



The Porosity Effect of Stainless Steel Balls on Forced Convection Pipe Flow

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Abstract

An experimental study has been carried out to investigate the effects of stainless-steel balls on forced convection flow in pipe under uniform heat flux. Water is used as the working fluid and stainless-steel balls as a porous media. The Reynolds number range from (5000 to 9000) based on the diameter of the pipe. The experiments were conducted on three various numbers of stainless-steel balls (N) with various diameters (d_p), which give various porosity (0.33, 0.38 and 0.41). These are ($N=2400$, $d_p=1\text{mm}$), ($N=1600$, $d_p=3\text{mm}$) and ($N=750$, $d_p=5\text{mm}$). Results show that, heat transfer coefficient increases with the decrease in the porosity due to the reduction in the space between balls. This led to an increase in turbulence and produced eddies. Furthermore, enhancement in heat transfer coefficient reached its maximum value of (45%) for ball diameter with ($d_p=1\text{mm}$) and water flow rate (9 L/min). New Correlation equations for the average heat transfer coefficient were obtained for three different diameters of balls (1, 3 and 5 mm).

Keywords: Porous Media, Nusselt Number, Particle size, Stainless steel balls.

تأثير مسامية كرات الفولاذ المقاوم للصدأ على تدفق الأنابيب الحراري القسري
أيسر منير فليح ، ساجدة لفته غشيم

الخلاصة:

اجريت دراسة عملية للتحقيق في تأثير كرات الفولاذ المقاوم للصدأ على التدفق الحراري القسري في الأنابيب تحت تدفق فيض حراري منتظم. يستخدم الماء كسائل العمل وكرات الفولاذ المقاوم للصدأ كوسط مسامي. يتراوح عدد رينولدز من (5000 إلى 9000) بناءً على قطر الأنابيب. أجريت التجارب على ثلاثة أنواع من الكرات المختلفة في العدد والقطر (القطر = 1 ملم ، عدد الكرات = 2400 كرة) ، (القطر = 3 ملم ، عدد الكرات = 1600 كرة) ، (القطر = 5 ملم ، عدد الكرات = 750 كرة) والتي تعطي مسامية مختلفة (0.33 ، 0.38 ، 0.41). تظهر النتائج أن معامل انتقال الحرارة يزداد مع انخفاض المسامية بسبب انخفاض المسافة بين الكرات وهذا أدى إلى زيادة الاضطراب والدوامات المتولدة. بالإضافة إلى ذلك ، فقد تم تحسين معامل انتقال الحرارة إلى أقصى قيمة له بمقدار (45%) لقطر الكرة (1 ملم) ومعدل تدفق الماء (9 لتر / دقيقة). تم الحصول على معادلات تصحيحية جديدة لمتوسط معامل انتقال الحرارة لثلاثة أقطار مختلفة من الكرات (1 ، 3 ، 5) ملم .

1. Introduction

Combination between two phases fluid and solid utilized in numerous industrial operations to gain a big ratio of total surface area, so that volume and fluid can take place above a crammed bed of the solid material such as chemical reactors, petroleum tanks, heat exchangers, and geothermal power Niold and Bejan [1]. Pu et. al. [4] showed experimental results of mixed convection heat transfer in a vertical channel filled with chrome steel particles. The size of test section was as follows: length (66.04 cm), width (20.32 cm), and depth (30.48 cm). The diameter of

particle is 6.3 mm. The experiments were achieved at the range of $2 < Pe < 2200$ and $700 < Ra < 1500$. A new correlation equation for Nusselt number was obtained from experimental work. The results showed that a secondary convective cell occurs in the mixed-convection regime by measured distributions of temperature at the five planes for three various values of Ra/Pe . Pei-Xue et. al. [5] investigated experimental fluid flow in rectangular channel filled with different types of particles. The dimensions of the test section are (58 mm ,80 mm ,5 mm). The channel heated from the top by uniform heat flux



and the other side was adiabatic. Experiments had been carried out with two types of particles (bronze and glass) and with diameter of ($d=0.278, 0.428$ and 0.7 mm). The experiments studied the effect of diameter and thermal conductivity of particle and fluid velocity under a wide range of heat flux. The results illustrated that the average heat transfer coefficients have increased with decreasing bronze diameter particle, but decreased when glass diameter decreased. Pei-Xue and Xiao-Chen [2] investigated numerical water flow with uniform sized small particles in channels under uniform heat flux. Darcy's model used in the numerical. In this model, permeability and inertia coefficient were calculated. The numerical results illustrated an increase in mass flow rate which led to increased local heat transfer coefficients and a slight decrease along the axial direction. Also, the difference in temperature between the solid particles and the fluid which indicates the importance of using the thermal non-equilibrium model in porous media. Tzer-Ming et. al. [3] investigated experimental characteristics of heat transfer in rectangular channel filled with solid particles. Air was utilized as the working fluid and brass beads as porous media. The diameters of brass balls were (2, 4 and 6 mm). The Reynolds number based on diameter of channel and particle (755 to 7921 and 38 to 2703), respectively. Effect of relative length of packed channel to diameter of particle and the relative height of packed channel to diameter of particle were studied. The numerical results illustrate Nusselt number increased with the decrease of the relative length of packed channel to diameter of particle. Also, the relative height of packed channel to diameter of particle was less effective on Nusselt number. A new correlation of average Nusselt number was gained. Pamuk [6] studied numerically and experimentally heat transfer flow in pipe filled with steel balls of 3 mm in diameter. Heat flux is utilized on the external surface of the pipe at 7.5 kW/m^2 . The experimental investigations cover the Reynolds number range of (150-500). The Brinkman-Forchheimer is used to study the flow in the porous medium (steel balls) and the Navier-Stokes equation in the fluid region which are simulated by using commercial software Comsol. The results represented by longitudinal temperature distribution along the pipe, velocity distribution at the pipe exit and Nusselt number variation along the pipe. The aim of this work is to investigate the effect of adding stainless steel balls on forced convection heat transfer in a horizontal pipe under uniform heat flux. Moreover, it studies the influence of the porosity on coefficient of heat transfer.

2. Experimental Setup

The experimental setup of the present work is shown photographically and schematically in Fig. 1 and Fig. 2, respectively. The test section is a circular pipe with diameter of (0.025m), thickness (0.003m), length (0.6 cm) and manufactured from copper as shown in Fig. 3. The outer surface of pipe was heated electrically with uniform heat flux by nickel-chrome wires of diameter (0.001m). The heater is electrically

insulated by ceramic beads. A layer of fiberglass insulation of thickness (0.04cm) is applied to insulate walls pipe and reduce the heat loss. Also, a cover of aluminum plate around the pipe, as shown in Fig. 4., Digital voltage regulator and clamp meter are utilized to measure the current and voltage passing during the heater.

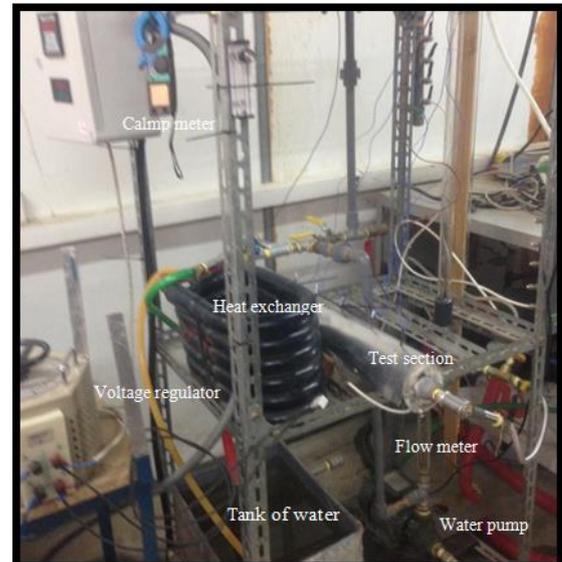


Figure (1): A photograph of the physical problem

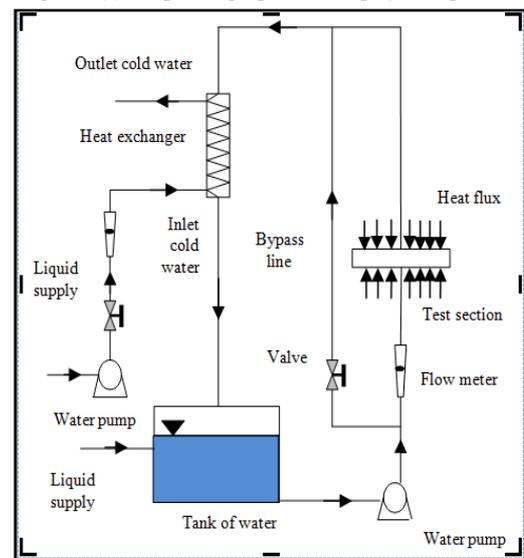


Figure (2): Graphic scheme of physical problem

The test section was filled with stainless steel balls which have thermal conductivity of ($15.1 \text{ W/m}^2 \cdot ^\circ\text{C}$). The experiments were conducted for three various numbers of stainless-steel balls (N) with various diameters (d_p), which give various porosity in the pipe. These are ($N= 2400, d_p=1\text{mm}$), ($N=1600, d_p=3\text{mm}$) and ($N= 750, d_p=5\text{mm}$). The stainless-steel balls inside the pipe were supported by two perforated disks at the inlet and outlet. The inlet and the outlet water temperature measured by using two thermocouples type-K. Also, the temperature inside the test section and outer surface of pipe are measured by six thermocouples distributed along the length of pipe as shown in Fig. 5. All Thermocouples connected to digital thermometer with standard male



plug. Water inlet temperature is constant at (25°C). Heat exchanger double pipe with dimensions of (inner diameter 1.6 cm and outer diameter and 2.3 cm) utilized to eject heat from the test section. Tank of water of dimension (33x53x43) cm and with wall thickness of (0.5 cm), was used to connect cold water to inlet test section. Pipe (Polyvinyl chloride PVC) of diameter (1.25 cm) was utilized to connect the essential parts of the apparatus.

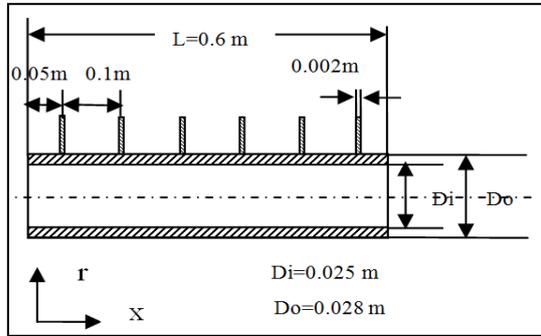


Figure (3): Dimension of test section

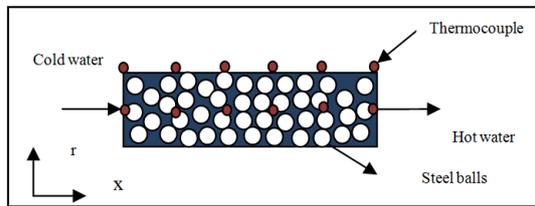


Figure (4): Test section



Figure (5): Position of thermocouples in test section

3. Heat Transfer Calculations

3.1 Equation of heat transfer coefficient for fluid only

The electric power supplied to the pipe surface Mohammed and Salman [7]:

$$Po = I \cdot Vo \quad \dots (1)$$

Where: I is electrical current, Am, Po is electrical power, W and Vo is voltage, Volt.

Total heat transfer:

$$\phi_{conv} = Po - \phi_{loss} \quad \dots (2)$$

$$\phi_{loss} = \phi_{cond} + \phi_{rad} \quad \dots (3)$$

Where: ϕ_{conv} the heat loss by convection, W, ϕ_{loss} is total losses, W, and ϕ_{cond} is conduction heat loss, W.

The convection heat flux can be represented by:

$$q'' = \frac{\phi_{conv}}{A_s} \quad \dots (4)$$

$$A_s = \pi \times D_o \times L$$

Where A_s is surface area, m^2 , D_o is outer diameter of pipe, m, and L is length of pipe, m.

The equation of bulk temperature along the length of pipe:

$$T_{bulk}(x) = T_{in} + \frac{q'' \pi D_i x}{m C_p} \quad \dots (5)$$

Local heat transfer coefficient:

$$h(x) = \frac{q''}{(T_{wall(x)} - T_{bulk(x)})} \quad \dots (6)$$

Where q'' is heat flux per unit area, $W \cdot m^{-2}$

Average heat transfer coefficient:

$$h = \frac{1}{x} \int_0^x h(x) dx \quad \dots (7)$$

The Reynolds number can be defined as:

$$Re = \frac{\rho u D_i}{\mu} \quad \dots (8)$$

Where

D_i = inner diameter of pipe, m

$u = \frac{Q}{A}$ = inlet velocity, $m \cdot s^{-1}$

μ = viscosity, $kg \cdot m^{-1} \cdot s$

ρ = Density, $kg \cdot m^{-3}$

3.2 Equation of heat transfer coefficient for (water -balls)

The local heat transfer coefficient is expressed as:

$$h(x) = \frac{q''}{(T_{wall(x)} - T_{bulk(x)})} \quad \dots (9)$$

The local temperature of the heat transfer surface T_{wall} and the local bulk fluid temperature T_{bulk} were calculated using the measured temperatures.

Average heat transfer coefficient

$$h = \frac{1}{x} \int_0^x h(x) dx \quad \dots (10)$$

The porosity can be described mathematically, (ϵ_{ma}) as:

$$\epsilon_{ma} = 0.32 + \left(0.45 * \left(\frac{d_p}{D_i} \right) \right) \quad \dots (11)$$

Where d_p is stainless steel balls diameter, mm.

or:

The porosity of the balls was measured, (ϵ_{me}) by Pei-Xue et. al. [5]:

$$\epsilon_{me} = \frac{\nabla_{total} - \nabla_{balls}}{\nabla_{total}} \quad \dots (12)$$

Where ∇_{balls} is the volume of stainless steel balls

and ∇_{total} is the total volume of two phases.

The permeability of stainless-steel balls depends on the porosity and diameter and is defined by Pu et. al. [4]:

$$K = \frac{\epsilon^3 d_p^2}{180(1-\epsilon)^2} \quad (13)$$

The error between the value of porosity obtained from mathematical equation and measured value small. Properties of the porous medium used in the experimental work as shown in Table 1.

Table 1. Properties of the porous medium (stainless steel balls)

N	d_p (mm)	ϵ_{ma}	ϵ_{me}	K
2400	1	0.33	0.34	$5.87 \cdot 10^{-10}$
1600	3	0.38	0.369	$8.00 \cdot 10^{-9}$
750	5	0.41	0.421	$3.299 \cdot 10^{-8}$



4. Uncertainty analysis

The accuracy of gaining experimental consequences be based on two factors: the accuracy of measurements and the nature of rig design. The last factor can be affected by the following:

1. The uniformities of heat flux on the wall.
2. Type of thermocouple and method of fixing it on the surface.
3. Heat lost by conduction from side walls.
4. Accuracy of measuring device.

There is no doubt that the maximum portion of errors in calculations referred essentially to the errors in the measured quantities. Holman [8] was used to find errors in the results.

R a function of n independent variables: e_1, e_2, \dots, e_n
 $R=R(e_1, e_2, \dots, e_n) \dots(14)$

This function can be expressed in linear form as:

$$\delta R = \frac{\partial R}{\partial e_1} \delta e_1 + \frac{\partial R}{\partial e_2} \delta e_2 + \dots + \frac{\partial R}{\partial e_n} \delta e_n \dots(15)$$

The uncertainty interval (w) in the result can be presented as:

$$w_R = \left[\left(\frac{\partial R}{\partial e_1} w_1 \right)^2 + \left(\frac{\partial R}{\partial e_2} w_2 \right)^2 + \dots + \left(\frac{\partial R}{\partial e_n} w_n \right)^2 \right]^{1/2} \dots(16)$$

Or;

$$w_R = \left[\sum_{i=1}^n \left(\frac{\partial R}{\partial e_i} w_i \right)^2 \right]^{1/2} \dots(17)$$

Equation (16) is greatly simplified upon dividing by equation (15) to nondimensionalize:

$$\left(\frac{w_R}{R} \right)^2 = \left[\left(\frac{\partial R}{\partial e_1} \frac{w_1}{R} \right)^2 + \left(\frac{\partial R}{\partial e_2} \frac{w_2}{R} \right)^2 + \dots + \left(\frac{\partial R}{\partial e_n} \frac{w_n}{R} \right)^2 \right]^{1/2} \dots(18)$$

Or;

$$\frac{w_R}{R} = \left[\sum_{i=1}^n \left(\frac{w_i}{e_i} \right)^2 \right]^{1/2} \dots(19)$$

Hence the experimental errors that may be happen in the independent parameters are given in the Table 2 which is taken from measuring devices.

Table 2: Experimental errors from measuring devices

Independent parameter (e)	Uncertainty (W)
Temperature	± 0.05 °C
Voltage	± 0.01 V
Current	± 0.01 A
Length	± 0.0005 m
Diameter	± 0.0005 m

The local heat transfer coefficient can be written as follows:

$$h(x) = \frac{q''}{(T_{wall(x)} - T_{bulk(x)})} \dots(20)$$

Therefore, the uncertainty intervals (w) in the measurement as follows;

$$w_{h_x} = \left[\left(\frac{\partial h_x}{\partial V_o} \cdot w_{V_o} \right)^2 + \left(\frac{\partial h_x}{\partial I} \cdot w_I \right)^2 + \left(\frac{\partial h_x}{\partial A_s} \cdot w_{A_s} \right)^2 + \left(\frac{\partial h_x}{\partial \Delta T_x} \cdot w_{\Delta T_x} \right)^2 \right]^{0.5} \dots(21)$$

Or,

$$\frac{w_{h_x}}{h_x} = \left[\left(\frac{w_{V_o}}{V_o} \right)^2 + \left(\frac{w_I}{I} \right)^2 + \left(\frac{w_{A_s}}{A_s} \right)^2 + \left(\frac{w_{\Delta T_x}}{\Delta T_x} \right)^2 \right]^{1/2} \dots(22)$$

The percentage uncertainties in local heat transfer coefficient obtained from experimental results was (1.43 to 2.7) %.

5. Results and discussions

The experimental investigation was carried out using heat flux of 26012.74 W/m² and Reynolds number range from (5000 to 9000). The variation of temperature distribution along the surface of pipe and the temperature of fluid in side pipe with effect of porous media are measured and presented. Three different diameters of balls are examined and resolved.

The variation of wall temperature along the length of pipe is shown in Fig.6 it's obviously, the wall temperature increased along the length of pipe at various values of water flow rate. Also, it was noticed that the wall temperature increases as the diameter of balls increases.

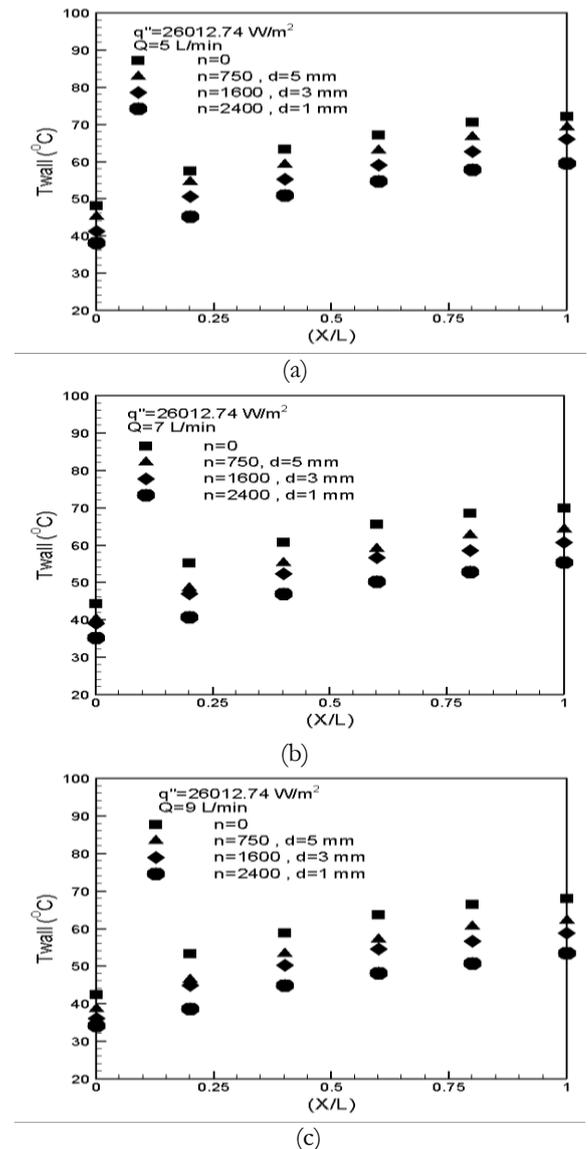


Figure (6): The variation of wall temperature with dimensionless length of pipe



Fig.7 shows results of bulk temperature distribution along length of pipe at ball diameter (1, 3 and 5 mm) and heat flux (26012.74 W/m²). The bulk temperature increases with increases distance from pipe inlet. It's obviously that in the case of adding balls an increase is caused in the turbulence and the produced eddies led to increases in the bulk temperature with different numbers of balls.

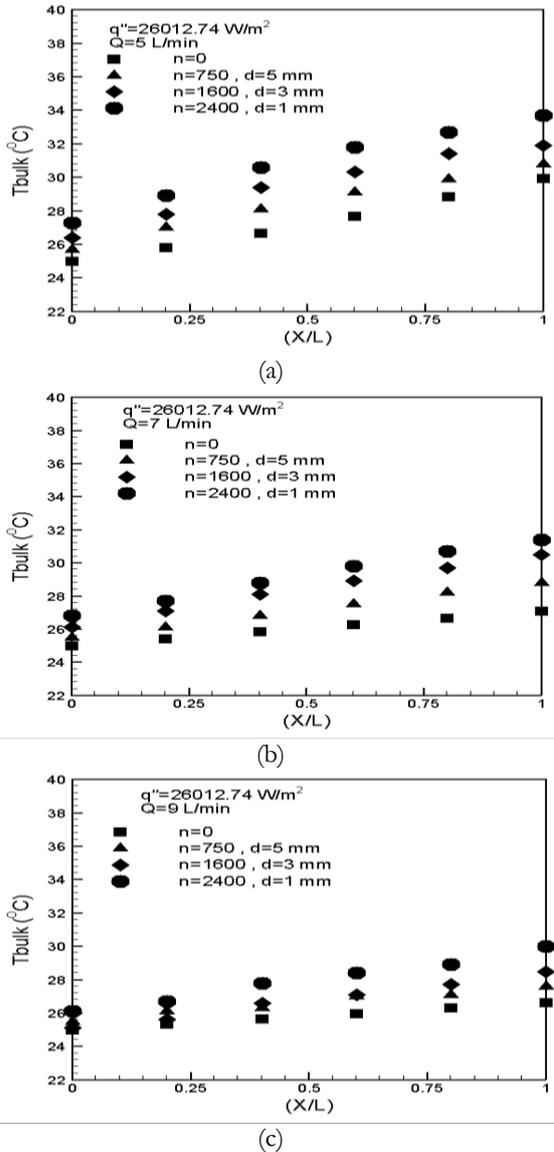


Figure (7): The variation of bulk temperature with dimensionless length of pipe

Fig.8 illustrated the variation of local heat transfer coefficient with length of pipe at different diameter of balls (1, 3 and 5 mm). From the Fig.8 (a,b,c), it is clear that the higher amount of local coefficient of heat transfer is at the entree length because zero thickness of thermal boundary layer and that will decreased along the length of pipe because thermal boundary layer developed. Also, the heat transfer coefficients increased with the decrease of ball diameter.

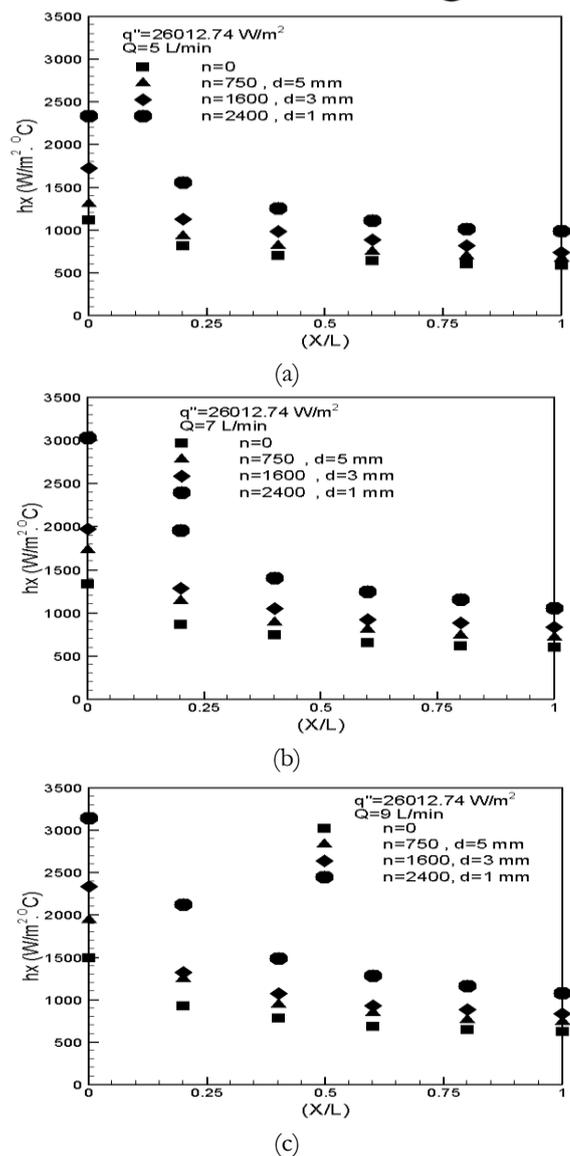


Figure (8): Local heat transfer coefficient variation with dimensionless length of pipe.

Fig.9 represented the effect of Reynolds number variation with average heat transfer coefficient at different diameters (1, 3 and 5 mm) of balls. It is obvious from the figure, as the average heat transfer coefficient increases the Reynolds number also increases. The influence of adding solid balls on the convection heat transfer is much more effective because thermal conductivity of the balls is much higher than thermal conductivity of water. Therefore, thermal conduction during the solid balls and convection heat transfer between the ball and water function is an essential service in the total convection heat transfer on the surface of pipe. Moreover, increasing touch surface area amidst the balls and the fluid with decreasing particle diameter led to an increase in heat transfer coefficients.

The variation of $h_{with}/h_{without}$ ratio with Re number at different diameters (1, 3 and 5 mm) of balls and heat flux (26012.74 W/m²) is shown in Fig. 10 the ratio of enhancement in average heat transfer coefficient increases with Re number increase. The maximum ratio of enhancement from (1.82 to 2) for



ball diameter with ($d=1\text{mm}$) and water flow rate (9 L/min).

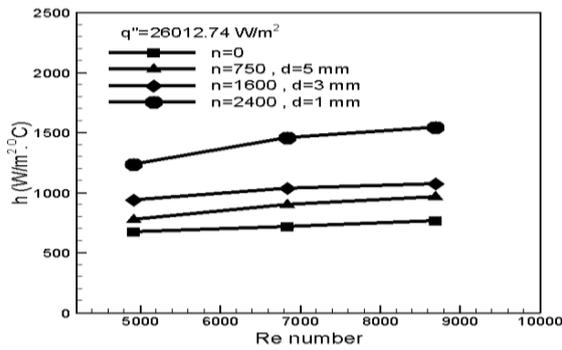


Figure (9): Average heat transfer coefficient versus Re number at different diameters of balls

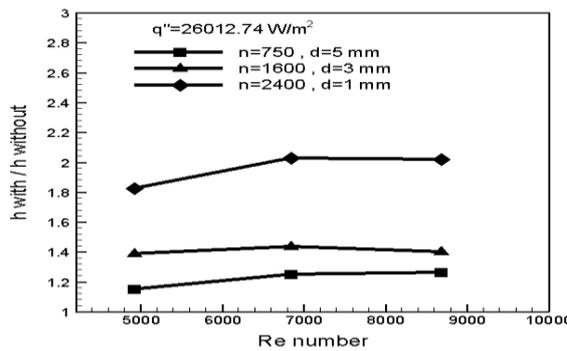


Figure (10): Ratio of average $h_{\text{with}}/h_{\text{without}}$ versus Re number at different diameters of balls

Fig.11 represented the variation of porosity with average heat transfer coefficient at value of water flow rates (5, 7 and 9) L/min and heat flux (26012.74 W/m^2). The average heat transfer coefficient increases considerably as the porosity decreases.

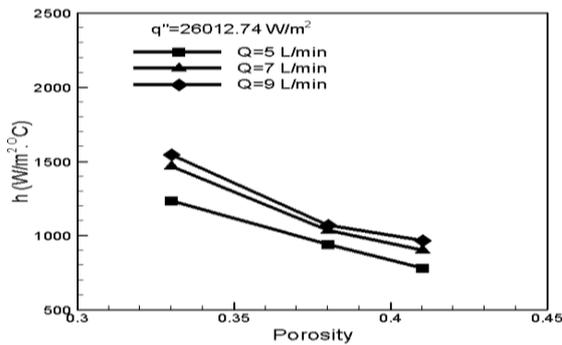


Figure (11): Average heat transfer coefficient versus porosity

Fig.12 indicate the variation in permeability with different porosity. the value of permeability is found to increase with the increased porosity. The average heat transfer coefficients plotted versus the three different ball diameters as shown in Fig.13. A statically method was used to obtain the new correlation equations for heat transfer coefficient for horizontal pipe filled with balls.

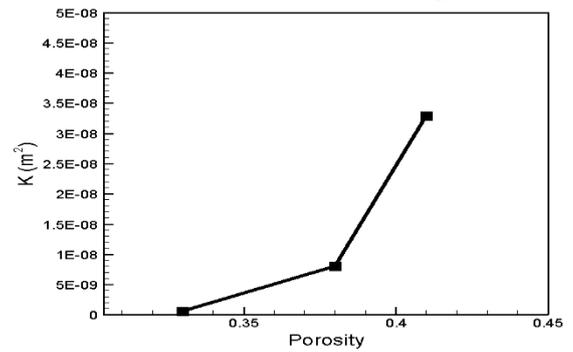
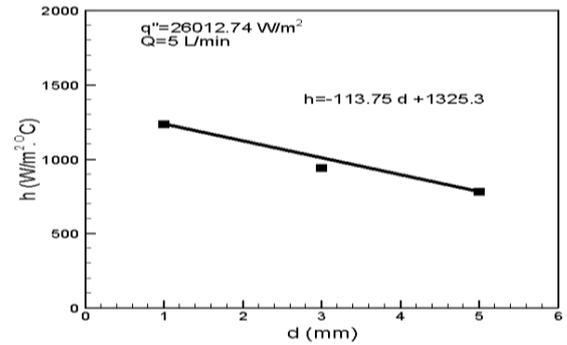
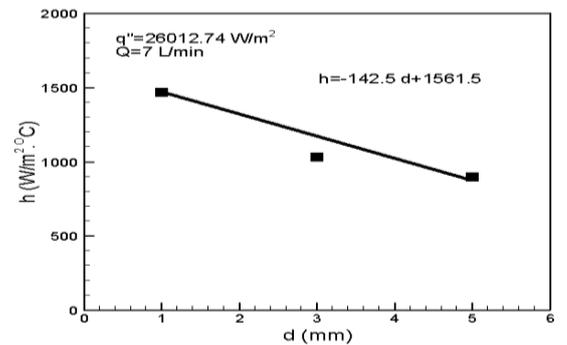


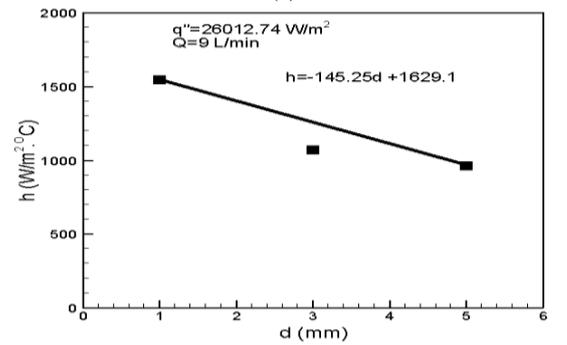
Figure (12): Permeability variation with porosity



(a)



(b)



(c)

Figure (13): Average heat transfer coefficient variation with diameter of stainless steel balls

6. Conclusions

1. The insertion of stainless-steel balls in the pipe caused an increased in the fluid temperature.
2. For fluid case or with effect of porous media, the local and average heat transfer coefficient increases with the increase of the Reynolds number.
3. The size of ball diameter has more effect on the value of porosity.



4. Average heat transfer coefficients increased with decreasing stainless-steel balls diameter.
5. Average heat transfer coefficient decreases with an increased in porosity from 0.33 and 0.41.
6. Average coefficient of heat transfer increases with decreasing porosity of the porous media.
7. The enhancement in ratio of average heat transfer coefficient with stainless steel balls to that without stainless steel balls increased from (1.8 to 2).
8. New Correlation equations for the average heat transfer coefficient were obtained for three different diameters of balls (1, 3 and 5 mm).

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