Experimental study for laminar forced convection heat transfer enhancement from horizontal tube heated with constant heat flux, by using different types of porous media

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Abstract
Forced convective heat transfer for water flowing through a circular pipe oriented horizontally has been investigated experimentally. The pipe was heated at constant heat flux and also packed with two different types of particles (steel and mixture of steel & plastic) as a porous media.

The flow features are an internal, laminar, incompressible and steady flow for a wide range of Reynolds numbers (500, 1000 and 1500) with variable heat flux (200, 400 and 600 W) for each case and three times for each number.

The experimental results comprehended temperature recording on different positions of test sections at steady state depending on the effect of the porous media type and Reynolds number employed (which indicate to the mass flux) in addition to the value of heat flux. The results of the study are represented in terms of temperature distribution to show temperature behaviors in the pipe, besides other diagrams uptake relation of the local and average Nusselt number with data aforementioned above. For each porous type, the results show an increase in local and average Nusselt number with increasing of the Reynolds number for steel and mixture porous media respectively. It has obverse from this study that in a porous steel media the heat transfer will occur mainly by conduction, while in the other types of porous media that the heat transfer will occur mainly by forced convection.

1. Introduction

View the importance of energy and rationalize its consumption for what it has of economic returns for the countries of the world, especially the countries that Lacking natural energy sources like crude oil lost. It resorted to finding alternatives or improving the efficiency of system performance Output or energy consumption [1]. Whereas the porous medium important in isolating and storing energy and its great ability to transfer heat dissipation...
and absorption by the three heat transfer lanes (conduction, convection and radiation). If the heat transfer pregnancy during this medium depends on the principle of increased fluctuation flow resulting from the presence of the porous medium its removal of the viscous fluid layer, as well as the contribution of the substance porosity in heat conduction and depending on the amount its thermal conductivity, so the porous media have many industrial applications, which include thermal insulation techniques in buildings, as well as energy storage granules, geothermal exploitation, and thermonuclear engineering, chemical, solar collectors, and exchangers which thermo-type regenerators contain fillings made of these porous media [2]. An experimental investigation inside a copper circular cylinder for forced convection, laminar flow under different values of constant heat flux was implemented by Tahseen Ahmed Tahseen the working fluid was air, and two types of porous media were used, glass and stone, two values of pressure gradient were employed for both porous media. It’s found that an opposite proportion between dimensionless heat transfer temperature and the length of the tube, the local Nusselt’s number decrease with increasing axial position and increase with increasing Peclet number, and its value for the glass sphere is better than stone with 12.5 % for ΔP= 1.7 kpa and 37.8 % for ΔP= 2.65 kpa. [3]. An experimental study by Thamir K. Salim used a copper circular channel to study the forced laminar air flow, throw pipe filled with two types of porous media, steel sphere and glass at constant heat flux wall with three different Peclet number (2111.71, 3945.52 and 4575.47) and heat flux rang (255.434-2119.565). The results show that an opposite proportion between dimensionless heat transfer temperature and the length of channel and its value was lowest for the glass sphere at highest heat flux. Also, the study declared that when Peclet number increases the average Nusselt number increases for both packed and the value of the average Nusselt number for the glass spheres is greater than the steel spheres [2]. Muhammad Asmail Eleiwi A study experimental and theoretical is investigated to forced convection heat transfer for horizontal copper pipe of 13mm diameter embedded in wooden duct filled with spherical glass for different size (4,8,12 mm) as a Porous Medium, he used Darcy flow model in a theoretical part to derivation and solve energy equation and governing momentum. For the Peclet number range. The result shows the average heat transfer Directly proportional to the Peclet number [4].S Ashok Kumar and M R Rajkumar, they present experiment research to study heat transfer and flow characteristics in a pipe filled with steel balls as the porous medium under constant heat flux condition. And the working fluid is water. The result shows A linear correlation for velocity signifies that at low Reynolds
number, the inertial effect can be neglected, and a linear dependence is established as in accordance with Darcy's law. At the entrance Regine, the variation of temperature is a linear almost, Synergistic with the forced convection in Hagen Poiseuille flow case and after that, the curve departs from this trend and becomes almost flat due to the predominance of porous media effect, and the temperature becomes more or less uniform, resulting in the spreading of heat because of the multidimensional nature of the flow. As velocity increases (Peclet number), heat becomes more dispersed, causing the thermal conductivity due to dispersion to increase [5]. Siva Murali Mohan Reddy.A et al., An experimental study is present for a heat transfer in horizontal pipe with steel balls as porous media under forced convection and used water as working fluid with a constant heat flux. The purpose of the experiment is to know how the Nusselt number varies with porosity, area, and position. They used two arrangement of porous media and porosity value (0.44 and 0.45). The results show a maximum Augmentation in heat transfer with minimum pressure drop was observed for the core of diameter 55mm with porosity 0.44, which was around 4.6 times higher as compared to a clear flow case where no porous materials are used [6]. Suhad A. Rasheed and Jasim M. Abood, they present experimental research of forced convection heat transfer for an air steady laminar flow in a fillly iron duct filled with glass balls as saturated porous media. They used three types of cross-section shape of the heater a circular cylinder shape, a square cylinder shape, and a triangular cylinder shape. They have studied the effect of changing the heater section on forced convection heat transfer with the same values of heat supply for Reynolds number range (1094 – 1510). The result shows that the surface temperature was highest for circular section type than the square section, then the triangular section [7]. Ahmed A. Mohammad Saleh, et al., Investigated experimentally forced convection heat transfer in a different duct cross-sections (circular, triangular, rectangular) for constant heat flux with different Reynolds number with glass spheres porous media filled the duct with air as working fluid. The result show The local surface temperature in the axial direction decreases as Reynolds number increases at constant heat flux. For the constant heat flux, the average Nusselt number increases as the Reynolds number increases. The local surface temperature for the triangular duct is more than in a rectangular duct, which is more than in the cylinder duct. At the same length and hydraulic diameter, the porosity in the triangular duct is more than porosity in the rectangular duct, which is more than in cylinder duct. The local Nusselt number for the triangular duct is more than the local Nusselt number in the rectangular duct, which is more than in cylinder duct [8].
Shahram Baragha et al., In this study, a single-phase flow of air in a channel having circle cross-section with different arrangements of porous for in three different flow (laminar, transient and turbulence) is experimentally studied. Results from these experiments show that the presence of porous media leads that the thermal flux applied to the channel walls transferred to the fluid due to creating a uniform space and high conductivity of porous media. Also, the fluid mean temperature increases and this leads to a decrease in the temperature difference between the mean temperature of the fluid and the channel wall. Also, the study shows that the best heat transfer enhancement obtains when the channel is fully filled with porous media (in both laminar and turbulent flows) [9]. Nicola Pastore et al., An experimental study is presented to evaluate the dynamics of heat transfer forced convection in a plastic pipe fixed vertically and thermally isolated by a roll of elastomeric foam, filled with two different sizes of grain as a porous media, they used water as working fluid enter from bottom of test section and out from top of it by using different range of Reynolds Number for smaller one (5.7–22.5) and for bigger one (23.5–105.5), the result shows that the heterogeneity of the porous medium affects heat transport dynamics, this mean the small porous media able to storage large amount of heat than bigger one [10]. Yanzhe Li et al., An experimental study is presented to show enhancement on heat transfer for a partially filled tube with copper metal foam for airflow throw it with high temperature. They study the effects of filling thickness of the metal foam and pore density on the gas heat transfer in the tube for a different range of filling rate, Pore density of metal foam The experiment results show they can be obtained a high heat transfer coefficient when the density of pore is high for the same filling rate and porosity. However, the comprehensive evaluation index of metal foam with a lower porosity is higher. Thus, to obtain more efficient heat transfer, they must be maintaining a lower porosity metal foam inserted at a higher filling rate, together with a low Reynolds number [11].

2. Experimental Work

The experimental origin is schematically illustrated in figure (1) and (2), as it consists of a circular section channel (copper tube) of internal and external diameter (38.4225 mm) and (41.065 mm), respectively and the actual heated test clip length is (1000 mm). The tube is supplied with water by an electrical water centrifugal pump, and the flow is controlled by using a bypass valve that returns the excess water to the tank. As for the porous fillings, which are steel balls with a diameter of (7.91 mm) and a thermal conductivity (44.5 W/m.K),with porosity of (0.39187) and the second filling
is a mixture of (steel and plastic) balls with a diameter of (7.83 mm) and a thermal conductivity of (22.945 W/m.K) with porosity of (0.44551). The channel surface temperature was measured using thirty-two thermocouples installed along the channel, thirty of which were used to measure the channel surface temperature for the test section, each position has three thermocouples, where the reading rate is taken for every three thermocouples, which represents one out of ten positions, as shown in figure (3) and position in table (1), one thermocouple to measure the temperature of entry and another one to measure the temperature of the exit. Where the test section was heated by a 1000-watt heater, and to reduce the heat loss for test section, it was thermally insulated by wrapping it with two layers of thermal insulation, the first layer of (50 mm) thickness of fiberglass wool type and the second one of (15 mm) thickness of Isocom type. A variac used to adjust the heater input power as required. The following procedures are done during experimental of the project:

1. The water pump is switched on with specifying fixed temperature of water to circulate the water through the test section, by using valve at the entrance and the by-pass valve for adjusting the required flow rate and Reynold number.
2. The power supply and data logger are switched on and settings.
3. The electrical heating switch on and settings on the required heat flux.
4. To establish a steady state condition, the apparatus is left between (30-60 minutes) depending on the Reynold number being used, a Reynold number (Re=1500) took shorter time than other values to be steady state thermally.

The flow rate is checked by flowmeter continuously to be sure that the flow is constant. The thermocouples readings are recorded every second and saved in data logger by SD-RAM. Their readings are observed with time until become constant, at that time Data-logger is switch off and SD-RAM is drowned. The Measurements Procedure during each test run, the following readings are recorded:

1. The air surrounding temperature is measured by thermocouple K-type and recording in data-logger.
2. Recording the new value of pressure difference.
3. The water volumetric flow rate through the test section that equivalent to desired Reynold number is measured by using a flow meter.
4. Recording the current of the heater in Ampere.
5. Recording the voltage of the heater in Volts.
6. The reading of all thermocouples from data logger after reaching to the steady-state

In this paper, an experimental study forced convection heat transfer for a laminar fully developed flow with constant heat flux in a horizontal circular pipe filled with porous media to show the experimental study to enhance heat transfer in a circular tube by using the porous medium, studying the effect of porous medium on the surface temperature of the circular tube, the local Nusselt number and the average Nusselt number and studying the effect of heat flux on heat transfer.

3. Experimental Data Analysis and Calculations

The experimental data recorded for the temperature and velocity are used for the following heat transfer analysis a simplified step is used to analyze the heat transfer process for the fluid flow in the test section channel.

The average heat flux to the test section is obtained by the measurement of the average increase in the temperature of the distillate water across the test section in steady state. The average increase in temperature may be found by measuring the temperature at the entrance and the exit of the test section according to the thermocouples reading.

\[ Q_{\text{absorb}} = \dot{m} \times C_p \times (T_{\text{out}} - T_{\text{in}}) \]  \hspace{1cm} (1)

Where \( (T_{\text{in}}) \) and \( (T_{\text{out}}) \) are the temperatures of distillate water at the entrance and the exit, respectively.

The total input power supplied to the pipe can be calculated from

\[ Q_{\text{input}} = I \times V_0 \]  \hspace{1cm} (2)

Where \( (I, V_0) \) is the current and voltage of heater, respectively.

The heat balance between the heat flux supplied by eq. (2) and the heat flux calculated from eq. (1) showed that the difference between them does not exceed (18.842 \%) for different values of voltage employed, due to the good insulation so, the actual heat used in calculations is

\[ Q_{\text{actual}} = Q_{\text{input}} - Q_{\text{Loss}} \]  \hspace{1cm} (3)

The wall heat flux can be calculated by:

\[ q'' = \frac{Q_{\text{actual}}}{A_s} \]  \hspace{1cm} (4)
Where, \( A_s \) the surface area of the desired pipe.

we can be obtained the local heat transfer coefficient by:

\[
h_x = \frac{q''}{\Delta T_x} = \frac{q''}{(T_s - T_b)} \quad \cdots \cdots (5)\]

Where \( \Delta T_x \), is the temperature difference between the local surface temperature \( T_s \) and the local bulk fluid temperature \( T_b \), at length \( x \) from the test channel entrance. The local bulk fluid temperature \( T_b \) at each measuring level is calculated by assuming a linear water temperature variation along with the flow in the channel,

\[
T_b = T_{in} + \frac{x}{L} (T_{out} - T_{in}) \quad \cdots \cdots (6)
\]

Consequently, the local Nusselt number \( (Nu_x) \) can be determined as:

\[
Nu_x = \frac{h_x \times D_{pipe}}{\kappa_{eq}} \quad \cdots \cdots (7)
\]

In addition, the average values of the Nusselt number \( (Nu_{ave}) \) can be calculated as:

\[
Nu_{ave} = \frac{1}{L} \int_0^L Nu_x \, dx \quad \cdots \cdots (8)
\]

The fluid velocity at the inlet of the test channel calculated by:

\[
v = \frac{\dot{V}}{A} \quad \cdots \cdots (9)
\]

Where \( \dot{V} \) is volume flow rate of water.

The Reynolds number is calculated according to:

\[
Re = \frac{\rho_f \, \nu_f \, D_{pipe}}{\mu_f} \quad \cdots \cdots (10)
\]

Porosity can be calculated from the formula below

\[
\varepsilon = \frac{V_{voids}}{V_{total}} = \frac{V_{total} - V_{soilds}}{V_{total}} \quad \cdots \cdots (11)
\]

where
\[
V_{soilds} = \text{number of particals} \times V_{\text{partical}} \tag{12}
\]

To calculated the porous medium thermal conductivity, can be use the formula below [2].

\[
k_{eq} = \varepsilon k_f + (1 - \varepsilon)k_p \tag{13}
\]

The tests have been performed for range of Reynold numbers (500, 1000 and 1500) with variable heat flux (200, 400 and 600 W) for each case and three times for each number. In each test the temperature change of the pipe surface is drawn with the length of the pipe, local Nusselt number with pipe length and average Nusselt number with Reynold number that will be illustrated in the results and discussion.

4. Result and dissuasion

Figure (3) through (5) shows the distribution of surface temperature of the pipe with the pipe length for porous steel media and at Reynold number (500, 1000 and 1500) respectively. For different values of heat flux, it has observed from the curves that the temperature increases as the length of the pipe increases, the highest value for temperature is at the highest heat flux and the lowest Reynold number and its value is (48 °C) because of increasing the velocity of flow inside the pipe leads to a decrease in the time required to transfer heat further to the liquid, which leads to a significant reduction in this quantity. The following figures, which are (6, 7 and 8), The same previous relationship is shown, but for the mixture porous medium (steel and plastic) and at the range of Reynold (500 - 1500) and for several values of heat flux, the highest value of surface temperature of the pipe is at the highest heat flux, and less Reynold and its value is (48.4 °C). While the figures (9) through (14) shows the values of the distribution of local Nusselt number with the length of the pipe for the steel and mixture (steel and plastic) porous media respectively, and at Reynold number range of (500-1500) constant for each filling media and for several values of the heat flux, it is noticed through these curves that the local Nusselt number decreases by increasing the length of the pipe in the axial direction to the middle of the pipe and then stabilizing in a straight line. The reason for this is due to the thermal resistance resulting from the thinnest thickness of the thermal boundary layers, and on the contrary, begins to decrease due to the increase in thermal resistance until it is almost fixed in the region of thermally fully developed. Figures (15 and 16) shows the average Nusselt number with the change of Reynold's number for both porous media. It is noticed that the average Nusselt number increases in a linear manner for the
steel porous media and is non-linear for the mixed porous media with the increase in the number of Reynold and its value for the mixed porous medium is greater than the steel porous medium.

5. Conclusions

1) The surface temperature distribution increases non-linearly with respect to the steel porous media, and its highest value is at Reynold number 500 and heat flux 600 W is (48 °C). As for mixture porous media, its highest value at Reynold number 500 and heat flux 600 W also and its value (48.4 °C).
2) The local surface temperature decreases as the Reynolds number increases at constant heat flux.
3) The local Nusselt number decreases as the axial position (the length of the pipe) increase at constant heat flux.
4) For the constant heat flux, the average Nusselt number increases as the Reynolds number increases.
5) The mixture of (steel and plastic) porous media was better than steel porous media.

6. Reference

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Fig (1) Photograph of the experimental setup of the apparatus

Fig (2) Schematic diagram of the experimental apparatus

Fig (3) Schematic diagram of the experimental apparatus
Fig (4) The relation between the pipe surface temperature and its length for many values of heat flux at Re = 500 and the pipe with a Steel porous medium.

Fig (5) The relation between the pipe surface temperature and its length for many values of heat flux at Re = 1000 and the pipe with a Steel porous medium.

Fig (6) The relation between the pipe surface temperature and its length for many values of heat flux at Re = 1500 and the pipe with a Steel porous medium.
Fig (7) The relation between the pipe surface temperature and its length for many values of heat flux at Re = 500 and the pipe with a Steel and plastic porous medium.

Fig (8) The relation between the pipe surface temperature and its length for many values of heat flux at Re = 1000 and the pipe with a Steel and plastic porous medium.

Fig (9) The relation between the pipe surface temperature and its length for many values of heat flux at Re = 1500 and the pipe with a Steel and plastic porous medium.
Fig (10) The relation between the Local Nusselt number and the pipe length for many values of heat flux at $Re= 500$ and the pipe with a Steel porous medium.

Fig (11) The relation between the Local Nusselt number and the pipe length for many values of heat flux at $Re= 1000$ and the pipe with a Steel porous medium.

Fig (12) The relation between the Local Nusselt number and the pipe length for many values of heat flux at $Re= 1500$ and the pipe with a Steel porous medium.
Fig (13) The relation between the Local Nusselt number and the pipe length for many values of heat flux at Re= 500 and the pipe with a Steel and plastic porous medium

Fig (14) The relation between the Local Nusselt number and the pipe length for many values of heat flux at Re= 1000 and the pipe with a Steel and plastic porous medium

Fig (15) The relation between the Local Nusselt number and the pipe length for many values of heat flux at Re= 1500 and the pipe with a Steel and plastic porous medium
Fig (16): The relation between the average Nusselt number and Reynolds number for many values of heat flux and the pipe with a Steel porous medium.

Fig (17): The relation between the average Nusselt number and Reynolds number for many values of heat flux and the pipe with a Steel and plastic porous medium.

Fig (18): Calibration of thermocouple

\[ y = 1.0153x + 0.8061 \]
Table (1) Distribution of thermocouples along the tube surface.

| No. | X (mm) |
|-----|--------|
| 1   | 33     |
| 2   | 63     |
| 3   | 103    |
| 4   | 163    |
| 5   | 243    |
| 6   | 333    |
| 7   | 443    |
| 8   | 583    |
| 9   | 783    |
| 10  | 958    |