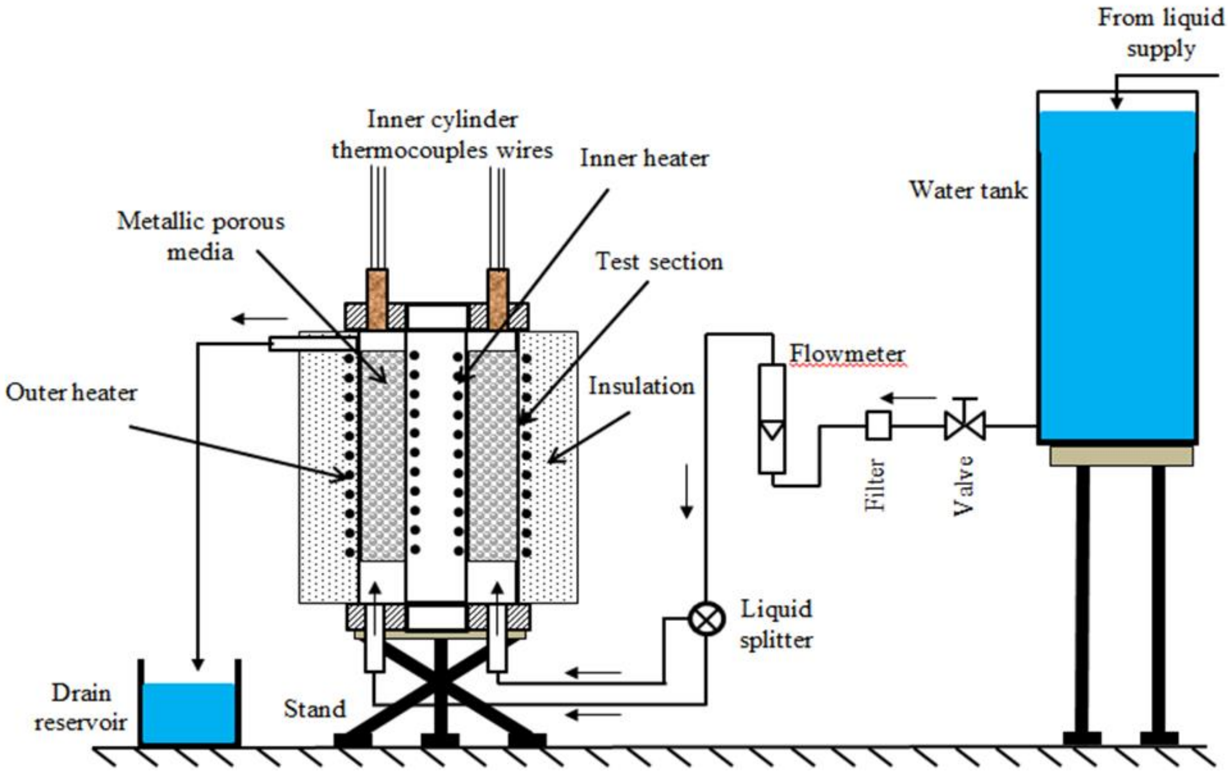


Figure 1.Schematic of the experimental apparatus

1.2 .2Electrical power measurement

The apparatus heating element and heater circuit consist of two variac voltages to adjust the heater input power as required while a digital multi meter was used to measure the heater voltage and ammeter was used to measure the heater current .

2.2Experimental Procedure.

Before starting the experimental measurements and in order to degas the air from the packed annulus, the valve is opened in such a manner that the water flow at the flow meter full range (0.8 LPM) and a moderate heat flux was applied for two hours. Then, the valve was adjusted to give the required water flow rate. Each experiment is performed using the following procedure:

The water is circulated through the open loop. A regulating valve that connected to the water cold tank was used for adjusting the required water flow rate which is measured by flow meter.

The electrical heaters were switched on and the heater input power then adjusted to give the required heat flux.

The supplied voltage and current to the inner and outer heater were recorded to calculate the required electrical power in accordance to the heat flux required .

The apparatus was left at least three hours to establish steady state condition. A thermocouples readings were measured every half an hour by means of the digital electronic multi meter until the reading became constant, a final reading was recorded. The input power to the heaters could be increased to cover another run in a shorter period of time and to obtain steady state conditions for next heat flux and same Reynolds number. Subsequent runs for other Reynolds number ranges were performed in the same previous procedure.

.3DATA ANALYSIS

Simplified steps were used to analyze the heat transfer process for the water flow in an annulus packed with saturated porous media where the inner and outer cylinders were subjected to a uniform heat flux .

3.1Outer and Inner cylinders.

The net heat flux to the saturated porous media is determined from recording the electrical power supplied to the inner and outer heater and applying the following equation:

$$q_i = P_o/A_i \qquad (1)$$

$$q_o = P_o/A_o \qquad (2)$$

Where;

Po= electrical power consumed by heater = I×V .

I = current flow through the heater .

.V = voltage across the heater

A_i, A_o = surface area of the annulus inner and outer cylinders respectively .

Heat losses from the outer heater across asbestos and fibre glass wool layers are calculated to be 5%. These losses are subtracted from the electric power to obtain the net heat transfer rate.

It is important to calculate the absolute permeability(K), the effective thermal conductivity (k_eff) and the porosity (ε) of the saturated porous media, as they used in the dimensionless groups that governs the fluid flow and heat transfer calculations. According to [2]:

$$K = \frac{\varepsilon^3 d_p^2}{180(1-\varepsilon)^2} \qquad (3)$$

$$k_{\text{eff}} = \varepsilon k_l + (1-\varepsilon) k_s \qquad (4)$$

Where k_land k_s are the thermal conductivities of the water and the solid porous media, respectively. The porosity of the packed stainless steel beads was found experimentally using the expression [2]:

$$\varepsilon = \frac{Vol_{total} - Vol_{solid}}{Vol_{total}} \qquad (5)$$

The Reynolds number can be defined according to the particle diameter and the fluid velocity at the inlet as :

$$Re_d = \frac{\mathrm{U_{in} d_p}}{v} \qquad (6)$$

The Grashof number for inner and outer cylinders can be defined as:

$$Gr_i = \frac{g\beta K q_i D_h^2}{k_{\text{eff}}\, v^2} \qquad (7)$$

$$Gr_o = \frac{g\beta K q_o D_h^2}{k_{\text{eff}}\, v^2} \qquad (8)$$

Then Rayleigh number (Ra) can be calculated using the following equation:

$$Ra_i = Gr_i\, Pr \qquad (9)$$

Table 2.Thermophysical properties of the water- stainless steel beads system.

Property	Symbol	Solid	Liquid
Density	ρ kg/m^3	7833	988.1
Thermal conductivity	k W/m K	15.1	0.644
Expansion coefficient	β, K^{-1}	---	0.451×10^{-3}
Dynamic viscosity	μ kg/m.s	---	0.547×10^{-3}
Prandtl Number	Pr	---	3.55

$$Ra_o = Gr_o\, Pr \qquad (10)$$

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

$$h_i = \frac{q_i}{T_{w,i} - T_b} \qquad (11)$$

$$h_o = \frac{q_o}{T_{w,o} - T_b} \qquad (12)$$

Hence, the local and the mean Nusselt number can be calculated as, [2:[$Nu_i = \frac{h_i D_h}{k_{\text{eff}}} = \frac{q_i D_h}{k_{\text{eff}}\,(T_w - T_b)}$ (13)

$$Nu_o = \frac{h_o D_h}{k_{\text{eff}}} = \frac{q_o D_h}{k_{\text{eff}}\,(T_w - T_b)} \qquad (14)$$

$$Nu_m = \frac{1}{L}\int_0^L Nu\, dy \qquad (15)$$

The physical parameters for the stainless steel beads that used in the present study are listed in **Table 1** and the thermo physical properties of the water-stainless steel beads that used in the present study are listed in **Table 2**

Table 1.Physical parameters for each size of the copper beads.

Mean diameter d_p (mm)	Porosity ε	Permeability K (m^2)
6	0.4	3.556×10^{-6}

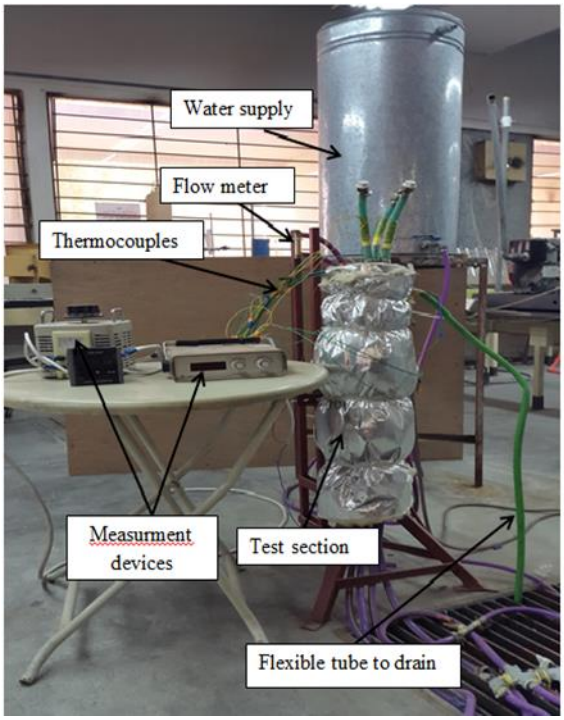


Figure 2. Photograph of the experimental apparatus.

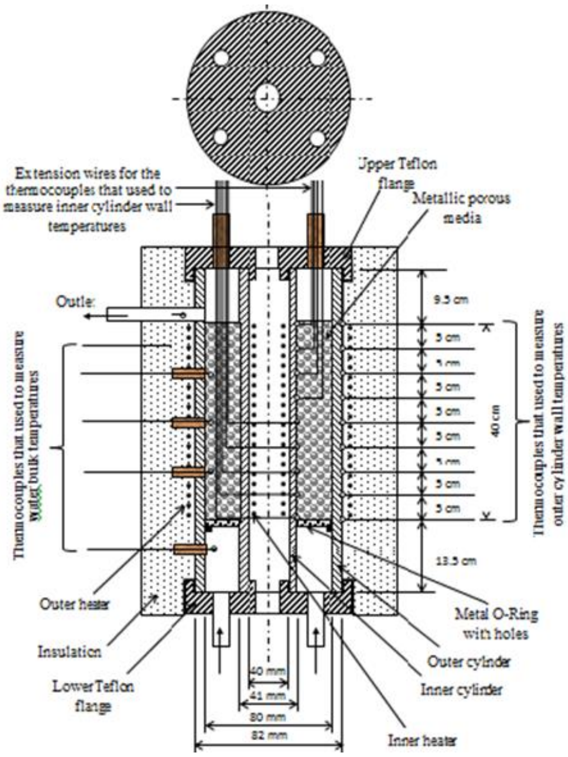


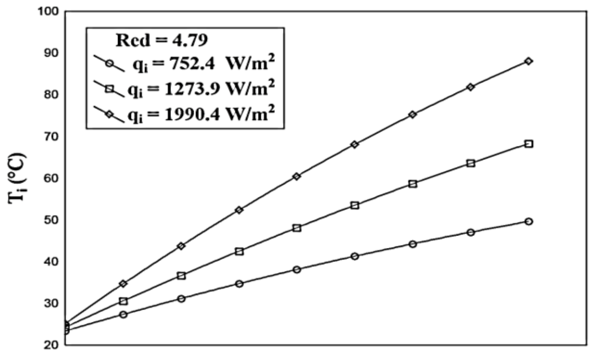
Figure 3. Arrangement and the locations of thermocouples

4. RESULTS AND DISCUSSION

Mixed convection heat transfer is experimentally investigated for water flow in the entrance region of a vertical annulus packed with the metallic porous media with asymmetric heating on the inner and outer cylinders. The comparison with the outcome of previous research is made and discussed, in addition to a discussion of the effects of the various parameters such as heat flux variation and Reynolds number variation on the heat transfer process.

4.1 Temperature Variation.

The effect of variation of the heat flux on the surface temperature along inner and outer cylinders for (Red= 4.79, 7.19, 9.58) respectively, are plotted in **Figs. 4&5.** Figures show that the values of inner and outer cylinders surface temperature increase as heat flux increases. Figures reveal also that the inner and outer surfaces temperature gradually increases at the annulus entrance and attains a maximum point at the end of heating length. The rate of surface temperature rise at early stage is directly proportional to the wall heat flux. The point of maximum temperature seems to move toward the annulus entrance as the heat flux increases. This can be attributed to the increasing of the thermal boundary layer growth along the porous annulus surface due to buoyancy effect as the heat flux increases for the same Reynolds number.



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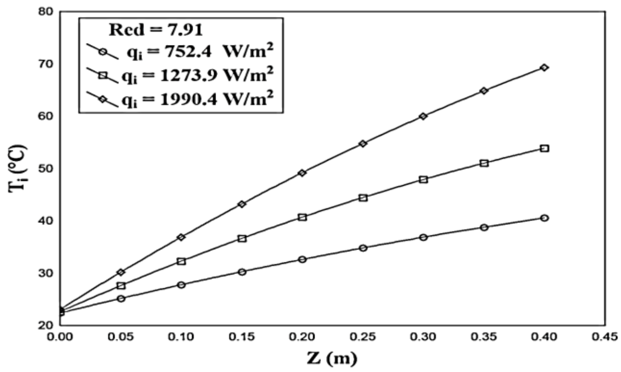

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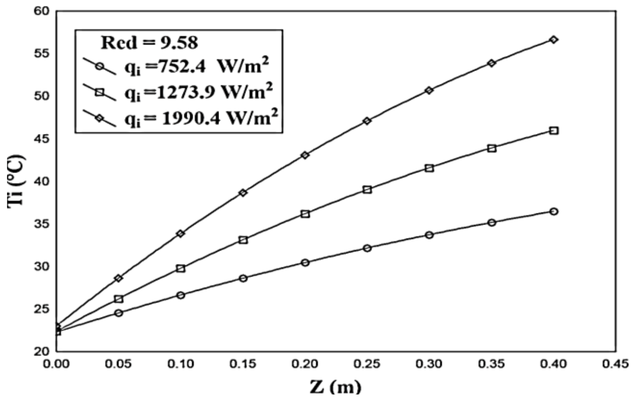

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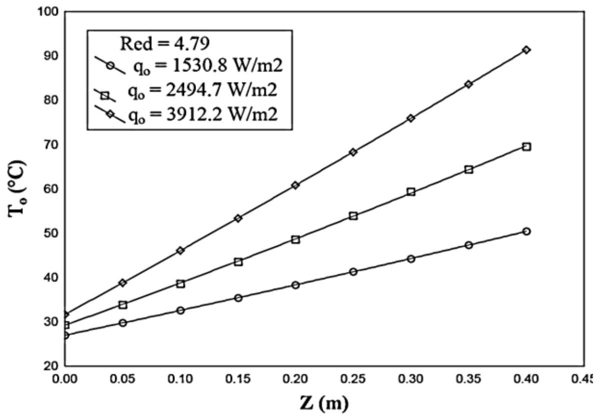
Figure 4. Variation of inner surface temperature versus axial distance for different inner heat flux and (a) Re_d= 4.79, (b) Re_d= 7.91, (c) Re_d= 9.58.



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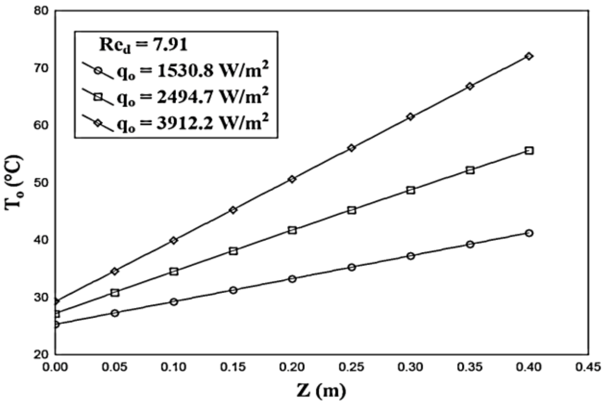

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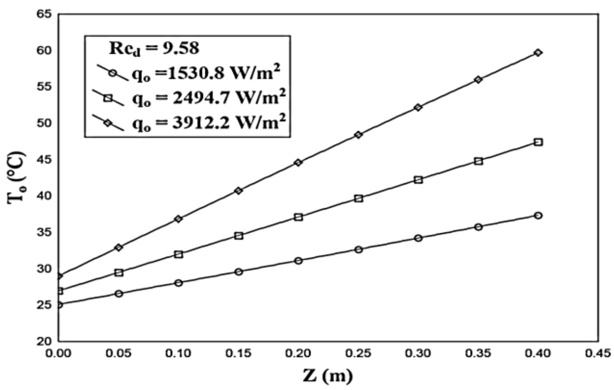

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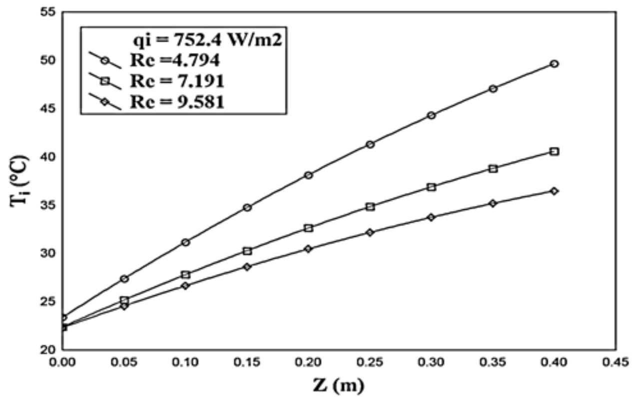
Figure 5. Variation of outer surface temperature versus axial distance for different outer heat flux and (a) Re_d= 4.79, (b) Re_d= 7.91, (c) Re_d= 9.58

Figs6&7 show the effect of Reynolds number variation on the inner and outer cylinders surfaces temperature for heat flux (q_{in}=752.38 to 1990.45 W/m^2&q_{out}= 1530.8 to3912.2 W/m^2); respectively. It is obvious that the increasing of Reynolds number reduces the surface temperature as heat flux kept constant. It is necessary to mention that as heat flux increases the inner and outer cylinders surfaces temperatures increase because the free convection is the dominating factor in the heat transfer process



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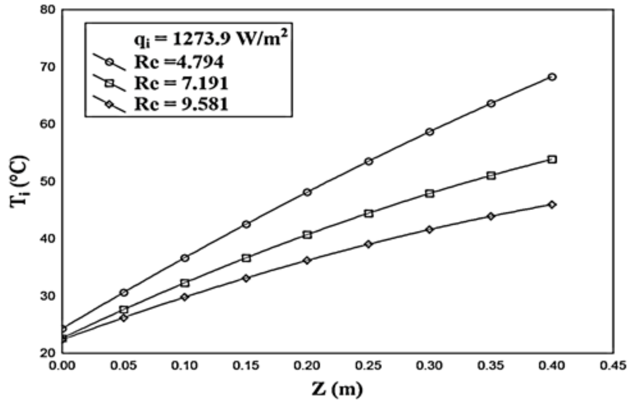

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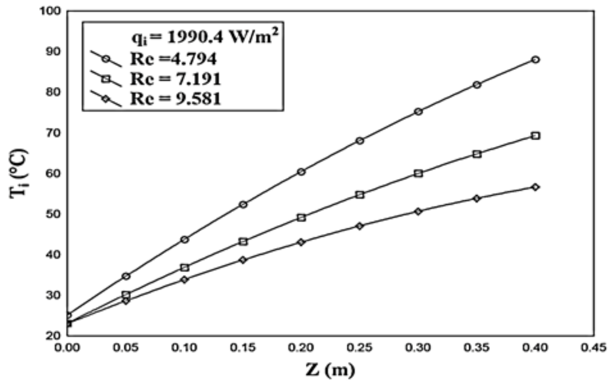

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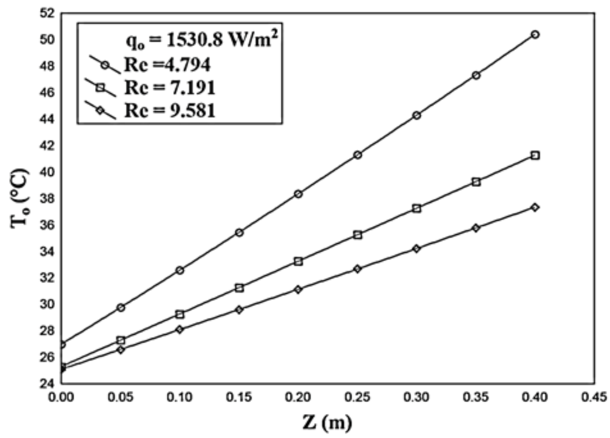
Figure 6. Variation of inner surface temperature versus axial distance for different Reynolds number and (a) q_i= 752.4 W/m^2, (b) q_i= 1273.4 W/m^2, (c) q_i= 1990.4 W/m^2



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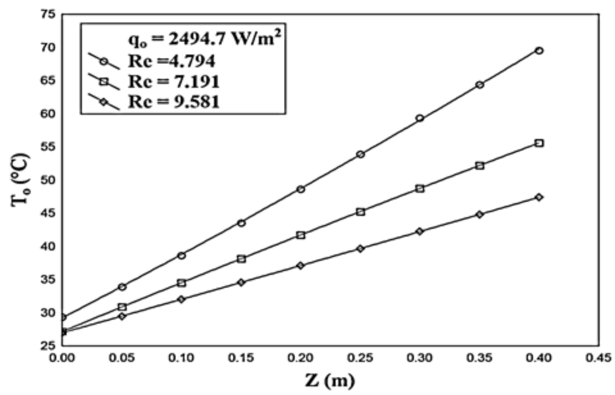

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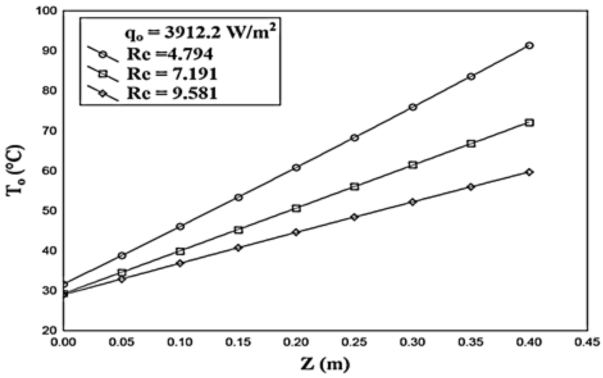

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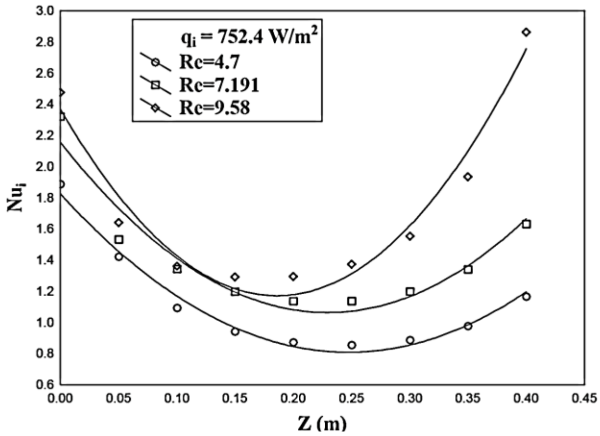
Figure 7. Variation of outer surface temperature versus axial distance for different Reynolds number and (a) q_o= 1530.8 W/m^2, (b) q_o= 2494.7 W/m^2, (c) q_o= 3912.2 W/m^2.

4.2 Local Nusselt Number Variation.

1. Inner cylinder.

The general behavior of the distribution of the local Nusselt number can be found in Figs. 15-20, The heat flux of outer cylinder wall ((1530.8, 7382 and 3912.2) W / m^2 is higher than of the heat flux of inner cylinder wall (752.4, 1273.9 and 1990.4) W/m^2, leading to the buoyancy force pushing the hot fluid toward the inner heated cylinder wall leading to the local Nusselt number started by decreasing from the entrance annulus channel to the point where it reached its minimum value. After this point, the metallic porous media play an important role to promote a more heat transfer from the outer heated wall of the inner cylinder lead to the buoyancy force for the inner heated cylinder wall become high overcome the hot next fluid from the outer heater thereby increase in the bulk water temperature in large quantities, as result as increase in the local Nusselt number values up to the annulus channel exit.

Figs. 8 & 9 show the influence of the Reynolds number variation (Red = 4.79, 7.19, 9.58) on the distribution of the local Nusselt number in the outer hot wall of the inner cylinder (752.4, 1273.9 and 1990.4) W / m^2. It can be seen **Fig. 8** to increase the local Nusselt number with Reynolds number to the same heat flux value. The thickness of the boundary layer decreases with cold liquids and the effect of the dominance of inbound and will cause in the largest local mixing fluids and higher local Nusselt number values.



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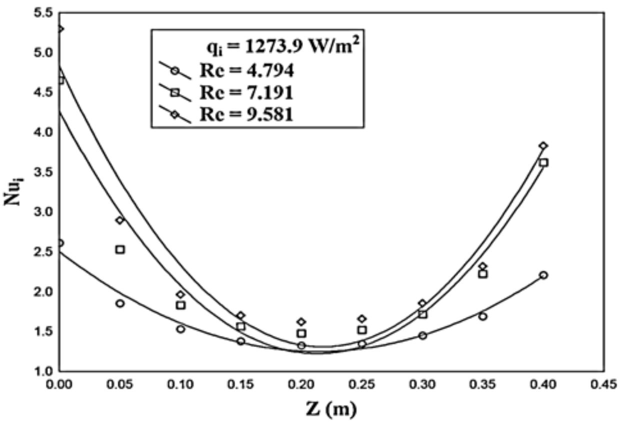

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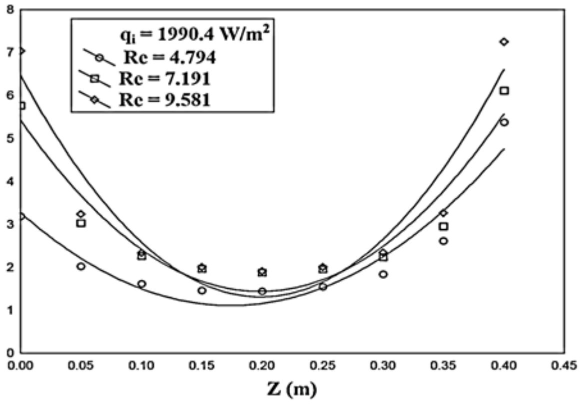

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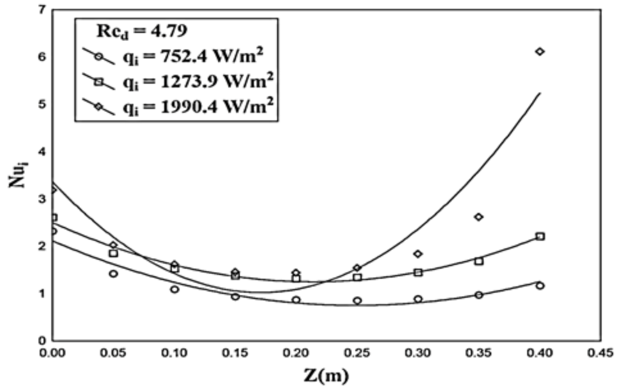
Figure 8. Variation of inner local Nusselt number versus axial distance for different Reynolds number and (a) q_i= 752.4 W/m^2, (b) q_i= 1273.4 W/m^2, (c) q_i= 1990.4 W/m^2.

Fig.9 shows the influence of the imposed heat flux variation on the distribution of the Nusselt number at the outer heated wall of the inner cylinder at (Re_d=4.79, 7.19, 9.58). It can be seen from **Fig.9** that the local heat transfer coefficient increased 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 increased and the thermal boundary layer growth more rapidly and causing a smaller temperature difference between the fluid bulk temperature and the heated wall temperature and as a result a higher local heat transfer coefficient.

inner heat flux and (a) Re_d= 4.79, (b) Re_d= 7.91, (c) Re_d= 9.58.



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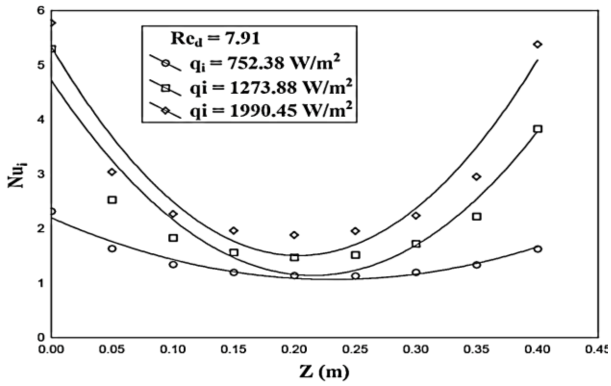

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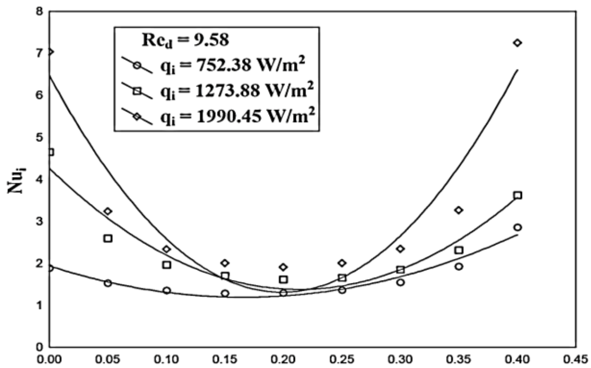

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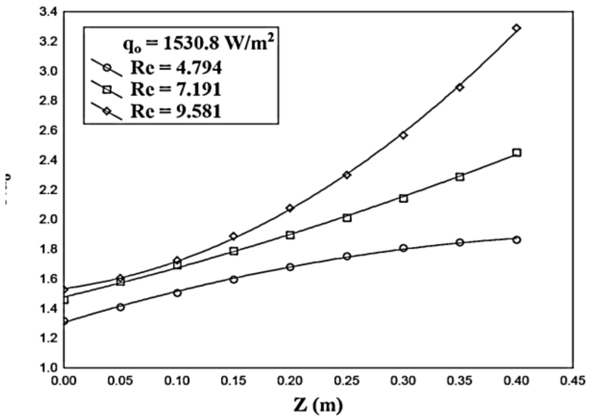
Figure 9. Variation of inner local Nusselt number versus axial distance for different

A general behavior can be seen from the distribution of the inner local Nusselt number in **Figs. (8 & 9)** that the inner local heat transfer coefficient decreased from the channel inlet to a point where it reached its minimum value and then it increased downstream up to the channel exit. At the channel inlet, the small thickness of the thermal boundary layer results in high temperature gradients at the heated wall and high heat transfer coefficient. As the thermal boundary layer increased in thickness upstream, the temperature gradients at the heated wall decreased and caused a reduction in the heat transfer and as a result in the local Nusselt number until it reached its minimum value. After this point and with the continuous heating, the porous media play a crucial role in the enhancement of heat transfer by conducting more heat from the heated wall to increase the fluid bulk temperature and as a consequence increasing the local Nusselt number values up to the channel exit.

2. Outer Cylinder.

The general behavior of the distribution of the local Nusselt number for outer cylinder in **Figs.10 & 11**, that the local Nusselt number begins from the entrance channel to increase until it reaches the maximum value as possible at the end of the outer heat source because of the heat flux of outer cylinder is large compared to heat flux of inner cylinder lead to increasing in the effect of average bulk water velocity along the annulus channel affected by water density decease to ensure continuity provision due to the thermal boundary layer thickness is increased in which the porous media play an important role in enhancing heat transfer through further heat from th outer hot wall to increase the degree o bulk fluid temperature and as a resu increase the local Nusselt numb values up to channel out.

Figs. 10 shows the effects o Reynolds number variation on the loca Nusselt number Nu_o along the oute cylinder for constant heat flux equal to q_o = (1530.8, 7382 and 3912.2) W / m^2; respectively. Results depicted that th deviation of Nu_o value moves toward the left and increases as the Reynold number increases. This situatio reveals the domination of force convection on the heat transfer proces with a little effect of buoyancy force a high Re. As Re reduces the buoyanc effect expected higher which improve the heat transfer process.



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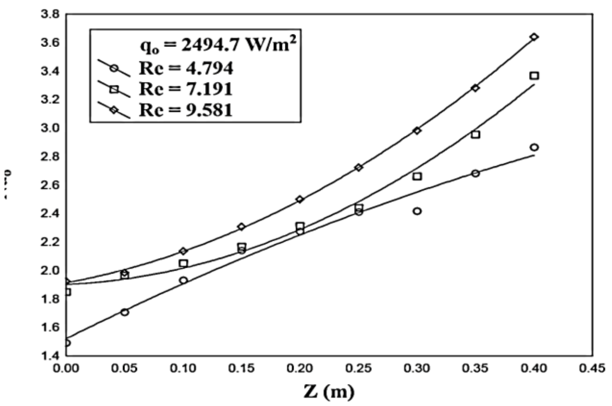

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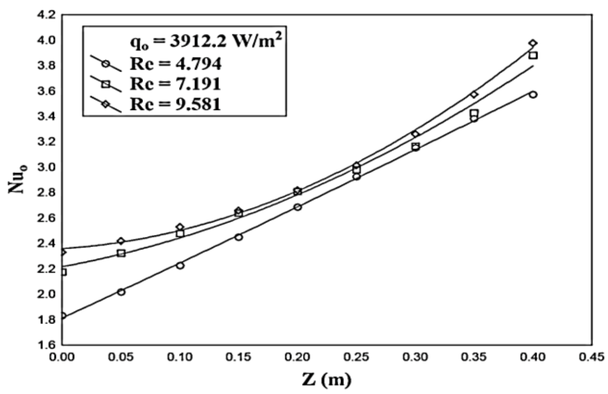

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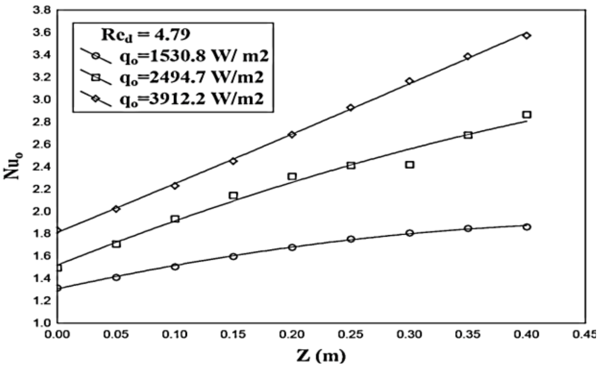
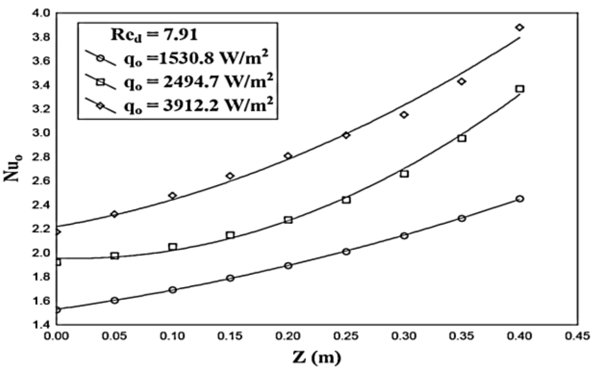
Figure 10. Variation of outer local Nusselt number versus axial distance for different Reynolds number and (a) q_o= 1530.8

W/m², (b) q_o= 2494.7 W/m², (c) q_o= 3912.2 W/m².

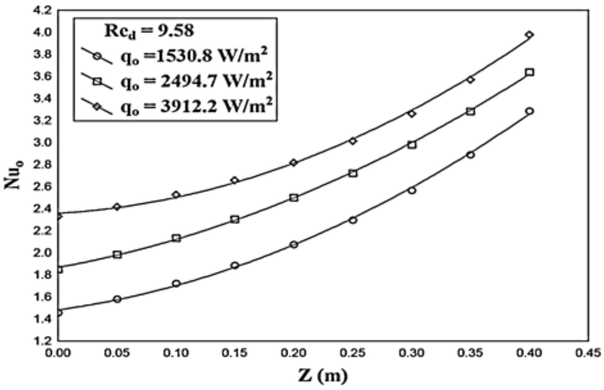
The effect of heat flux on the local Nusselt number along the outer cylinder Nu_o for different heat flux on the outer cylinder and Re= 4.79 is shown in **Figs.11**. The figures show that the local Nusselt number is proportional to the heat flux increase It is clear that at the higher heat flux, the results of the local Nusselt number are higher than the results of lower heat flux. This may be attributed to the secondary flow superimposed on the forced flow effect which increases as the heat flux increases leading to higher heat transfer coefficient.



(a)



(b)



(c)

Figure 11. Variation of outer local Nusselt number versus axial distance for different outer heat flux and (a) Re_d= 4.79, (b) Re_d= 7.91, (c) Re_d= 9.58.

In vertical annulus, the effect of the secondary flow is high, hence at low Reynolds number (4.79) and high heat flux (3912.2 W/m²) situation makes the free convection predominant. Therefore as the heat flux increases, the structure of the cellular motion changes from one cell on each side of the annulus to two and gradually into a metallic porous structure. The cellular motion behaves so as to reduce the temperature difference between the outer cylinder surface and the water flow it which lead to increase the growth of the hydrodynamic and thermal boundary layers along the annulus and causes an improvement in the heat transfer coefficient. But at low heat flux (1530.8 W/m²) and high Reynolds number (9.58) the situation makes forced convection predominant and vortex strength decreases which decrease the temperature difference between the heated surface and the water, hence, the Nu_o values become close to the vertical cylinder values for the same conditions, **[3].**

4.3 Mean Nusselt Number.

Figs.12&13 clear the mean Nusselt number versus Rayleigh number for Re_d=4.79, 7.91 and 9.58 for the inner and outer cylinder, respectively. The figures show an increase the mean Nusselt number when Rayleigh number is increased for the same Reynolds number value because the increase in the buoyancy force effect for higher Rayleigh number values lead to improve in the heat transfer process.

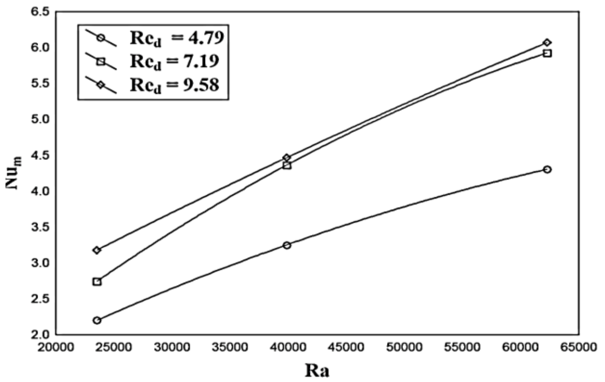


Figure 12. Mean Nusselt number versus Rayleigh number for inner cylinder and Re_d=4.79, 7.91 and 9.58.

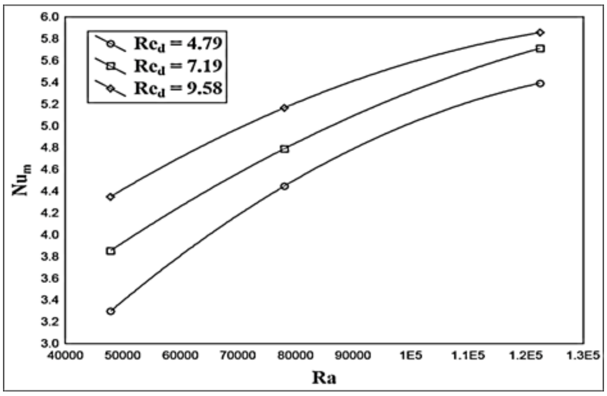


Figure 13. Mean Nusselt number versus Rayleigh number for outer cylinder and Re_d= 4.79, 7.91 and 9.58.

The mean Nusselt number versus Reynolds number is plotted in **Figs. 14 &15** for Ra= 47903.05, 78064.24 and 122418.92) for outer cylinder and Ra= 23500, 39900 and 62300) for inner cylinder, respectively. The figures demonstrate an increase the mean Nusselt number as Reynolds number is increased for the same Rayleigh number value. The reason be returned to the higher fluid mixing that associated with the domination of the incoming cold-fluid effect which causes enhancement in the heat transfer for higher Reynolds number values.

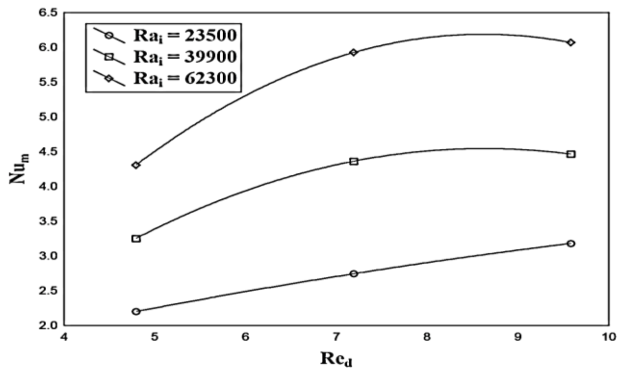


Figure 14. Mean Nusselt number versus Reynolds number for inner cylinder and Rai=23500, 39900 and 62300

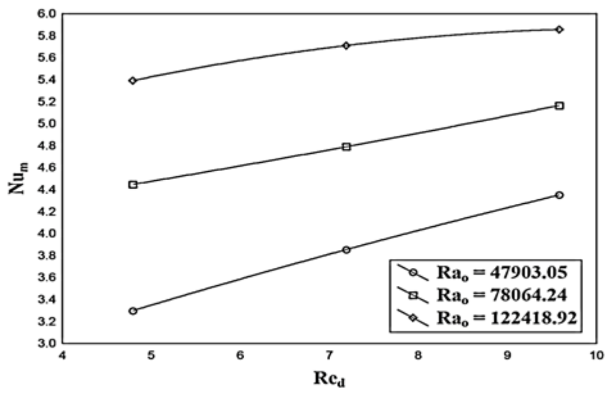


Figure 15. Mean Nusselt number versus Reynolds number for outer cylinder and Ra_o=47903.05, 78064.24 and 122418.92.

4.4 Correlation of Average Heat Transfer Data.

The values of the mean Nusselt number (Nu_m) are plotted in **Fig. 16** for outer cylinder and **Fig. 17** for inner cylinder in the form of log(Nu_m) against log(Ra/Re) for the range of Re_d from 4.79 to 9.58, and Ra from 47903.05 to 122418.92 for outer cylinder, and Ra from 23500 to 62300 for inner cylinder, . All the points as can be seen are represented by linearization of the following equations:

<u>Outer cylinder</u>

$$Nu_m = 1.7092\ (Ra/Re)^{-0.3655} \quad (16)$$

<u>Inner cylinder</u>

$$Nu_m = 2.6193\ (Ra/Re)^{-0.9152} \quad (17)$$

The heat transfer equations for inner and outer cylinders have the same following form:

$$Nu_m = a(Ra/Re)^d \quad (18)$$

The experimental data and the correlation equation agree very well. The standard deviation between **Figs16&17** and the experimental data is about 8.6 % for outer cylinder and 9.3% for inner cylinder. Note that the correlation **Fig.18** is valid for mixed convection of water flowing the metallic porous media packed in a vertical concentric annulus.

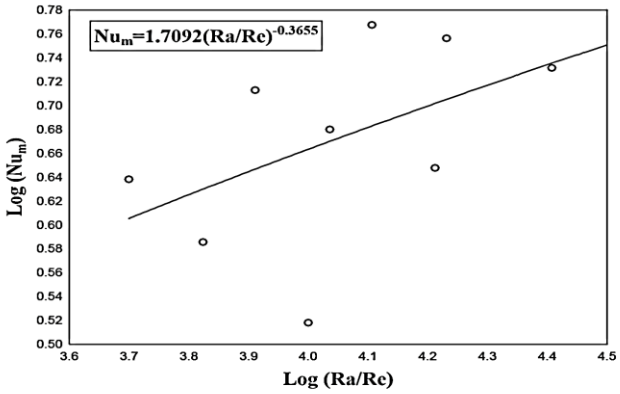


Figure 16. Mean Nusselt number versus Ra/Re for outer cylinder.

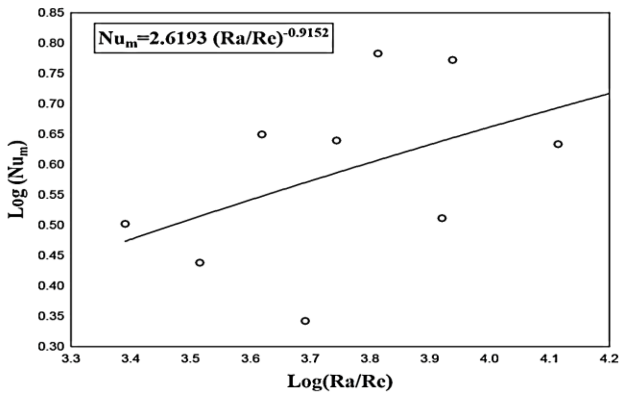


Figure 17. Mean Nusselt number versus Ra/Re for inner cylinder.

4.5 Comparison with Previous Experimental Results.

[10] studied experimentally the mixed convection heat transfer with asymmetric heating of opposing walls and packed with a saturated Porous media. This work is the nearest previous published research that found in the literature using the same setup with the mixed convection. Variations of the average Nusselt number with the increase of the Peclet number at selected Rayleigh numbers are presented in **Fig18**. It is evidenced that at lower Peclet numbers the effect of Rayleigh number is significant. In this regime, the higher the Rayleigh number, the higher is the Nusselt number. The variation of the average Nusselt number is plotted against the Rayleigh number at selected Peclet numbers. As shown in **Fig. 19**, at lower Peclet numbers the Nusselt number increases with increasing the Rayleigh number. The behavior shown in the **Figs.18&19**, respectively are consistent with the results of the present work is shown in **Figs. 12&15** for inner and outer cylinders, respectively (mean Nusselt number increases are also increasing the number of Rayleigh number for the same value of Reynolds number) and **Figs12&13**(mean Nusselt number increases as Reynolds number is increased to the same value Rayleigh number).

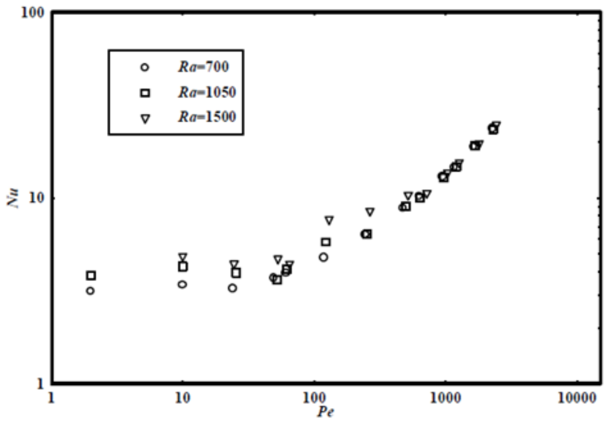


Figure 18. Average Nusselt number of a vertical annulus as a function of Peclet number [10].

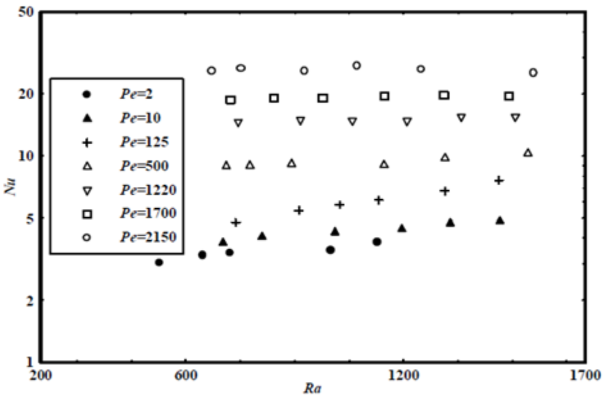


Figure 19. Average Nusselt number of a vertical annulus as a function of Rayleigh number [10].

5. CONCLUSIONS

Experimental investigations of the mixed convection heat transfer for thermally developing laminar flow in an annuals subjected to constant heat flux on both inner and outer cylinders have been performed in the present work. The following conclusions have been drawn from the present investigation.

- When the imposed heat flux increased, the outer and inner annulus surface temperature increased for a constant heat flux.
- The temperature distribution along inner and outer surface of concentric annulus decreased when Reynolds number increased for constant inner and outer heat flux, respectively.
- The inner and outer local Nusselt number increased with the increase of the imposed heat flux and Reynolds number.
- Mean Nusselt number is increased with the increase of Rayleigh number and Reynolds number.
- The effect of buoyancy is small at the annulus entrance and increases in the flow direction.

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7. NOMENCLATURE

A = area, m^2.
β = expansion coefficient, K^{-1}.
d_p = mean diameter, m.
D_h = hydraulic diameter, m.
ε = porosity.
K = absolute pearmability, m^2.
k_{eff} = effective thermal conductivity, $W/m^2.K$.
Gr = Grashof number.
h = local heat transfer coefficient, $W/m^2.K$.
L = effective heating length, m.
Nu = Nusselt number.
Pr = Prandtl Number.
ρ = density, kg/m^3.
Ra = Rayleigh number.
q_w = heat flux, W/m^2.
Q = volumetric flow rate, L/min.
Re_d = Reynolds number based on the particle diameter.
T = temperature, K.
U_{in} = inlet velocity, m/s.
ν = kinematic viscosity, m^2/s.
Vol_{solid} = volume of the metallic porous media, m^3.
Vol_{total} = total volume of the concentric annulus, m^3.

Subscript Meaning

b = bulk.
i = inner
m = mean.
o = outer
w = wall

دراسة تجريبية لخواص انتقال الحرارة والجريان في تجويف حلقي عمودي قيد التسخين الغير المتماثل

سرمد عزيز عبد الحسين
مدرس مساعد
قسم الهندسة الميكانيكية
كلية الهندسة – جامعة بغداد /العراق

الخلاصة

اجريت دراسة عملية لانتقال الحرارة بالحمل المختلط لجريان الماء خلال تجويف حلقي عمودي ذو اسطوانتين متمركزتين, الداخلية والخارجية مسخنتين تسخين منتظم, وتم حشوه بوسط مسامي معدني. البحث العملي تضمن مدى رالي 47903.05 الى 122418.92 للاسطوانة الخارجية وللاسطوانة الداخلية مدى رالي 23500 الى 62300 ومدى رقم رينولد الذي اعتمد على قطر الوسط المسامي 4.79 , 7.91 و 9.58 . التدفق الحراري يتغير من 752.4 واط/م2 الى 1990.4 واط/م2 على السطح الداخلي, والتدفق الحراري يتغير من 1530.8 واط/م2 الى 3912.2 واط/م2 على السطح الخارجي. تحت شروط الحالة المستقرة, القراءات المقاسة حللت وجمعت. اظهرت المحصلة بان تغير درجات الحرارة على طول السطح الداخلي والخارجي للاسطوانتين يتأثر بتغير التدفق الحراري وتغير رقم رينولد.تم تحليل ودراسة تغير رقم نسلت المحلي مع المسافة المحورية ومعدل رقم نسلت . اقترحت علاقات تجريبيه لحساب معدل رقم نسلت للشكل والشروط الحدية التي تم بحثها. وايضا اجريت مقارنة بين البحث المقدم والبحوث السابقة لمعدل رقم نسلت واعطي نتائج جيدة.

الكلمات المفتاحية: الحمل المختلط، أسطوانتين ذات تجويف متمركز، وسط مسامي معدني، تسخين غيرمتماثل