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Experimental Study of Forced Heat Convection through a Vertical Porous Annuli

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ABSTRACT: In this study, an experimental work of forced heat convection has been conducted for water flowing through a vertical porous annuli. Three various sizes of glass ball porous media and porosities are utilized. The outer cylinder is made of Teflon (TPFE) while the inner cylinder is made of polished copper and it is electrically heated from inner with four various input power levels. Wide range of water flow rate in term of Reynolds number are utilized. The influence of porous media particles size, porosity, Reynolds number and wall heat flux on the heat characteristics and performance of forced convection have been tested and analyzed experimentally. The experimental results appear that the Nusselt number increases with increasing of ball sizes and decreasing porosity. Also, the forced heat convection performance of vertical porous annuli in term of Nusselt number is about three to three and half times more than that annuli without porous media for constant porosity.

KEYWORDS: Forced convection; experimental study; annuli; porous media; vertical; water.

INTRODUCTION

Forced heat convection through a porous media has been the topic of focused studies and researches in recent years because it's very important engineering and industrial applications to enhance the heat transfer and heat performance. These applications including transpire cooling, heat pipes, geothermal energy, tribology and lubrication, nuclear reactors, coolants flow through radioactive pellets, melting solidification, storage of absorbed solar energy, mold and core-sand formation, filtering, drying systems, trickle bed reactors, water purification systems, packed bed chromatography, fuel cells, oil and gas flow through reservoirs, oil production and natural gas production.

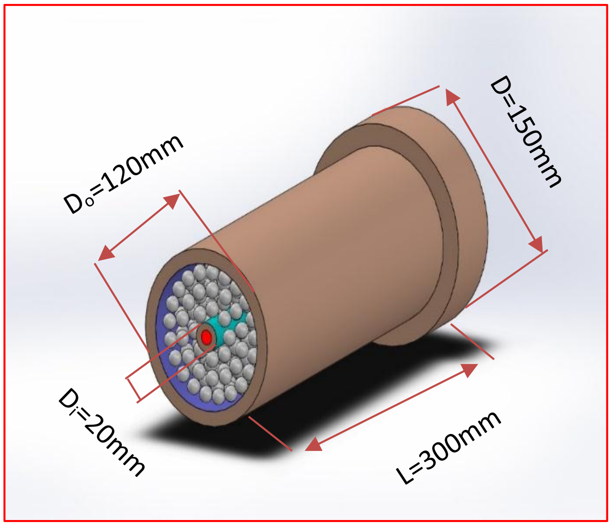
Bu-Xuan and Jian-Hua [1] studied experimentally the heat transfer by forced convection using water or oil as a fluid working into a vertical annular tube filled with different porous material. They used a packed bed with diameter of 0.7 mm and glass beads with diameter of 2.5 mm. They concluded the thermal dispersion from turbulence due to the presence of the porous medium has a significant impact on the values of Nusselt number in the fully developed region. Atwan et al. [2] presented an experimental investigation of flow and heat characteristics into a horizontal tube filled with a porous medium. They studied the effect of different parameters such as pipe radius ratio, flow velocity and thermal conductivity of the porous material. They used different materials of porous media namely, carbon steel and polyvinyl chloride and ranging of Reynold number between of (5200-12000). They showed that the presence of porous material increases the mean number of Nusselt fourfold compared to the absence of porous media, the average number of Nusselt increases with increasing radius. Hesham et al. [3] investigated numerically and experimentally the forced convection heat transfer inside a heated horizontal cylinder filled with a porous medium. They used copper cylinder with a length of 1 m and diameter of 38 mm and water as a fluid working and a carbon steel ball with a diameter of 6.35 mm as a porous material inserted into the cylinder. They used a wide range of Reynolds number between (400 - 2000). They noted that the Nusselt number and coefficient of heat transfer increases with increasing the Reynolds number. Suhad [4] studied numerically and experimentally the convection heat transfer in an inclined circular porous copper tube. He utilized a polypropylene ball as a porous media with different diameters like 2.88, 3.85, 4.73, and 7.85 mm and Reynolds number ranges from 55 to 130. He observed that the Nusselt number increases as the particle size increasing. Adnan et al. [5] studied experimentally and numerically the forced heat convection through a vertical annular tube filled with a porous medium. They used different radius of inner cylinder like 0.012, 0.02, and 0.026 m and constant outer cylinder radius of 0.120 m. They used ANSYS-Fluent software to solve the governing equations of numerical part.

They obtained a correlation equations to predict a Nusselt number as a function of Rayleigh number and aspect ratio. Omer et al. [6] introduced an experimental and numerical investigation of free convection heat transfer between two concentric cylinders filled with porous material. They used two types of porous media, glass balls (11 mm diameter) and iron balls (3.28 mm diameter). They showed that the internal cylinder's ability to dissipate heat is a function of Rayleigh number and the Nusselt number increases with Rayleigh number increasing. Manal and Ahmed [7] studied numerically and experimentally the free convective heat transfer into annular cylinder filled with a porous medium. They used three different with diameters of 100, 82 and 70 mm and an internal cylinder with constant diameter of 27 mm with and without ring fins. They showed that the number of Nusselt increases with increasing Rayleigh number and is also affected by the size of glass beads. Also, they noted that the Nusselt number increases as aspect radius ratio increasing and increasing in the length and number of fins lead to an enhancement in heat performance. Abhilash and Asok [8] presented an experimental and numerical study for the effect of mineral porous materials in tube at a temperature transfer rate in non-Darcy flow conditions with different velocities and porosities. They showed that the heat transfer from inner tube surface to water increases with increasing Reynolds number and the heat transfer rate decreases with increasing porosity. Mohammed et al. [9] introduced an experimental investigation of mixed convection heat transfer into a vertical annular tube filled with a porous medium and a water as fluid working. They used three different diameters of porous material like 24.36, 19.48, and 14.62 mm. They showed that the heat transfer coefficient increases as Reynolds number and surface heat flux increasing. Saad [10] presented an experimental investigation of natural-convective heat transfer through a concentric annular tube in a horizontal position with the influence of the porous medium. He used glass and plastic balls with different sizes. He utilized four different aspect ratios of annuli. He showed that the Nusselt number increases with increasing the aspect ratio annular tube and diameter of balls. Also, he noted that the glass porous media gave values of Nusselt number larger about 26% than that the plastic porous media for the same diameters. Isbeyeh et al. [11] presented an experimental study of forced heat convection through a heat exchanger filled with a porous media to enhance the heat transfer process. They used two concentric cylinders with a constant dimensions and alumina balls with diameter of 2.5 mm as a porous medium (d=2.5mm). They observed that the effectiveness of the two cylinders and internal and external cylinders is higher about 39%, 34% and 25% respectively of the case without porous medium but the increase of porosity leads to an increase in the efficiency of heat transfer of the two cylinders. Shahram et al. [12] presented a study of single-phase - air flow in a circular cross-sectional channel with different porous media. They noted that the presence of the porous medium in the transmission of a thermal flow on the walls of the channel to the fluid due to the creation of a uniform area and high conductivity of the porous material. Also, they found that the heat transfer rates clearly larger compared than that cases without porous media. Siva and Venkatesh [13] investigated numerically and experimentally the heat transfer of flow in a tube partially filled with porous media. They used a finite element with the splitting scheme on the properties. They determined that the porous entry of the internal shape symbolizes better rate of heat transfer compared with the annular shape, also the increase in heat transfer to the porous area is 30% more than heart-shaped insertion compared than annular shape. Mehmet [14] presented a numerical study of heat transfer in single phase flow with a porous medium using a COMSOL software to predict of the heat characteristics of pipe with porous media steel balls. They used a constant porous medium diameter of 3 mm placed in the pipe with inner diameter of 51.4 mm and Reynolds number ranges from 150 to 500. They found the Nusselt numbers in the longitudinal direction of pipe and studied the flow rate.

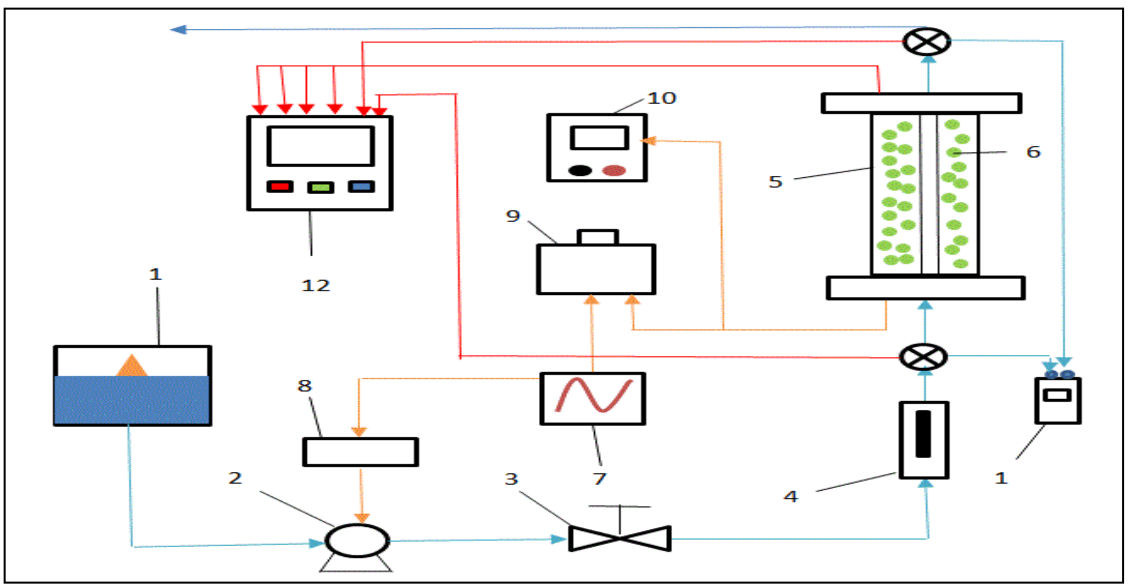
The objective of present study is to study and analyze of force heat convection into a vertical porous media annuli utilizing water as a working fluid and glass balls as porous media. It is focused on the effect of multi parameters namely, glass balls diameter, porosity, Reynolds number, wall heat flux and surface temperatures along the annulus on the heat characteristics and performance.

EXPERIMENTAL WORK

The experimental test-rig with accessories shown in Figure (1) Is specially fabricated to cover the tests of present study. It consists of test section, test loop and measurement tools.



a. Photograph of experimental test-rig b. Test section



c. Schematic diagram of experimental test-rig

1. Water tank 2. Pump tank 3. Valve 4. Flow rate water 5. Test section 6. Porous media 7. AC power supply 8. DC power supply 9. Voltage regulator 10. Digital multi-meter 11. Manometer 12. Temperature recorder

Figure 1. Photograph and Schematic diagram of experimental test-rig

The test section is consists of a concentric vertical annuli, glass balls, covers, heating element and thermocouples. The vertical annuli is consists of two concentric cylinders, outer and inner cylinders. The outer cylinder is made of Teflon as a good insulation and high resistance of corrosion with length of 300 mm, inner diameter of D_o=100 mm and thickness of 10 mm, it's drilled from outer surface with six longitudinal holes with diameter of 3 mm and equally pitches to fix the thermocouples. The ends of outer cylinder are screwed to put the Teflon covers. The inner cylinder is made of copper with a length of 400 mm (the addition length 100 mm to be supported with a Teflon covers), the outer diameter D_i= 20 mm and the thickness of 5 mm. The thermocouples are installed on the outer surface of the inner cylinder with same approach and distances of outer cylinder. Heating element are used and inserted inside the inner cylinder. The two ends of vertical annuli are covered using two Teflon covers with diameter of 150 mm and big thickness of 50 mm to minimize heat losses, to fix the two cylinders of annuli in concentric position and to allow the water passing through the annuli. The glass balls are inserted into space between the two cylinders of annuli, three sizes are used 11, 16 and 25 mm. Type-K thermocouples are used in all tests.

The test loop consists of a water tank and water piped out to go to the centrifuge pump, when running the pump, the water goes through the tubes to the valve, which controls the amount of water going out to the water flow meter, the flow meter is used to control the flow of water and several rates are used for different volumetric flow ranging from 0.3 l/m to 1.9 l/m, this is consistent with Reynolds number values ranging from 100 to 600. Then the water comes out from pipes into two branches, a pipe that enters to the test section and another pipe go to a pressure gauge to measure the pressure at the inlet. The cold water entering the test section is heated, then the hot water exit to two branches, one to the system and another to the pressure gauge to measure the outward pressure.

The measurement instruments are used like AC power supply, DC power supply, voltage regulator, digital multi-meter, and manometer and temperature recorder.

EXPERIMENTAL PROCEDURE AND CALCULATIONS

Multi tests are performed to study the effect of porous media on the heat performance of vertical annuli. Different parameters namely, glass ball size (11, 16, 25 mm), porosity (0.65, 0.75 and 0.85), water flow rates in term of Reynolds number ranging from 100 to 600 and wall heat flux ranging from 1.7 to 20.2 kW/m^2 are studied and analyzed.

The main steps of experimental work after assembly the test rig are:

- The water tank is fully filled with water.

- After running the pump, the water flow is adjusted to the required value using a valve to control the water quantity.

- A power supply unit is turned on to supply the electrical heater, when the required heat flow is needed to the inner surface of the internal cylinder of annuli by adjusting the input voltage using the power supply DC.

- Every 10 minutes the temperature is recorded when the steady-state is reached, when the temperature is observed at fixed value. The time required to reach the steady-state is about 30 minutes.

- The input and outlet water temperatures of the annuli, water flow rate, current and voltage, pressure drop water and surface temperatures are measured.

- Repeat all steps to cover all water flow rates, heat fluxes, glass balls sizes and porosities.

The power input (Q_{in}) to the electrical heater into inner cylinder of vertical porous annuli is calculated as follows:

$$Q_{in} = I\,V \tag{1}$$

It's converted to heat energy and transferred into vertical annuli by free convection (Q_{cn}), thermal radiation (Q_{rd}) and thermal conduction (Q_{cd}), then [15]:

$$Q_{in} = Q_{cn} + Q_{cd} + Q_{rd} \tag{2}$$

The loss by thermal radiation (Q_{rad}) occurs between the outer surface of inner cylinder and inner surface of external cylinder of vertical annuli [10, 16]:

$$Q_{rd} = \frac{\sigma\, A_i\ (T_i^4 - T_o^4)}{\frac{1}{\varepsilon_i} + \frac{A_i}{A_o}\left(\frac{1}{\varepsilon_o} - 1\right)} \tag{3}$$

The emissivity of Teflon (PTFE) is (ε_o= 0.85) and of polished copper is (ε_i= 0.05) [16], therefore the loss of thermal radiation is found small. Thermal losses by conduction (Q_{cd}) are very small because of using a good insulators from ends of annuli Teflon and can be neglected. Also, the rate of heat transfer by forced convection (Q_{cn}) through porous media can be calculated from the Newton's equation for cooling [15, 16]:

$$Q_{cd} = h_{av}\, A_i\, (T_{iav} - T_{oav}) \qquad (4)$$

Then, the average heat transfer coefficient (h_{av}) is calculated as follows:

$$h_{av} = \frac{I\, V}{A_i\, (T_{iav} - T_{oav})} \qquad (5)$$

Where,

I and V are the current intensity and voltage respectively.

T_{oav} and T_{iav} are the average wall temperatures of outer and inner cylinders of annuli respectively, evaluated as:

$$T_{oav} = \sum_{i=1}^{n} \frac{T_o}{n} \qquad (6)$$

And,

$$T_{iav} = \sum_{i=1}^{n} \frac{T_i}{n} \qquad (7)$$

A_o and A_i are the inner surface area of external cylinder and the outer surface area of inner cylinder of annuli respectively, they are calculated as follows:

$$A_o = \pi\, D_o\, L \qquad (8)$$

$$A_i = \pi\, D_i\, L \qquad (9)$$

The average Nusselt number (Nu_{av}) based on the hydraulic diameter (D_h) can be computed as [10, 16]:

$$Nu_{av} = \frac{h_{av}\, D_h}{K_{eff}} \qquad (10)$$

Where,

$$D_h = (D_o - D_i) \qquad (11)$$

L is the annuli length.

K_{eff} is the effective thermal conductivity of porous medium, and defined as a weighted arithmetic average of thermal conductivities of water and glass balls, it is calculated as follows [10, 16]:

$$k_{eff} = \Phi\, k_w + (1 - \Phi)\, k_g \qquad (12)$$

K_w is thermal conductivity of water.

K_g is thermal conductivity of glass spheres.

Φ is the porosity of porous medium, and computed as follows [10]:

$$\Phi = \frac{V_b - V_{balls}}{V_b} \qquad (13)$$

V_b and V_{balls} are bulk and glass balls volumes respectively, they are calculated as follows:

$$V_b = \frac{\pi}{4}\,(D_o - D_i)\,L \tag{14}$$

$$V_{balls} = \frac{4}{3}\,\pi\, r^3\, n \tag{15}$$

r and *n* are the glass ball radius and glass balls number respectively.

Define Reynolds number (*Re*) based on the water mass flow rate as [16]:

$$Re = \frac{4\dot{m}}{\nu} \tag{16}$$

The mass flow rate ($\dot{m}$) of water can be calculated as [16]:

$$\dot{m} = \rho_f\, Q \tag{17}$$

The water properties are taken at mean temperature (T_m), and calculated as follows:

$$T_m = \left(\frac{T_{iav} + T_{oav}}{2}\right) + 273 \tag{18}$$

Where,

Q is the flow rate.

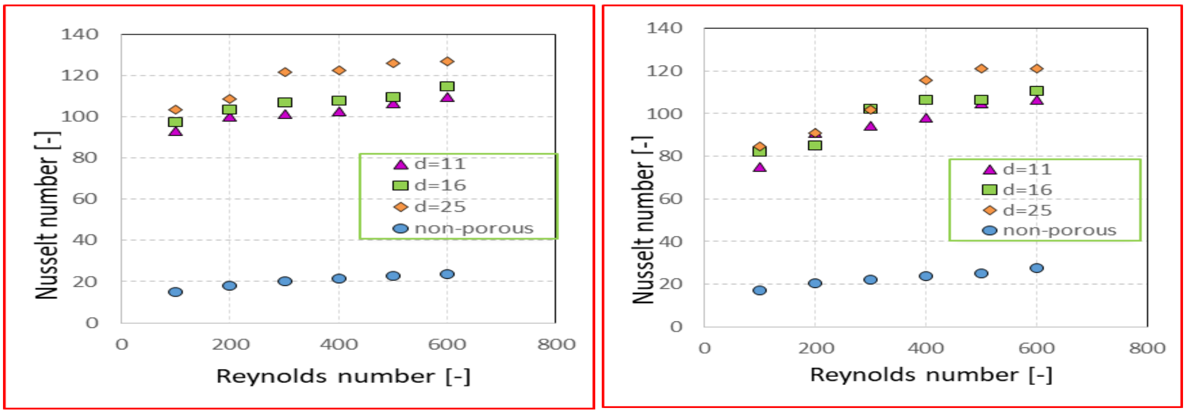
$\dot{m}$ is the mass flow rate of water.

RESULTS AND DISCUSSION

Different parameters are used for present study their effect on the thermal characteristics and performance of forced heat convection into vertical porous annuli. These parameters are diameter of glass balls, porosity, and water flow rate in term of Reynolds number and wall heat flux. The experimental results are plotted and analyzed.

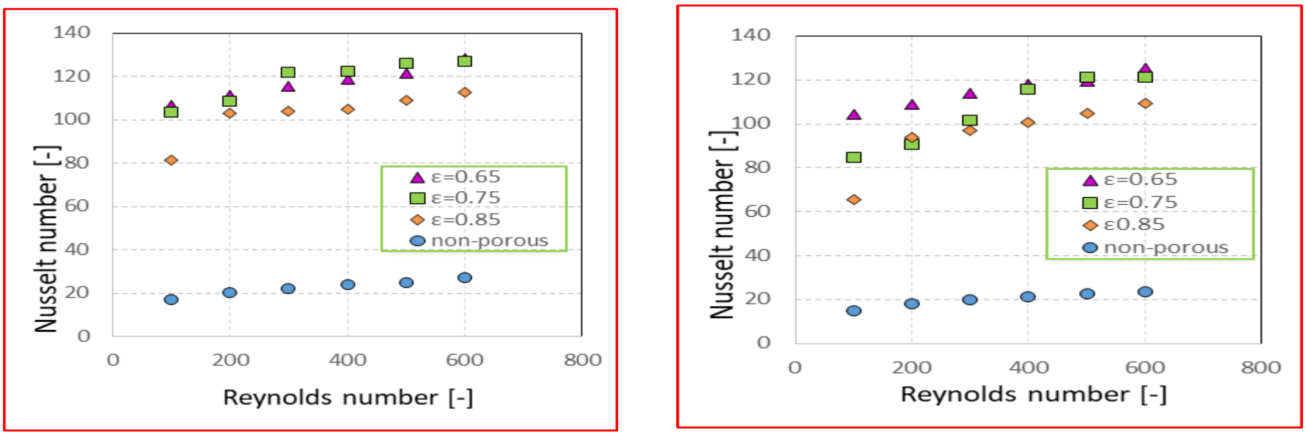
Figure (2) Shows the behavior of Nusselt number against Reynolds number for three different glass ball diameter (d= 11, 16 and 25 mm) and for annuli without porous media at wall heat fluxes 20.2 and 1.7 kW/m2 respectively). It is noted that the Nusselt number gradually increases with the increase of Reynolds number for all cases. It is because of increasing the currents of forced convection into a space of vertical annuli and leads to increasing of rate of heat transfer. It is also observed that the Nusselt number increases with increasing of the diameter of glass balls as well as for annuli without porous media a little change in values of Nusselt number. The best heat performance and heat transfer at case of glass balls diameter (d=25 mm) compared.

Figure (3) Illustrates the behavior of Nusselt number versus Reynolds number for three porosities (Φ= 0.65, 0.75 and 0.85) at constant glass ball diameter (d= 25 mm) and compared than case of annuli without porous medium at surface heat fluxes (q=1.7 kW/m2 and 20 kW/m2). It is showed that the Nusselt number increases as decreasing porosity about 4.25, 4.2 and 3.5 times for porosities 0.65, 0.75 and 0.85 respectively compared than annuli without porous media at constant glass ball diameter. It's because of increasing in viscous friction due to the decrease of void space in the porous medium and also because the high contact of glass balls surfaces, and this leads to high conductivity of porous medium [17-20].



a. heat flux (q= 20.2 kW/m2) b. heat flux (q= 1.7 kW/m2)

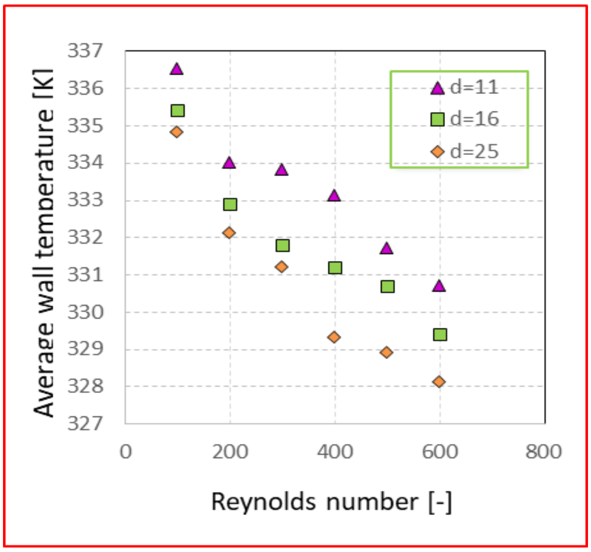
Figure 2. Nusselt number versus Reynolds number for different glass balls diameters at same porosity.

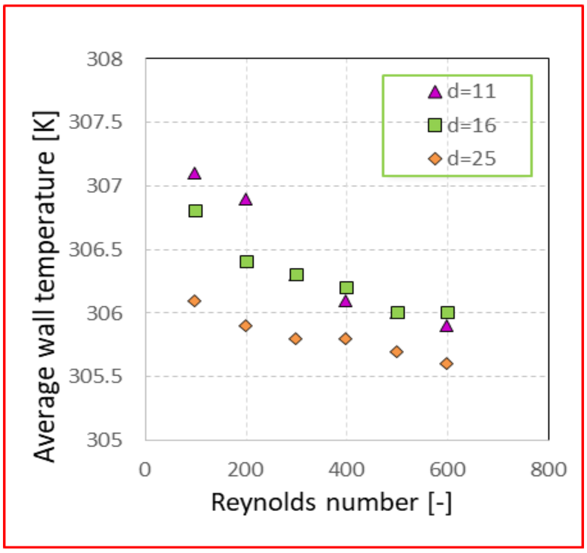


a. heat flux (q= 20.2 kW/m2) b. heat flux (q= 1.7 kW/m2)

Figure 3. Nusselt number versus Reynolds number for different porosities at ball diameter (d= 25 mm).

Figure (4) Appears the average temperatures of outer surface for the inner cylinder of annuli versus Reynolds number for different glass ball diameters at wall heat fluxes (q= 1.7 and 20.2 kW/m2). It is observed that the wall temperatures gradually decrease with Reynolds increases. It is also observed that the wall temperatures decrease with increasing diameter of glass balls, and the highest values of temperatures at the minimum glass ball diameter (d=11 mm) compared with other particle diameters. It is because of decrease the void space between particles through annuli.





a. heat flux (q= 20.2 kW/m2) b. heat flux (q= 1.7 kW/m2)

Figure 4. Average wall temperature versus Reynolds number for different glass ball diameter**.**

CONCLUSIONS

Forced heat convection through a vertical annuli with glass balls as porous media and water as working fluid is studied and analyzed experimentally. The inner cylinder of annuli is electrically heated with constant wall heat flux. Three sizes of glass balls, three porosities and wide range of surface heat fluxes and Reynolds number are utilized. The important conclusions can be drawn as follows:

1. The Nusselt number increases as the glass ball diameter increases and reaches a maximum values for ball diameter (*d*= 25 mm) at same porosity.
2. The Nusselt number increases with decreasing porosity about 4.25, 4.2 and 3.5 times for porosities 0.65, 0.75 and 0.85 respectively than that annuli without porous media at same glass ball diameter.
3. Best heat performance in term of Nusselt number at glass ball diameter (*d*= 25 mm) and porosity (Φ= 0.65).
4. The forced heat convection performance of vertical porous annuli in term of Nusselt number is about 3 to 3.5 times larger than that vertical annuli without porous media for constant porosity.
5. The average surface temperatures decreases clearly with increasing Reynolds number and glass balls diameter.
6. The pressure drop increases gradually as increasing the Reynolds number.

Nomenclature

A_i	outer surface area of inner cylinder, (m^2)
A_o	inner surface area of outer cylinder, (m^2)
A_s	surface area of heat transfer, (m^2)
d	diameter of glass ball, (m)
D_h	hydraulic diameter of annuli, (m)

D_i	outer diameter of inner cylinder, (m)
D_o	inner diameter of outer cylinder, (m)
h	forced heat convection coefficient, ($W/m^2.K$)
I	intensity current, (A)
k_a	thermal conductivity of glass balls, (W/m.K)
K_{eff}	effective of thermal conductivity, (W/m.K)
k_w	thermal conductivity of water, (W/m.K)
L	length of annuli, (m)
$\dot{m}$	water mass flow rate , (kg/s)
n	glass balls number
Nu	Nusselt number
Q	flow rate, (l/min)
Q_{cd}	heat conduction losses, (W)
Q_{cn}	rate of forced heat convection, (W)
Q_i	power input, (W)
Q_{rd}	heat radiation loss, (W)
r	adius of glass ball, (m)
Re	Reynolds number
T_i	average wall temperatures of inner cylinders for annuli, (^oC)
T_m	mean temperature, (K)
T_o	average wall temperatures of outer cylinders for annuli, (^oC)
V	voltage, (V)
V_b	bulk volume, (m^3)
V_{ball}	glass balls volume, (m^3)

Greek Letters

ε_i	emissivity of polished copper
ε_o	Emissivity of Teflon (PTFE)
v	kinematic viscosity, (m^2/s)
Φ	porosity
av	average

Subscript Symbols

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