Experimental Investigation of Natural Convection into a Horizontal Annular Tube with Porous Medium Effects

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ABSTRACT

In this work, an experimental investigation has been done for heat transfer by natural-convection through a horizontal concentric annulus with porous media effects. The porous structure in gap spacing consists of a glass balls and replaced by plastic (PVC) balls with different sizes. The outer surface of outer tube is isothermally cooled while the outer surface of inner tube is heated with constant heat flux condition. The inner tube is heated with different supplied electrical power levels. Four different radius ratios of annulus are used. The effects of porous media material, particles size and annulus radius ratio on heat dissipation in terms of average Nusselt number have been analyzed. The experimental results show that the average Nusselt number increases with increasing annulus radius ratio and particle diameter for same porous media material. Furthermore, two empirical correlations of average Nusselt number with average Rayleigh number for glass and PVC particles are developed. The present experimental results are compared with previously works and good correspondence is showed.

Key words: annular tube, porous medium, horizontal, balls, natural-convection.

تقصي عملي للحمل الطبيعي داخل أنبوب حلقي أفقي مع تأثير الوسط المسامي

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الخلاصــة

في هذا البحث أجري تقصي عملي لأنتقال الحرارة بالحمل الطبيعي خلال تجويف حلقي أفقي متحد المركز مع تأثير الوسط المسامي. يتكون الوسط المسامي في الفراغ البيني للتجويف الحلقي من كرات زجاجية وأستبدلت بكرات بلاستيكية وبأحجام مختلفة. تم تبريد السطح الخارجي للأنبوب الخارجي بثبوت درجة الحرارة بينما تم تسخين السطح الخارجي للأنبوب الداخلي بثبوت الفيض الحراري. تم تسخين الأنبوب الداخلي بعدة مستويات للقدرة الكهربائية المجهزة. أستخدمت أربع نسب مختلفة لنصف قطر التجويف الحلقي. تم تحليل تأثير مادة الوسط المسامي المستخدم وحجم (قطر) الكريات ونسبة نصف القطر للتجويف الحلقي على التبديد الحراري بدلالة معدل رقم نسلت. بينت النتائج العملية أن معدل رقم نسلت يزداد مع زيادة نسبة نصف القطر للأنبوب الحلقي وقطر الكريات لنفس مادة الوسط المسامي. أضافة الى أستنباط علاقتين أرتباطيتين عمليتين تحكم معدل رقم نسلت مع معدل رقم ريليه للكريات الزجاجية والبلاستيكية. تم مقارنة النتائج العملية للبحث الحالي مع بحوث سابقة وأظهرت توافقا جيدا.

الكلمات الرئيسية: أنبوب حلقي ، وسط مسامي ، أفقي ، كريات ، الحمل الطبيعي .

1. INTRODUCTION

The porous media is used generally to improve the heat transfer rate in thermal systems which work by heat convection. The convective heat transfer through a porous media has many thermal engineering and industrial applications including the systems of nuclear reactor cooling, nuclear waste disposal, engineering of thermal insulation, grain storage and drying, ground water flows and filtration processing, systems of water purification (RO), solar collectors, heat exchangers, electrochemical processes, oil recovery processes, extraction of the geothermal energy, thermal storage systems, regenerators systems, furnaces and ceramic processes, … etc.

Prasad, and **Kulacki**, **1985** studied numerically and experimentally the natural-convection through short concentric vertical cylinders filled with porous media for wide range of radius ratio and height to gap width ratio. They used glass beads and water as porous media. They obtained a good agreement between measured temperatures at different locations with predicted temperatures from numerical solution. **Atwan**, **et al.**, **2000** investigated experimentally the heat transfer and flow by forced-convection through a horizontal annulus filled with porous medium. They found that the larger values of convection heat transfer coefficient are obtained with packing particles of higher thermal conductivity and the Nusselt number increases with the ratio of inner to outer cylinders radii increasing. **Khanafer**, and **Chamkha**, **2003** presented a numerical study for mixed-convective heat transfer inside a horizontal annulus filled with a fluid saturated porous media by heating the inner cylinder and cooling the outer cylinder. They used the finite element method to solve the governing equations of problem. They discussed the influence of Darcy number on the isotherms, streamlines and heat transfer rate as well as the average Nusselt number at different annulus gap. They compared the numerical data with a previous works and the results show excellent agreement. **Shi**, **et al.**, **2006** presented a numerical solution using finite difference method (FDM) based lattice Boltzmann model to simulate the fluid flows and isotherms of free-convection heat transfer in a horizontally concentric annulus for wide range of Prandtl and Rayleigh numbers. They showed good stability of the numerical model and well agreement of presented results compared with the previous studies. **Al-Joboury**, **et al.**, 2009 studied theoretically and experimentally free heat convective between two concentric horizontal pipes filled with two types of materials, iron and glass beads as a porous medium under conditions of constant heat flux on the inner pipe surface and constant temperature on the outer pipe surface. They showed that the using of iron spheres porous media as high conductivity material canceled the heat transfer by convective through the gap of annulus. Also they found that the heat dissipation ability from the surface of inner pipe is a function of Rayleigh number. **Hussein**, **et al.**, **2009** investigated theoretically and experimentally natural-convection heat transfer through the annulus of two concentric vertical cylinders filled with gravel porous media under constant heat flux condition of inner cylinder. They solved that the governing equations by Fluent package and plotted the temperature profiles for three different radius ratio and different heat fluxes. They developed the empirical correlations between Nusselt number with Rayleigh number and Nusselt number with radius ratio of concentric cylinders. **Mahdi**, **et al.**, **2013** presented a review investigation for improvement of natural convective heat transfer using porous medium and nano-fluids. They showed that the convection heat transfer in terms of heat transfer coefficient increased with porous medium because the best thermal conductivity and thermal dissipation area of beads. **Ahamad**, **et al.**, **2014** presented a numerical study of natural heat convection inside vertical annular cylinders with porous medium effects. They visualized the fluid flow and convection heat transfer as streamlines and isotherms with influence of radius ratio and aspect ratio using finite element technique.

The purpose of the present work is to present an experimental study for natural convective heat transfer through a horizontal concentric annular tube filled with a porous media. It is focused on the influence of several parameters like packing material, particle (ball) diameter, radius ratio of the annular tube and wall heat flux on the characteristics of natural-convective such as average convection heat transfer coefficient, average Nusselt number and average Rayleigh number. The present work is carried out for two packing materials namely, glass and plastic (PVC) balls with air as a fluid working for different particle sizes and wide range of the annulus radius ratios.

2. EXPERIMENTAL WORK

2.1 Experimental Test Rig

The experimental test rig shown in **Fig. 1** is especially designed and manufactured for covering experiments of the present work. It consists of test-section, two V-wooden blocks, water inlet and outlet, voltage regulator with digital display, digital multi-meter, data logger thermometer, thermocouples and planar table.

The test section involves a horizontal annulus from two concentric tubes as shown in **Fig. 2**, the outer tube made of polished steel has fixed inner radius (r_o) of 38 mm and wall thickness of 2 mm while the inner tube made of polished aluminum has four different outer radius (r_i) of 6, 8, 10 and 12.5 mm to obtain a wide range of the annulus radius ratio (R) and wall thickness of 1 mm. Each tube has constant length (L) of 400 mm. The annulus gap is carefully filled with glass solid balls (thermal conductivity of 1.1 W/m.K) with diameter (d_p) of 12.5 mm and replaced by plastic PVC (polyvinyl-chloride) solid balls (thermal conductivity up to 0.16 W/m.K) with two particle diameters (d_p) namely, 6.25 and 12.5 mm. The inner tube of annulus is heated electrically from inside with different heat input levels using heating element with power up to 600 W and kept along the center of inner tube. The small gap between the heating element and inner surface of inner tube is filled by the sand to ensure natural-convection in the electrical heater. The voltage regulator with digital display type SAKO-TDGC$_2$ is used to govern the supplied voltage input to the electrical heater. Also a digital multi-meter type VICTOR-VC890C$^+$ is used to measure voltage and alternating current (AC) passing through the heater.

The ends of the concentric annulus are closed by two Teflon (polytetrafluoroethylene) covers (thermal conductivity up to 0.25 W/m.K) with large thickness up to 25 mm to minimize heat lost from annulus ends and to fix the two tubes of concentric annulus. One of these covers is drilled with small drill to pass the thermocouples wires and electrical heater cable and then sealed by thermal epoxy. The outer tube of annulus is isothermally cooled by circulation water system through a coiled copper pipe around the outer surface of outer tube and then the whole test-section is covered and insulated by thick layer about 50 mm of glass wool to minimize the heat dissipating to surrounding. The test-section assembly is horizontally put on two V-wooden blocks and planar table.

The outer surface temperature of inner tube and inner surface temperature of outer tube are measured using twelve type-K calibrated thermocouples; six to every tube are inserted longitudinally with angular displacement of 180^o between them at suitable locations. The thermocouples are joined with twelve-channels of data logger thermometer type BTM-4208SD. Another two same K-type calibrated thermocouples are oppositely inserted in the middle of outer surface for the outer tube to record the temperatures by digital multi-meter. Additional two same type-K calibrated thermocouples are used to read the temperatures difference through Teflon covers to estimate the thermal dissipation lost by conduction as illustrated in **Fig. 2**.

2.2 Experimental Procedure and Calculations

Twenty four cases are tested and studied according to the flow chart in **Fig. 3**. The cases includes two different wall heat fluxes (q= 800 and 1750 W/m^2), glass and plastic (PVC) porous media, one particle (ball) diameter (d_p= 12.5 mm) for glass porous media and two particle diameters (d_p= 6.25 and 12.5 mm) for plastic porous media and four radii ratios of annular tube (R= r_i /r_o) namely, 0.16 , 0.21 , 0.26 and 0.33. The readings of thermocouples have been recorded after (90-120) minutes and when the difference between two temperature readings within was 0.5 ^{o}C under steady-state condition, the supplied voltage and current to the heating element are recorded.

The electrical power input (Q_{in}) to the heating element inside inner tube of the annulus is:

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

It is transformed to thermal energy and transferred through the annulus gap by natural heat convection (Q_c), heat radiation (Q_r) in addition to heat conduction (Q_{cd}) from ends of horizontal annular tube. Hence,

$$Q_{in} = Q_c + Q_r + Q_{cd} \tag{2}$$

The radiation heat lost (Q_r) between outer surface of inner tube and inner surface of outer tube of the annulus is computed as **Lienhard**, **2008** and **Long**, and **Sayma**, **2009**:

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

It is found that (Q_r) is small about 4% to 7% of electrical heat input (Q_{in}) for all cases because of inner and outer tubes are made from low emissivity materials namely polished aluminum (ε_i= 0.05) and polished steel (ε_o= 0.1) respectively.

The thermal lost by conduction (Q_{cd}) is defined as thermal potential difference to thermal resistance. It is found very small (less than 2%) of heat input (Q_{in}) across ends of annular tube because it's well insulated using Teflon covers and can be neglected, then equation (2) can be written as follows:

$$Q_c = I\ V - Q_r \tag{4}$$

Also the heat transfer rate by free-convection (Q_c) to porous medium is calculated by Newton's equation of cooling, **Favre-Marient**, and **Tardu**, **2009**:

$$Q_c = h_{av}\, A_i (T_{iav} - T_{oav}) \tag{5}$$

Hence, the average convection heat transfer coefficient (h_{av}) is evaluated as:

$$h_{av} = \frac{I\ V - Q_r}{A_i (T_{iav} - T_{oav})} \tag{6}$$

where, T_{iav} and T_{oav} are the average surface temperatures of inner and outer tubes respectively, measured as:

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

and;

$$T_{oav} = \sum_{i=1}^{n} \frac{T_o}{n} \qquad (8)$$

A_i and A_o are the outer surface area of inner tube and the inner surface area of outer tube respectively, they are computed as follows:

$$A_i = 2\,\pi\, r_i\, L \qquad (9)$$

$$A_o = 2\,\pi\, r_o\, L \qquad (10)$$

The average Nusselt number (Nu_{av}), based on the annulus gap (δ) as characteristics length of geometry can be simply computed as, **Vafai**, **2005** and **Nield**, and **Bejan**, **2013**:

$$Nu_{av} = \frac{h_{av}\,\delta}{k_e} \qquad (11)$$

where,

$$\delta = 2(r_o - r_i) \qquad (12)$$

k_e is the effective (overall) thermal conductivity of the porous media and defined as a weighted arithmetic mean of thermal conductivities of fluid (air) and solid (balls) as **Vafai**, **2005** and **Nield**, and **Bejan**, **2013**:

$$k_e = \phi k_f + (1-\phi) k_s \qquad (13)$$

ϕ is the porosity of porous media and defined as ratio between volume of the void space (ϑ_v) to bulk volume (ϑ_b) then Vafai, **2005** and **Nield**, and **Bejan**, **2013**:

$$\phi = \frac{\vartheta_v}{\vartheta_b} = \frac{\vartheta_b - \vartheta_p}{\vartheta_b} \qquad (14)$$

The bulk volume (ϑ_b) is calculated as, **Vafai**, **2005** and **Nield**, and **Bejan**, **2013**:

$$\vartheta_b = \pi\, L\,(r_o^2 - r_i^2) \qquad (15)$$

and the particles volume (ϑ_p) is the product of particles (balls) number (N) by volume of one ball as:

$$\vartheta_p = N\left[\frac{4}{3}\pi\left(\frac{d_p}{2}\right)^3\right] \qquad (16)$$

The average Grashof number (Gr_{av}) is **Vafai**, **2005** and **Nield**, and **Bejan**, **2013**:

$$Gr_{av} = \frac{g\beta(T_{iav} - T_{oav})\delta^3}{\nu^2} \tag{17}$$

Define average Rayleigh number (Ra_{av}) as the product of the average Grashof number and Prandtl number, **Lienhard**, **2008**:

$$Ra_{av} = Gr_{av}\, Pr \tag{18}$$

where,

$$Pr = \frac{\nu}{\alpha} = \frac{C_p\, \mu}{k_e} \tag{19}$$

and,

$$\nu = \frac{\mu}{\rho} \tag{20}$$

By substituting Eqs. (17), (19) and (20) in Eq. (18), the average Rayleigh number based on the annulus gap (δ) can be written as follows:

$$Ra_{av} = \frac{g\beta C_p \rho_f \rho_s (T_{iav} - T_{oav})\delta^3}{\mu k_e} \tag{21}$$

$$\beta = \frac{1}{T_m} \tag{22}$$

All properties of fluid (air) inside a gap spacing of the annular tube are taken at mean temperature (T_m). It is calculated as the arithmetic mean between average surface temperature of the inner and outer tube along the annulus gap as follows:

$$T_m = \left(\frac{T_{iav} + T_{oav}}{2}\right) + 273.18 \tag{23}$$

3. RESULTS AND DISCUSSION

The present work investigates experimentally the effect of porous media material, particle diameter (d_p), radius ratio of the annular tube (R) and heat flux (q) on the performance and characteristics of natural-convective heat transfer through a horizontal concentric annular tube filled with porous medium.

Figs. 4 and **5** illustrate the behavior of average Nusselt number with variation of annulus radius ratio for two tested materials namely glass and PVC balls at two different heat fluxes. Two different particle diameters are used for PVC porous medium to study the influence of particle size on performance of natural-convection while it's constant for glass. The figures illustrate that average Nusselt number always increases by increasing the annulus radius ratio and heat flux. This increases because average Nusselt number depends directly on annulus gap and natural-convection heat transfer coefficient as function of heat input. **Fig. 4** clears that the highest values of average Nusselt number is linked with the use of glass porous media compared with plastic PVC for same particles size because the thermal conductivity of glass balls are higher seven times approximately than PVC particles and the thermal conductivity of balls effects directly on heat transfer rate. An increase in annulus radius ratio or a decrease in the annulus gap yields faster flow and giving higher and thinner of boundary layers and thus, higher rate of heat transfer is based on average Nusselt number. Also **Fig. 5** illustrates that the average Nusselt number increases with the increase of particle diameter for same material because increasing of the convection currents through annulus gap and consequently increasing in the heat transfer rate.

Figs. 6 and **7** show the variation of average Nusselt number with average Rayleigh number based on the annulus gap, heat flux and particle diameter for two different porous media materials. The average Rayleigh number increases always as the average Nusselt number, heat flux and particle diameter increases. Highest values of the average Rayleigh number with higher thermal conductivity of glass particles respect to the PVC packing medium is shown in **Fig. 6**. The high thermal conductivity of glass particles causes high contact conduction for particles dabbing the heated surface inside of the annulus. Hence, an increase in thermal conductivity of particle leads an increase in rate of heat transfer in terms of average Nusselt and Rayleigh numbers at same particle diameter.

Fig. 8 shows a comparison of average Nusselt number of present work with experimental results of Al-Joboury et al. work for glass particles. It shows a good similarity in the behavior, but the present work gives an average Nusselt number higher than that done in Al-Joboury et al. work because higher heat flux levels are used in the present work. Based on the experimental results showed in **Figs. 8** and **9**, two empirical correlations of average Nusselt number as a function of average Rayleigh number are developed for glass and PVC packing materials as follows:

$$Nu_{av} = 0.000002Ra_{av}^3 - 0.0005Ra_{av}^2 + 0.0546Ra_{av} + 0.7269 \tag{24}$$

which is valid for glass particle at: d_p= 12.5 mm , $20 \le Ra_{av} \le 175$ and $0.16 \le R \le 0.33$.

$$Nu_{av} = 0.000004Ra_{av}^3 - 0.0006Ra_{av}^2 + 0.044Ra_{av} + 1.0468 \tag{25}$$

which is valid for PVC particle at: 6.25 mm $\le d_p \le$ 12.5 mm , $15 \le Ra_{av} \le 135$ and $0.16 \le R \le 0.33$.

Figs. 10 and **11** illustrate a comparison of the correlated average Nusselt numbers using the correlation Eqs. (24) and (25) with the experiments of present work for glass and PVC packing materials. They are clear that most of experimental results are located within 10% and 6% of the mentioned correlation equations for glass and PVC particles respectively.

4. CONCLUSIONS

Steady state natural-convective heat transfer through a porous structure inside gap spacing of concentric annular tube in horizontal position is investigated experimentally. The inner tube is electrically heated under constant heat flux while the outer tube is isothermally cooled. Two types of porous media materials, two different wall heat fluxes and particle diameters with four radii ratios are used. The conclusions can be summarized as follows:

- The average Nusselt number increases as the particle diameter increases for same material of porous media.
- Average Nusselt number always increases with increasing radius ratio of annular tube and wall heat flux.

- The average Rayleigh number based on the annulus gap increases as the average Nusselt number increases.
- The glass porous media gave better natural-convective heat transfer performance in terms of average Nusselt number reaching to 26% higher compared with plastic (PVC) particles for same size (diameter).
- Two experimental correlations are developed to predict the average Nusselt number for annulus tube filled with glass and PVC porous medium.

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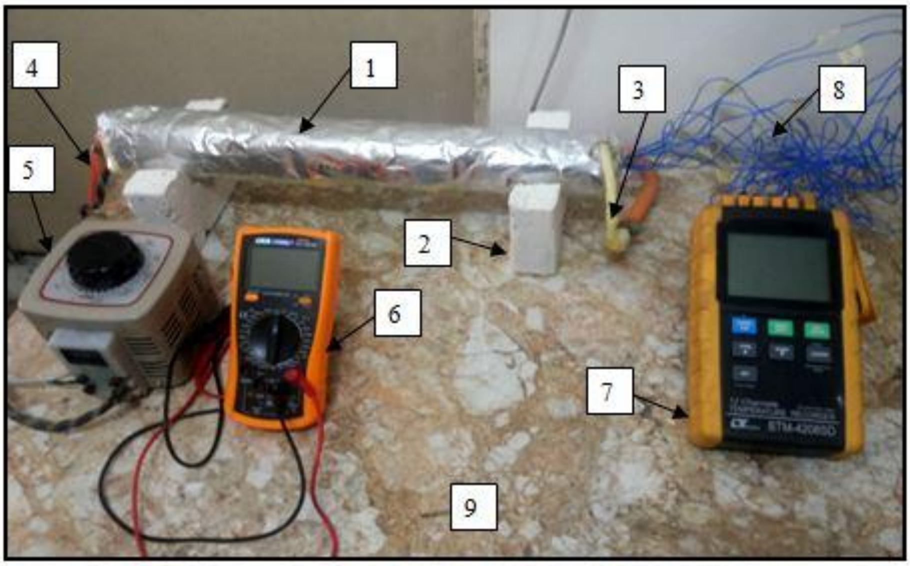
NOMENCLATURE

A= surface area of heat transfer, m^2.
C_p= specific heat at constant pressure, kJ/kg.K .
d_p= diameter of particle (ball), m.
g= gravitational acceleration, m/s^2.
Gr= Grashof number, dimensionless.
h= convection heat-transfer coefficient, $W/m^2.K$.
I= input current intensity, A.
k= thermal conductivity, W/m.K .
L= length of the annular tube, m.
N= number of particles (balls), dimensionless.
Nu= Nusselt number, dimensionless.
Pr= Prandtl number, dimensionless.
q= heat flux, W/m^2.
Q_c= convection heat transfer, W.
Q_{cd}= conduction heat lost, W.
Q_{in}= electrical power input, W.
Q_r= radiation heat transfer, W.
r= radius tube, m.
R= radius ratio ($R= r_i /r_o$), dimensionless.
Ra= Rayleigh number, dimensionless.
T= surface temperature, ^oC.
T_m= arithmetic mean of the surface temperatures, ^oC.
V= voltage supplied, V.
β= volumetric coefficient of thermal expansion, 1/K.
δ= annulus gap, m.
ε= emissivity of the surface, dimensionless.
ϑ_b= bulk volume, m^3.
ϑ_p= volume of particles (balls), m^3.
ϑ_v= volume of a void space, m^3.
μ= dynamic viscosity, kg/m.s.
ρ= density, kg/m^3.
σ= Stefan-Boltzmann constant, $\sigma = 5.67 \times 10^{-8}$ W/ $m^2.K^4$.
ϕ= porosity of media porous, dimensionless.
$\boldsymbol{\nu}$= kinematic viscosity, m^2/s.

Subscript symbols:

av= average.
e= effective.
f = fluid (air).
i= outer surface of inner tube for annulus.

o= inner surface of outer tube for annulus.
s= solid (balls).



1. Test-section 2. V-wooden block 3. Water flow inlet 4. Water flow outlet 5. Voltage regulator with digital display 6. Digital multi-meter 7. Data logger thermometer 8. Thermocouple wires 9. Planar table.

Figure 1. Photograph of the experimental test rig.

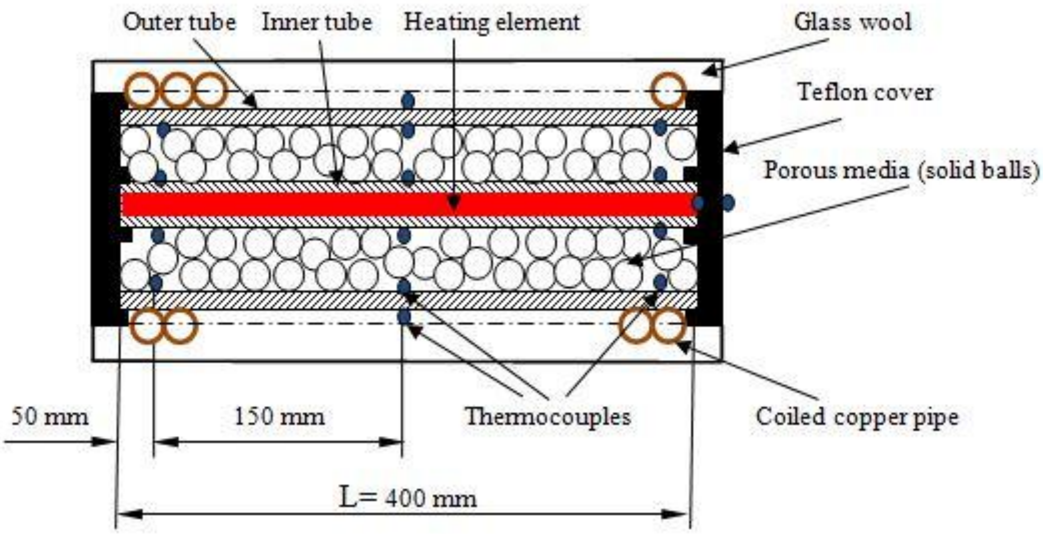


Figure 2. Longitudinal section of the test-section with locations of the thermocouples.

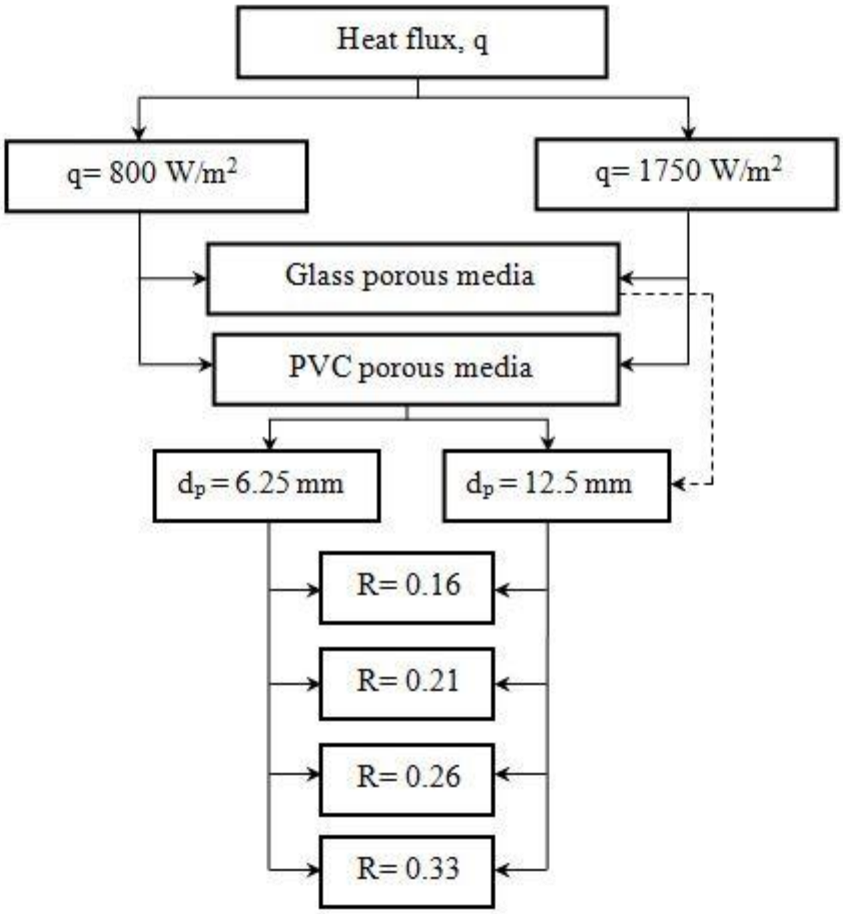


Figure 3. Flow chart of the tested cases.

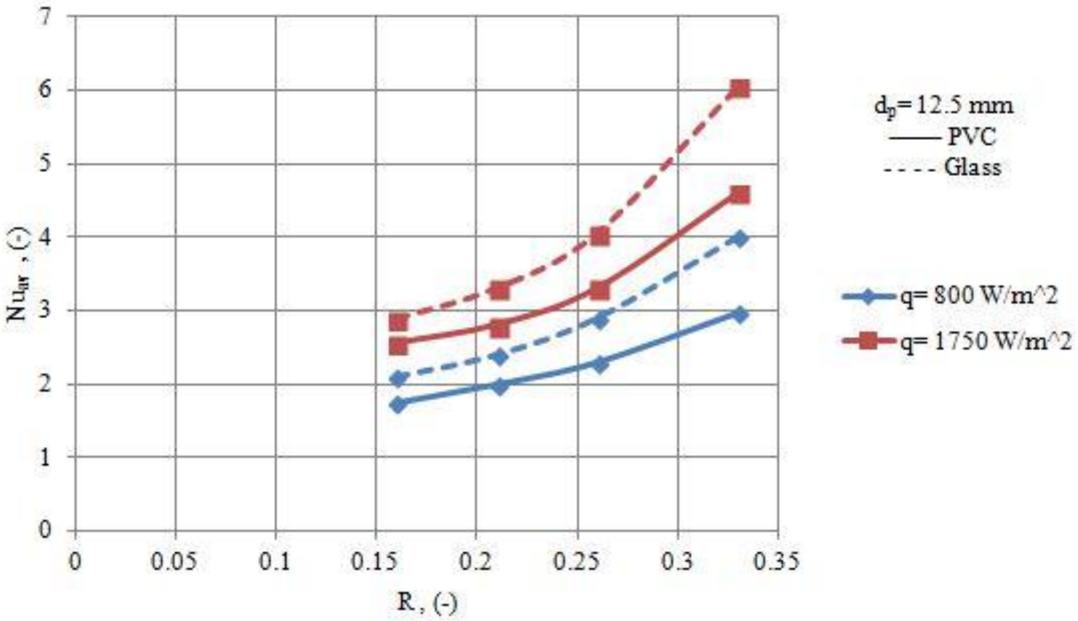


Figure 4. Average Nusselt number against radius ratio for PVC and glass porous medium with same particle diameters.

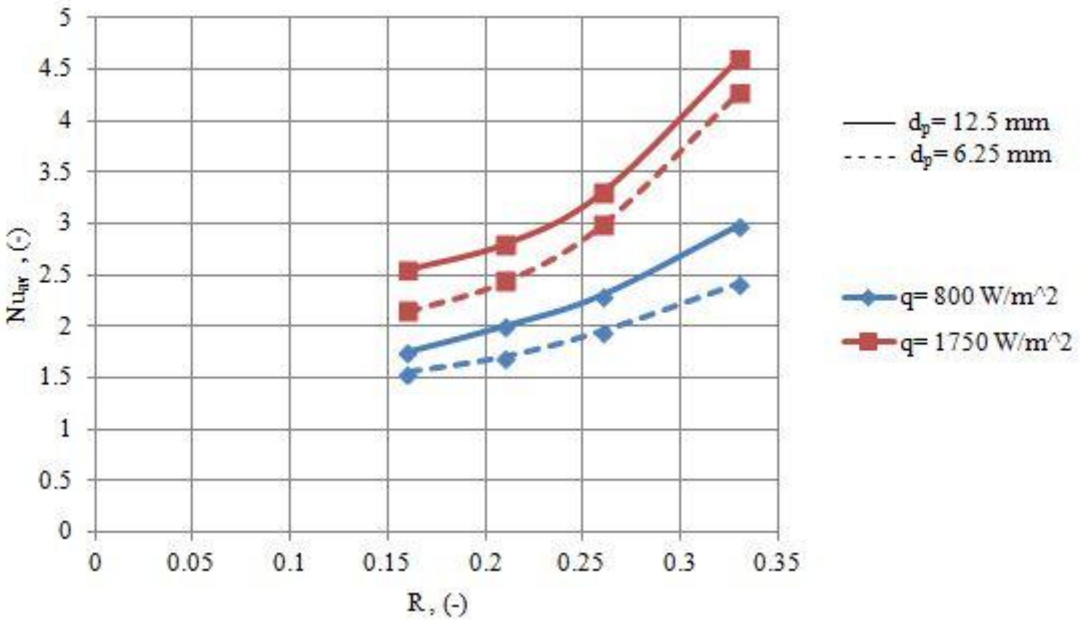


Figure 5. Average Nusselt number against radius ratio for PVC porous medium with different particle diameters.

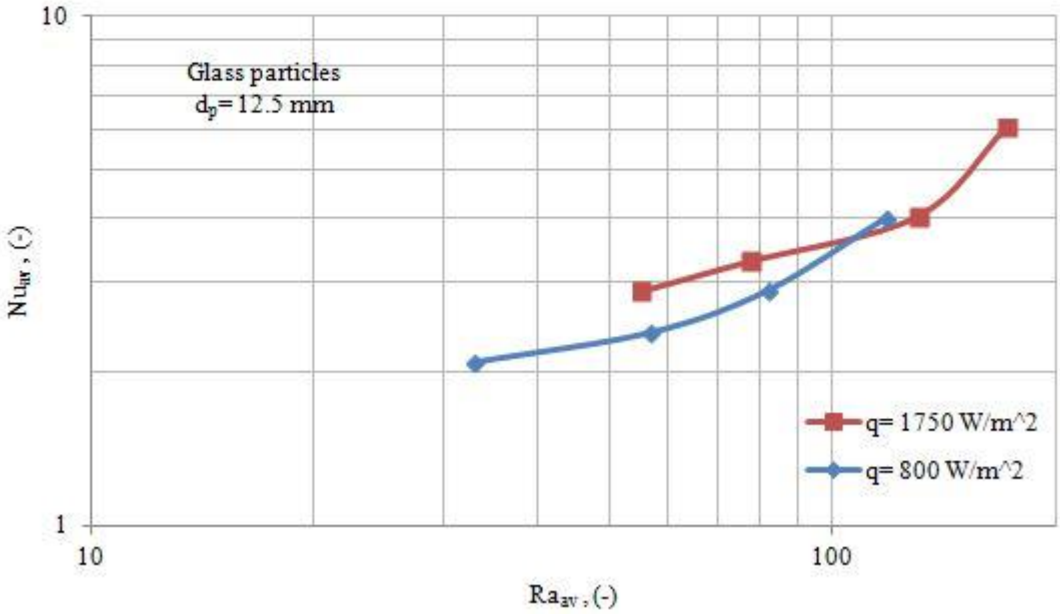


Figure 6. Average Nusselt number against average Rayleigh number for glass porous medium and different heat fluxes.

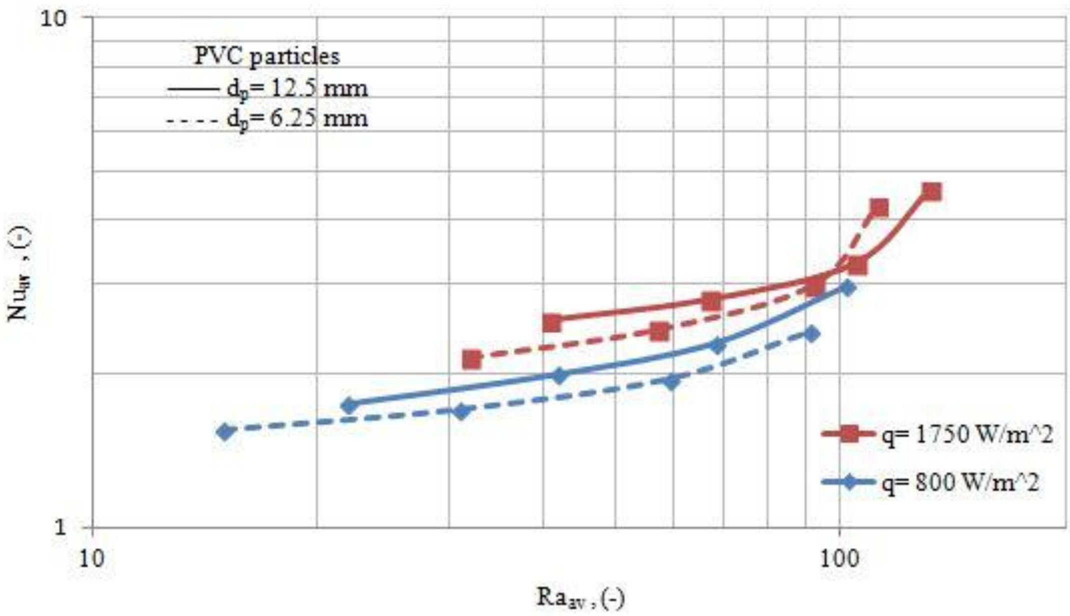


Figure 7. Average Nusselt number against average Rayleigh number for PVC porous medium with different particle diameters and wall heat fluxes.

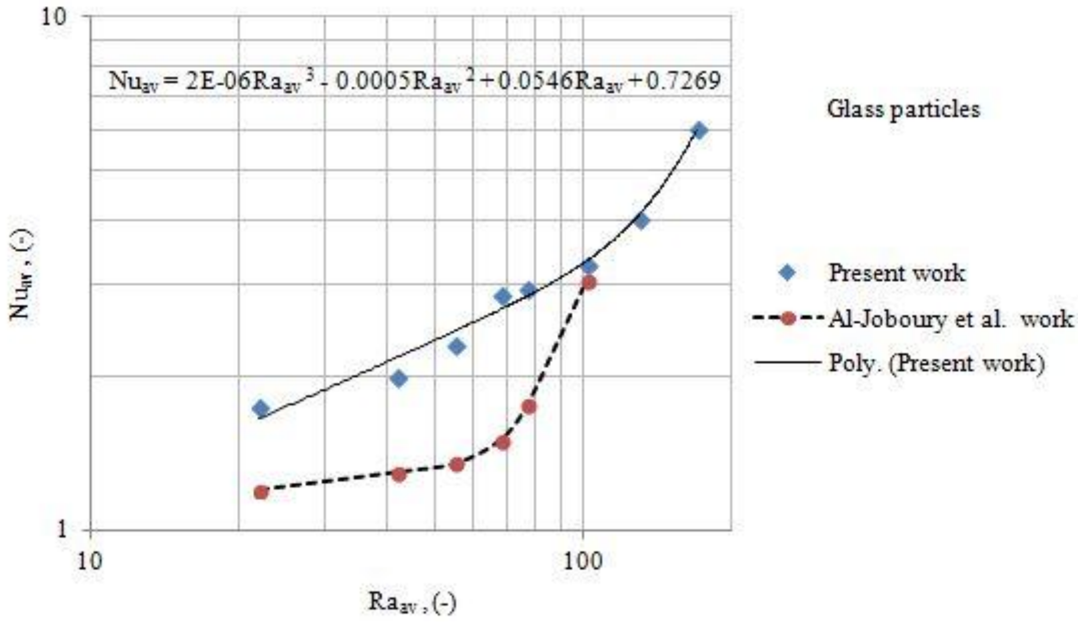


Figure 8. Analysis of the present work to the polynomial function fits with comparison of average Nusselt number than experimental results of Al-Joboury et al. work for glass particles.

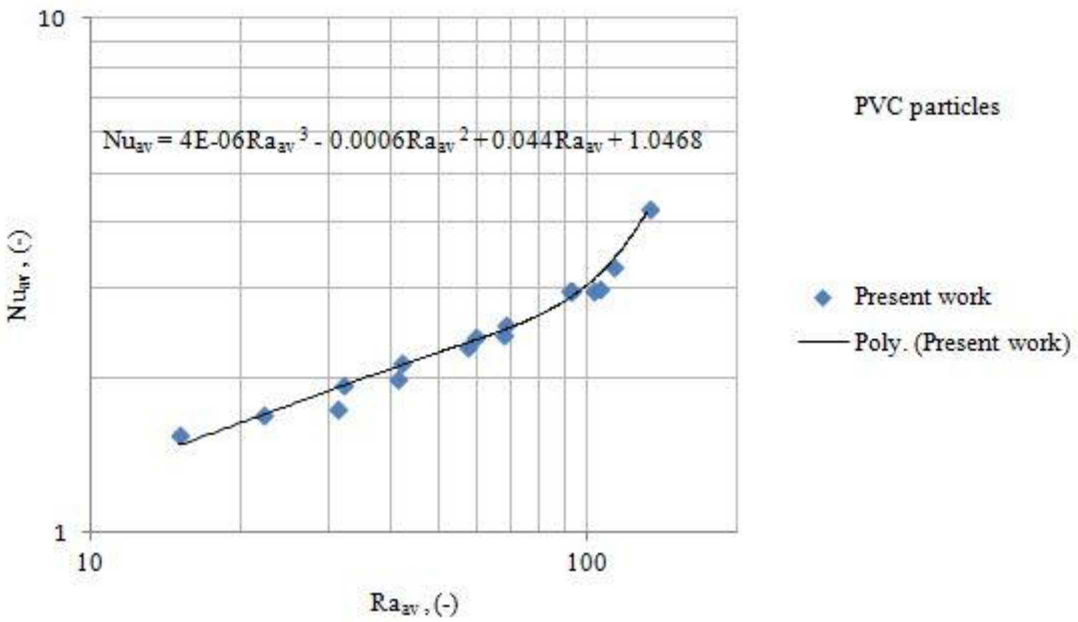


Figure 9. Analysis of the present work to the polynomial function fits for PVC particles.

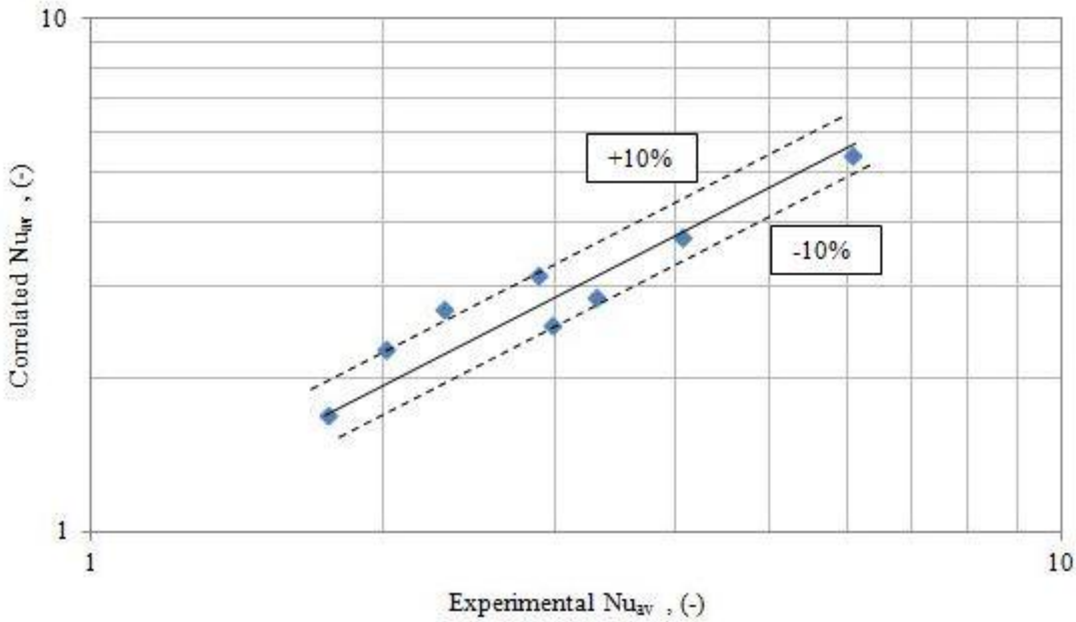


Figure 10. Comparison of correlated average Nusselt numbers (Nu_{av}) than those experimental of present work for glass particles.

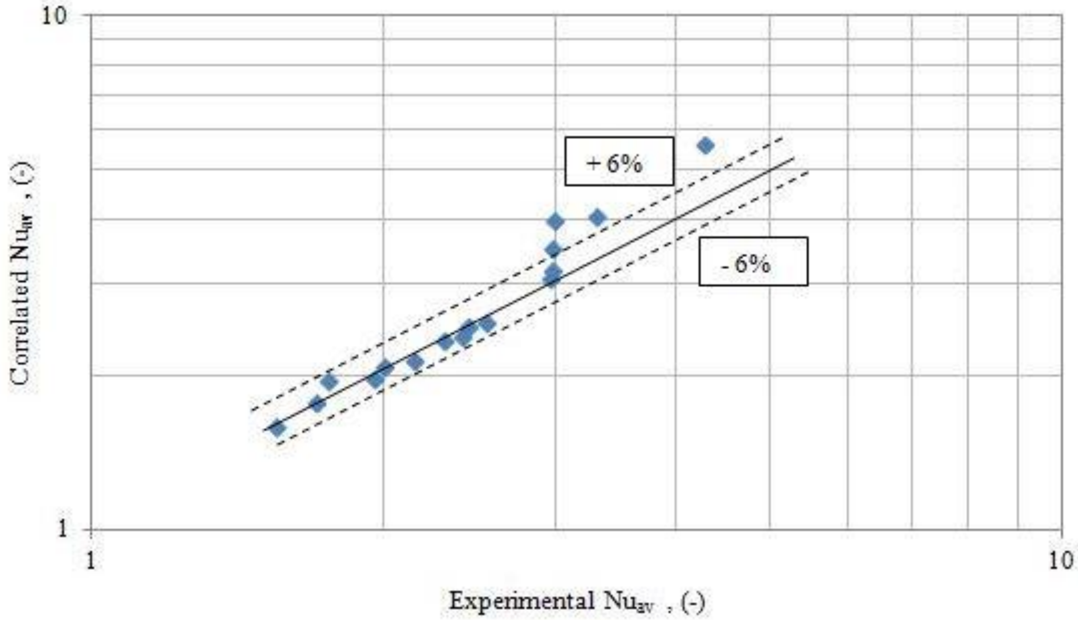


Figure 11. Comparison of correlated average Nusselt numbers (Nu_{av}) than those experimental of present work for PVC particles.