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EXPERIMENTAL STUDY ON HEAT TRANSFER COEFFICIENT AND FRICTION FACTOR OF Al_2O_3 NANOFLUID IN A PACKED BED COLUMN

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ABSTRACT

Forced convection heat transfer coefficient and friction factor are determined for flow of water and nanofluid in a vertical packed bed column. The analysis is undertaken in the laminar and transition Reynolds number range. The column is filled with spherical glass beads as the bed material. The heat transfer coefficients with Al_2O_3 nanofluid increased by 12 to 15% with the increase of volume concentration from 0.02 to 0.5% compared to water. The experimental values of axial temperature are in good agreement with NTU-ε method proposed by Schumann's model.

Keywords: Packed bed, Al_2O_3 nanofluid, convective heat transfer, friction factor, heat transfer enhancement.

INTRODUCTION

The process of forced convection in various installations such as boilers, solar collectors, heat exchangers and electronic devices is employed. However, low thermal conductivity of heat transfer fluids such as water, oil, and ethylene glycol mixture is a serious limitation for improving their performance. To overcome this disadvantage, there is strong reason to develop advanced heat transfer fluids with significantly higher conductivity. An innovative means of improving the thermal conductivities of fluids is to suspend nanosize solid particles in the fluid. (Tuckerman and Pease, 1982) conducted experiments for laminar flow in a channel. The heat transfer coefficient estimated is inversely proportional to the width of the channel, since the limiting Nusselt number is a constant. (Mahalingam,1985) confirmed the superiority of micro-channel cooling on a silicon substrate with a surface area of 5cm x 5cm using water and air as coolants. Many studies are directed towards evaluation of heat transfer coefficients for fluid flow in micro-channels.

Porous structures are also used for heat transfer augmentation as these aggravate the mixing of the flowing fluid and improve the convection heat transfer. Hence, studies are undertaken due to its broad applications. The early works are due to the effect of various parameters through experiments and statistical methods and developed a set of equations based on theoretical models for packed beds in tubular flow (Sadri, 1952).

Jeigarnik et al. (1991) experimentally investigated convection heat transfer of water on flat plates and in channels packed with sintered spherical particles, nets, porous metal and felt. The majority of the experiments is for the evaluation of heat transfer coefficients with different thickness (0.86 to 3.9 mm) and particle diameters (0.1 to 0.6 mm). They found that the porous media increased the heat transfer coefficient 5-10 times, although the hydraulic resistance increase is even more. Experiment to study the percolation behavior of fluids through a packed bed is undertaken by (Yagi et al., 1964), (Gunn et al., 1987; Lamine et al., 1992a). Analysis for heat transfer coefficients by Weekman and Myers (1995); Silveira (1991) and Lamine et al. (1992a) on gas-liquid flow, however, presented with limited results. Adeyanju (2009) experimentally determined the velocity variations with in a porous medium for packed beds. He concluded that the pressure drop across the porous medium was due to various factors, which included form drag, viscous drag, from bounding wall and inertia force. The results from this study confirmed that the pressure drop is a linear and quadratic function of flow velocity, at low and high Reynolds number respectively. (You et al., 2010) analyzed the thermal characteristics of an N_2O catalytic igniter as a hybrid system for small satellites. The authors analyzed the problem theoretically to determine the thermal performance of the catalytic igniter results on porosity, pumping capacity and the ratio of length to diameter. Carlos et al. (2008) predicted a generalised equation for radial velocity distribution in a packed bed having low tube to particle diameter ratio from six hydrodynamic models. Their calculations show that the use of an effective viscosity parameter to predict experimental data can be avoided, if the magnitude of the two parameters in Ergun's equation, related to viscous and inertial energy losses, are re-estimated from velocity measurements for the packed beds.

Maxwell (1904) showed the potential for increasing the thermal conductivity of a solution by mixing with solid particles. Fluids containing small quantities of nanosized particles are 'nanofluids'. The particles are less than 100nm dispersed in a liquid uniformly. The dispersion of nanoparticles in normal fluids enhances heat transfer even when added in small quantities. The nanofluids show greater potential for increasing heat transfer rates in a variety of cases. Lee et al. (1999) demonstrated CuO or Al_2O_3 nanoparticles in water and ethylene glycol exhibit enhanced thermal conductivity. The thermal conductivity increased by 20% at 4.0% concentration, when 35nm size CuO nanoparticles are mixed in ethylene glycol. (Mansour et al., 2007) used Al_2O_3 particles with a mean diameter of 13 nm at volume concentration of 4.3% and reported an increase in thermal conductivity by 30%.(Xuan and Roetzel, 2000) presented a relation for the evaluation of forced convection heat transfer coefficient for flow in tubes with Cu nanofluid.

Various concepts are proposed to explain the reasons for enhancement in heat transfer.Xuan (2003) and Xuan et al. (2004) have identified two causes for improvement in heat transfer with nanofluids; the increased dispersion due to the chaotic motion of nanoparticles that accelerates energy exchanges in the fluid and the enhanced conductivity of nanofluids considered by Choi (1995). Thermal conductivity of Al_2O_3 nanofluid has been evaluated by (Das et al., 2003) in the temperature range of 21-51^{o}C. They observed 2 to 4 fold enhancement of thermal conductivity in the range of concentration tested. (Wen and Ding, 2004) evaluated the heat transfer of nanofluid in the laminar region through experiments. They used equations available in the literature to determine viscosity at bulk temperature. (Maiga et al., 2005) investigated water-Al_2O_3 and Ethylene glycol - Al_2O_3 nanofluids observed adverse effects of wall shear when tested with the later. The heat transfer enhancement of the nanofluids can be expected due to intensification of turbulence, suppression of the boundary layer as well as dispersion or back mixing of the suspended particles, a large enhancement in the surface area of nanoparticles and a significant increase in the thermo physical properties of the fluid. Therefore, the convective heat transfer coefficient with nanofluid is a function of the physical properties of the constituents, dimension and volume fraction of suspended nanoparticles, and flow velocity. (Sarma et al., 2003) developed a theoretical model for the estimation of heat transfer coefficient under laminar flow in a tube with twisted tape inserts. (Syam Sundar et al., 2007) investigated heat transfer enhancement for nanofluid flow in a circular tube with twisted tape insets.

The impact of operating parameters on nanofluid flow in a packed bed on heat transfer has not been attempted. The influence of nanofluid concentration on parameters affecting forced convection heat transfer in a vertical tube filled with packing materials is undertaken. The temperatures are measured at different axial positions with p-type thermocouples as shown in Figure 1. Al_2O_3 nanofluid at 0.02, 0.1 and 0.5% volume concentration is pumped through the test section against gravity at different flow rates and inlet temperatures. The Nusselt number, friction factor and heat transfer coefficients are evaluated next.

Figure 1. Experimental test rig for packed bed

MODELS FOR PREDICTING THERMAL ANALYSIS OF PACKED BED

In order to determine the heat transfer of a packed bed system, several theoretical models have been reported in literature based on experimental investigations. A bed is of height LB, diameter D_p and cross sectional area '*A*' and packed with material having a void fraction 'ε' as shown in Figure 2. It is assumed that the temperature of the bed is uniform at an initial value T_{Bi}. The fluid enters at ' T_{fi}' with a mass flow rate '*m*' and leaves the bed at 'T_{f0}'. The bed height L_B is divided into a certain number of elements of thickness 'Δx'. The temperature at the entry of the element is 'T_{fm}' and exits at 'T_{fm+1}'.

2.1 Schumann Model

Schumann (1929) has modeled the thermal behaviour of packed beds which is extended by Sagara and Nakahara (1991). The model estimates the mean fluid and solid material temperatures at a given cross section as a function of time and axial position. The assumptions made by Schumann according to (Duffie and Beckman, 1991) are:

1. The bed material has infinite thermal conductivity in the radial direction with plug flow i.e. no temperature gradient in the radial direction.
2. Bed material has zero thermal conductivity in the axial direction.
3. Thermal and physical properties of the sold and fluid are constant.
4. The heat transfer coefficient does not vary with time and position inside the bed.
5. No mass transfer occurs.
6. No heat loss to environment.
7. No phase change of the fluid in the axial direction.
8. The flow is steady and uniform.

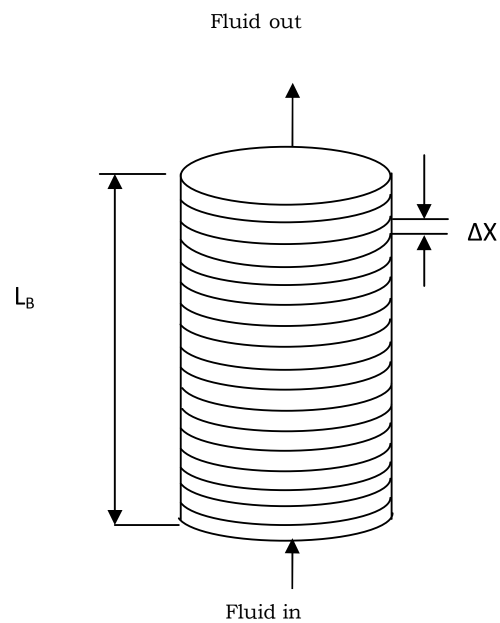

Figure 2. Elemental representation of packed bed domain

The energy balance equation for the fluid and solid components in Schumann model for packed bed can be written as

Energy in fluid at entry to bed = (Energy transferred to bed) + (Energy with the fluid in the bed) + (Energy with the fluid leaving the bed) + (Energy loss to environment)

$$m_f Cp_f T_{fi} = h_v (T_f - T_b) A dx + \rho_f C_{pf} \varepsilon A dx \frac{\partial T_f}{\partial t} + m_f C_{pf} \left(T_{fi} + \frac{\partial T_{fi}}{\partial x} dx \right) + Ul\ x (T_f - T_{fmb}) \quad (1)$$

Energy with the fluid in the bed and energy loss to the environment can be neglected as per the assumptions. Based on the assumptions stated, Eq. (1) becomes

$$\frac{\partial T_f}{\partial X} = - \frac{h_V A L_B}{\dot{m}_f C_{pf}} \left[T_f - T_S \right] \quad (2)$$

The above equation can be also written as

$$\frac{\partial T_f}{\partial X} = -\frac{h_V A L_B}{\dot{m}_f C_{pf}}\left[T_f - T_S\right] = NTU\left(T_f - T_S\right) \tag{3}$$

where, $X = \dfrac{x}{L_B}$, NTU (Number of transfer units) = $\dfrac{h_V A L_B}{m_f C_{pf}}$

Energy transferred to the material = Energy stored by the material

$$h_V\left(T_f - T_s\right)A_{cs}\,dx = \rho_s C_{ps}(1-\varepsilon)A_{cs}\,dx\,\frac{\partial T_s}{\partial t} \tag{4}$$

The above equation can be expressed as;

$$\frac{\partial T_s}{\partial \tau} = NTU\,(T_f - T_s) \tag{5}$$

where τ (dimensionless time) = $\dfrac{\dot{m}_f \dot{C}_{Pf}\, t}{(\rho_f C_{ps})(1-\varepsilon)A_{cs} L_B)}$

The equations (3) and (5) give the thermal performance of the packed bed. The exit fluid temperature from the bed is obtained by integrating the equation (3) and can be written as

$$\theta_{Th} = 1 - e^{-\frac{NTU}{N}} \tag{6}$$

The rate of heat transfer from fluid to bed element of thickness 'Δx' is given by;

$$Q = \dot{m}C_{pf}\left(T_{f,m+1} - T_{f,m}\right) \tag{7}$$

Eq. (7) with the aid of Eq. (6) can be written to obtain to determine the exit temperature of the fluid

$$\dot{m}C_{pf}\left(T_{f,m+1} - T_{f,m}\right) = \dot{m}_f Cp_f\left(T_{f,m} - T_{S,m}\right)\left(1 - e^{-NTU/N}\right) \tag{8}$$

Similarly Eq. (8) can be modified to calculate the mean temperature of bed elements 'm' as given below

$$\frac{dT_{S,m}}{d\tau} = CN\left(T_{f,m} - T_{S,m}\right) \tag{9}$$

where C is a constant and equal to $1 - e^{-NTU/N}$. Eq. (9) permits energy loss to environment at temperature T_{amb} and can be written as

$$\frac{dT_{S,m}}{d\tau} = CN\left(T_{f,m} - T_{S,m}\right) + \frac{U_m \Delta A_{CS,m}}{m_f C_{Pf}}\left(T_{atm} - T_{S,m}\right) \tag{10}$$

FABRICATION OF THE EXPERIMENTAL SETUP

The experimental setup consists of 4.0 cm diameter 50 cm height of a packed bed column. Figure3 shows the process and instrumentation diagram of the experimental setup. An immersion heater heats the water which is in connection with a feed water storage tank of 50 liter capacity. A pump with flow control and bypass valves supply a regulated flow of circulating working fluid through the test section. The flow rate of working fluid, the pressure drop across the bed and the variation of axial temperature are measured and recorded using suitable instrumentation.

The working fluid flows through a helical coil immersed in the hot water tank with the aid of a pump. It achieves the desired temperature before it enters the test section. The interaction between the cold bed and the hot fluid takes place. As a result, the fluid temperature at bed outlet decreases. The fluid recirculates in a closed circuit. When the bed reaches steady state, the pressure drop across the bed and temperatures along the bed length are obtained from personal computer through a data loger for two different glass beads of sizes 6 mm and 14.6 mm diameter.

ESTIMATION OF PRESSURE DROP

Of interest for the flow through packed beds is the relationship between flow velocity and the drop in pressure across the bed. Many theoretical correlations are available in literature to calculate this. However, the Sadri (1952) equation is used to calculate the pressure drop through a packed bed given by

$$\Delta P_{Th} = \frac{150\mu V_0 L_B}{D_P^2}\frac{(1-\varepsilon)^2}{\varepsilon^3} + \frac{1.75\rho V_0^2 L_B}{D_P}\frac{(1-\varepsilon)}{\varepsilon^3} \tag{11}$$

where the bed void fraction can be determined from the relation $\varepsilon = 1 - \frac{Vol_P}{Vol_B}$ where Vol_p and Vol_B are the volume of particles and bed respectively, V_0 is superficial velocity, D_P equivalent particle diameter given by $D_P = \frac{6V_P}{S_P}$ and S_P is surface area.

The experimental pressure drop is calculated with the help of differential height in mercury manometer given by the equation,

$$\Delta P_{Exp} = R_m(\rho_A - \rho_{bf})g/g_c \tag{12}$$

The pressure drops obtained from the Eq. (11) for different flow rates are compared with the experimental values and presented. The friction factors are calculated using the equation of Sadri (1952) applying pressure drop relations and presented as

$$f_{Ex} = \frac{\Delta P_{th} D_P}{L_B \rho_f V_S^2}\left(\frac{\varepsilon^3}{1-\varepsilon}\right) \tag{13}$$

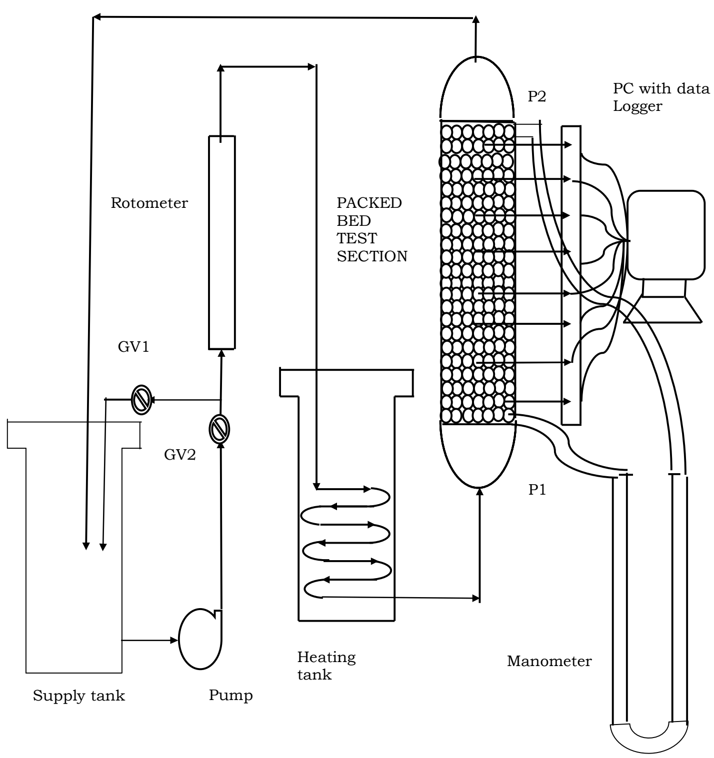

Figure 3. Schematic diagram of packed bed column Experimental

CALCULATION OF HEAT TRANSFER COEFFICIENT

The Energy balance equation for the packed bed can be estimated from the relation

$$Q_{Exp} = mC_{PL}(T_I - T_O) \tag{14}$$

where m is the mass flow rate. The Heat Transfer coefficient is estimated using Q_{Exp} and the difference between surface temperature of the bed and bulk mean temperature of the fluid is given by

$$h_{Exp} = \frac{Q_{Exp}}{A_S(T_S - T_{bf})} \tag{15}$$

where $\overline{T}_s = \sum_{i=1}^{i=8} T_{Si}/8$ and $T_{bf} = (T_I + T_O)/2$. The experimental Nusselt number is estimated with the relation

$$Nu_{Exp} = \frac{h_{Exp} D_P}{k} \tag{16}$$

Alazmi and and Vafai (2000) a correlation by conducting experiments with air, hydrogen, carbon dioxide and water. Experiments are undertaken in a narrow range of Prandtl numbers for packed bed Reynolds number ≤ 10000 with the characteristic dimension in Re_P taken as the bed particle diameter D_P. The validation of the correlation has been undertaken by Gnielinski (1980) who presented the relation as

$$Nu_{lam} = \frac{hD_p}{k} = 0.664 Re_p^{0.5} Pr^{1/3} \tag{17}$$

$$Nu_{tub} = \frac{hD_p}{k_f} = \frac{0.037\, Re_P^{0.8}\, P_r}{1 + 2.443\, Re_P^{-0.1}\left(P_r^{2/3} - 1\right)} \tag{18}$$

Gunn et al. (1987) presented an equation which is similar to Eq. (17) of Gnielinski (1980) in the absence of Nu_{tub} as

$$Nu = \frac{hD_p}{k} = 3.8 + 1.5\, Re_P^{0.5}\, Pr^{1/3} \tag{19}$$

Figure 4 represents the pressure drop in the packed bed with water and nano fluids at various volumetric concentrations. The pressure drop decreases with increase in bed particle diameter and Reynolds number. The pressure drop increases with increase in volume concentration of the nano fluid. The values of friction factor from theory are compared with those from experiment in Figure 5. The values are compared for water and nano fluid at various concentrations for 6mm and 14.56 mm particles. A regression equation is developed for the estimation of friction factor with an average deviation of ±0.08% and standard deviation of 1.68% as

$$f = 20.06\, Re_p^{-0.3117} (1 + \varphi)^{0.2919} \tag{20}$$

Figures 6 to 9 represent the variation of heat transfer coefficient for various concentrations of nano fluid. Figure 6 shows the variation of heat transfer coefficient with particle Reynolds. Nanofluids predict higher heat transfer coefficients compared to base fluid water. A regression equation is developed for the estimation of Nusselt number as a function of Reynolds number, Prandtl and volume concentration of nanofluid. It is obtained with a standard deviation of 1.56% and an average deviation of 3.92% given by

$$Nu = 0.188\, Re_p^{0.98} (1 + \phi)^{0.5310}\, Pr^{-0.4403} \tag{21}$$

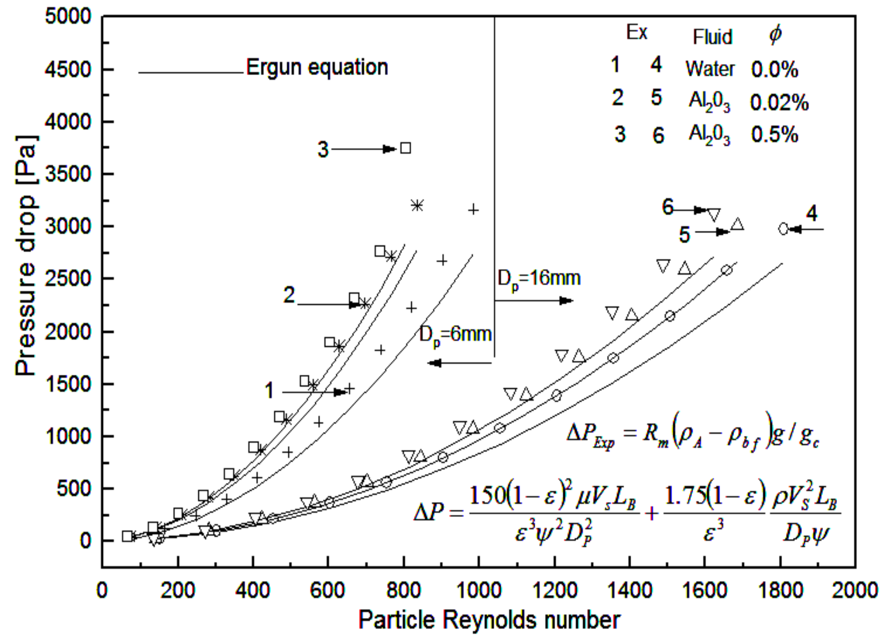

Figure 4. Comparison of experimental and theoretical pressure drop for water and nanofluids for 14.56 mm and 6 mm particles in packed bed

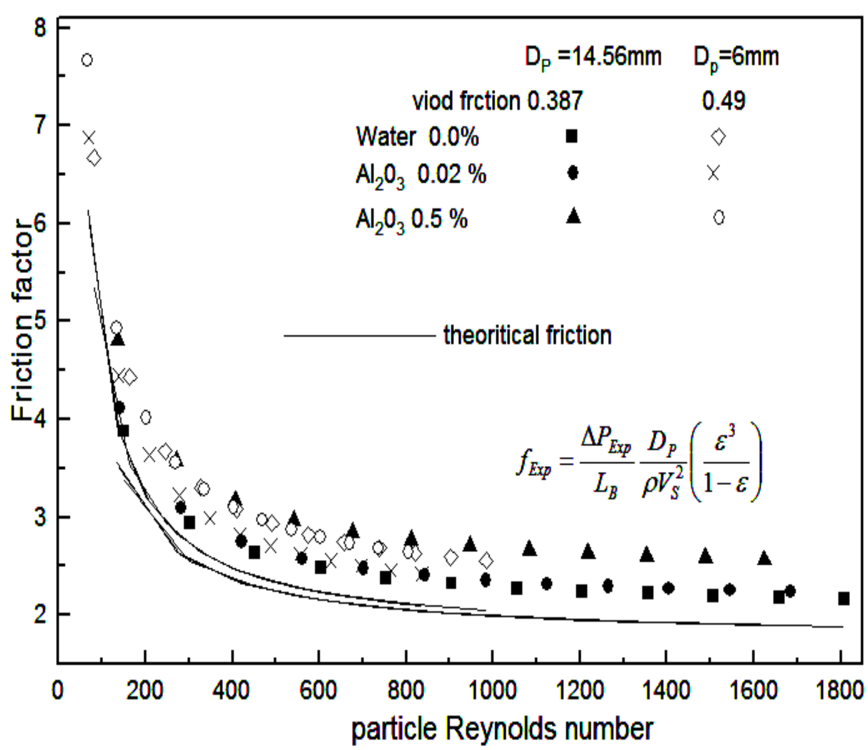

Figure 5. Comparison of experimental and theoretical friction factor for water and nanofluids for 14.56 mm and 6 mm particles in packed bed

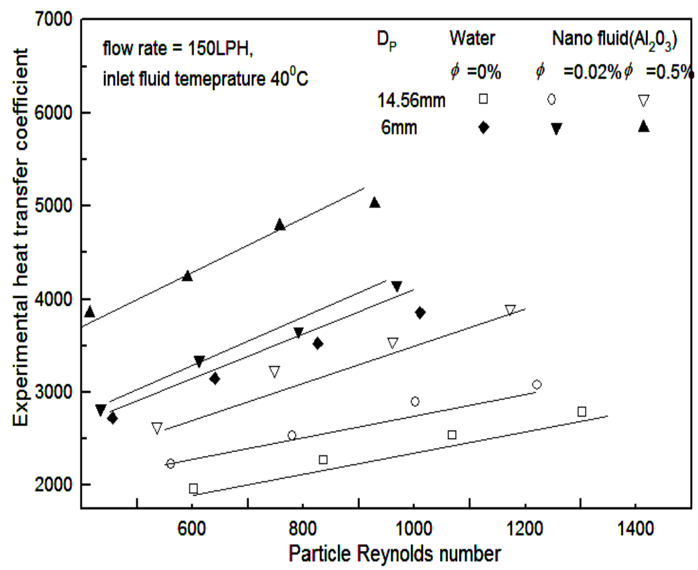

Figure 6. Comparison of heat transfer coefficient in packed beds with particle Reynolds number with 6mm and 14.56 mm glass particles beds with water and nanofluids

Figure 7 represents the variation of heat transfer coefficient with non dimensional axial distance along the bed length at 40^{0}C for minimum and maximum flow rates for water and nanofluid at two different concentrations for the two particles. The heat transfer coefficient increase with increasing flow rate and concentration of the nanofluid. Figure 8 represents the variation of heat transfer coefficient for 6mm, 14.56 mm particles for different operating conditions. The flow rate is 150 LPH at 40^{0}C for water and nano fluids at different concentration, the heat transfer coefficient increased with decreasing particle diameter.

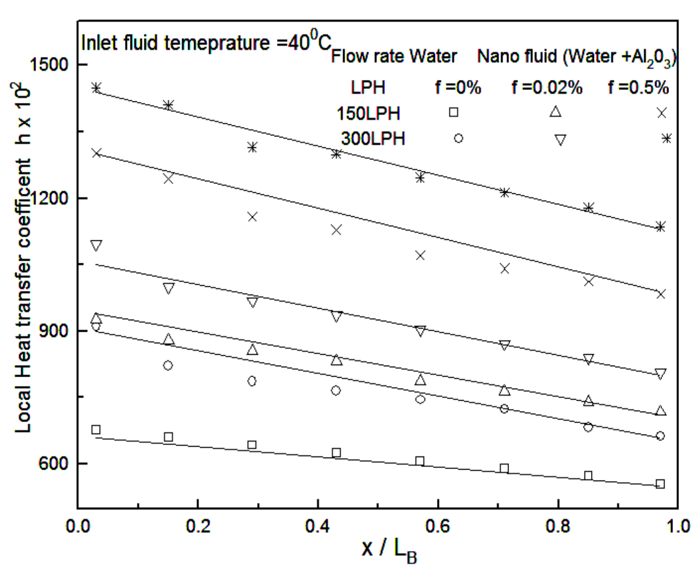

Figure 7. Effect of Al_2O_3 concentration on heat transfer coefficient comparison with non dimensional axial distance with14.56mm particles beds with water and nano fluids

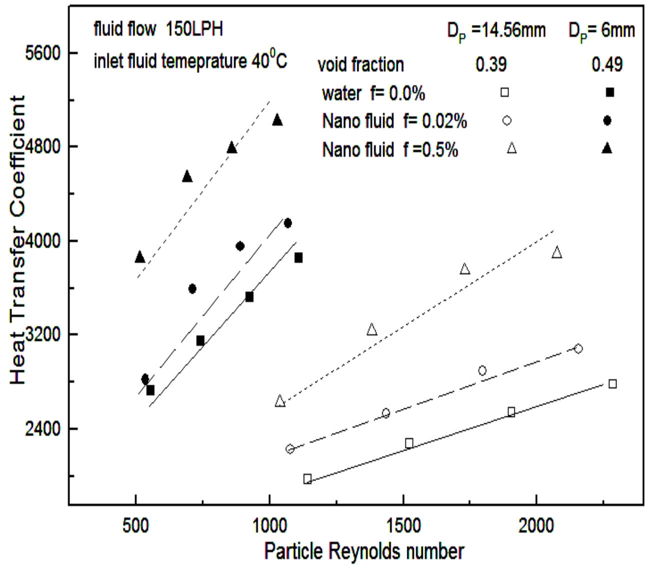

Figure 8. Heat transfer coefficient Vs Particle Reynolds number at fluid rate 150 LPH for 6 mm, 14.56 particles

Figure 9 shows the variation of heat transfer coefficient of water and nanofluid at high flow rates for two temperatures and particle concentrations. At higher flow rates and temperatures, the heat transfer coefficient is greater, for 6mm compared to 14.56 mm particle size.

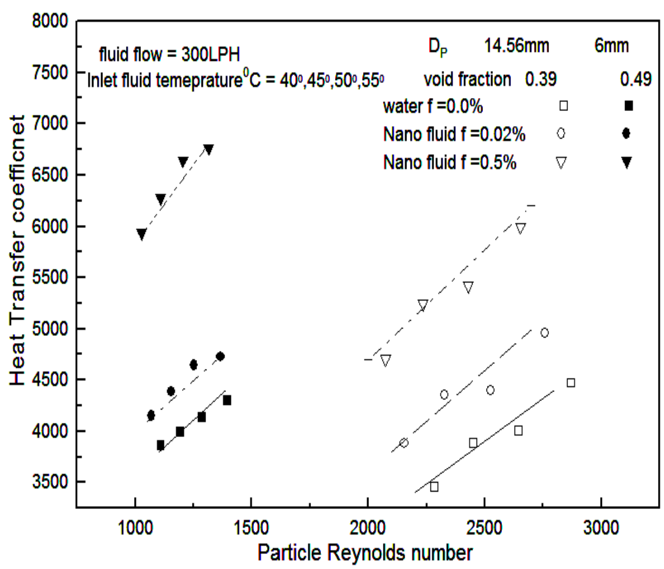

Figure 9. Heat transfer coefficient Vs Particle Reynolds number at fluid rate 300 LPH for6mm, 14.56 mm particles

Figures 10 to 11 represent the temperature distribution of the bed for two particle sizes. The experimental values are in agreement with Schumann-NTU method and other authors from literature. There is a reasonable agreement of experimental data with other theoretical investigations. The pressure drop with nanofluids is higher by 10%, and it increases with concentration of the nanofluid. Figure 11 shows a comparison of temperature profiles at minimum and maximum flow rate for bed of 14.56 mm particles. At the low flow rate, the temperature is greater than at high flow rate. There is no significant temperature variation with flow rate. The temperature variation is significant at higher concentrations of the nanofluid. Figure 12 represents the non dimensional fluid exit temperature distribution for 6mm particles at different flow rates in comparison with NTU method. The temperatures obtained with nanofluid are higher than for water. The theoretical results indicate reasonable agreement with the experimental values with a deviation of 10%.

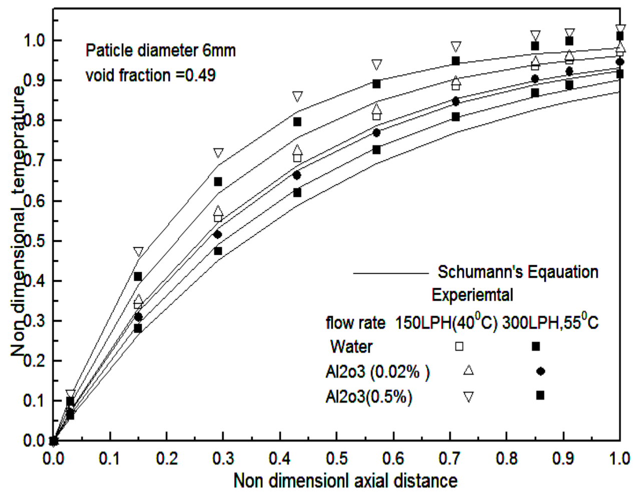

Figure 10. Comparison of non dimensional temperature distributation with non dimensional axial distance with NTU-ε method at 150 LPH and 300 LPH

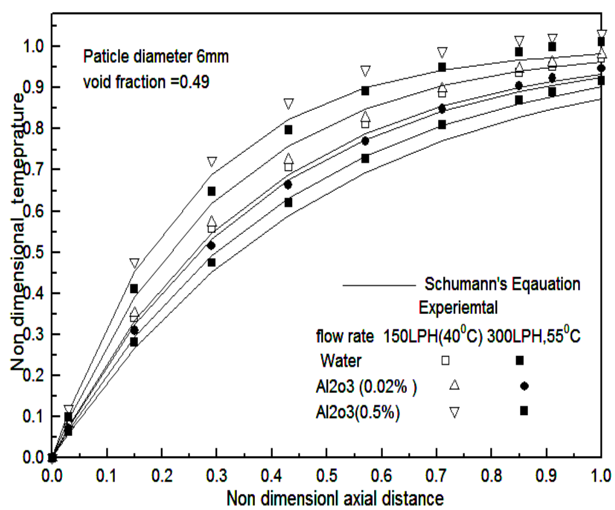

Figure 11. Comparison of non dimensional temperature distribution with non-dimensional axial distance with NTU-ε method at 150 LPH and 300 LPH for 6 mm particles

CONCLUSIONS

Heat transfer in a packed bed column filled with glass beads of 6.0 and 14.56 mm size is employed to determine heat transfer coefficient and pressure drop. The friction factor increased with decreasing particle diameter and increasing volume concentration of nanofluids compared to base fluid. The pressure drop is higher with nanofluids than with water by 10 to 15%. The pressure drop increased with nanofluid concentration. At lower concentration, the deviation of friction factor with nanofluid and water is significant than at higher concentration. The heat transfer coefficient is higher with 6mm particles due to larger surface area and the number of particles. Similarly, the heat transfer coefficient is greater at higher concentrations of the nanofluid. With an increase in volume concentration, the heat transfer is more and increases with the flow rate and inlet fluid temperature. The enhancement in heat transfer coefficient with nanofluids than base fluid lies between 10 to 15% due to higher values of thermal conductivity. The values from Schumann model agree with the experimental data for the two bead sizes of 6.0 and 14.56mm. The deviation between the two is less than 10%.

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Nomenclature

A	area of the bed, m^2
C	constant in Equation (9)
C_P	specific heat, J/kgK
D	diameter, m
f	friction factor
g	local acceleration of gravity, m/s^2
g_c	gravitational constant, $kg-m/N-s^2$
h	heat transfer coefficient, W/m^2K
$\overline{h}$	average heat transfer coefficient, W/m^2K
h_V	Volumetric heat transfer coefficient, $W/m^{-3}K^{-1}$
k	thermal conductivity, W/mK
L	length, m
$\dot{m}$	mass flow rate, kg/s
N	number grids in axial direction, $\Delta x / L$
NTU	Number of transfer units
Nu	Nusselt number , $h\,D_P/k$
P	pressure, Pa
ΔP	pressure drop, Pa
P_r	Prandtl number, $\mu\, C_{PL}/k$
Q	rate of heat transfer, W
R_e	Reynolds number, $\rho V_s D_P/\mu$
Re_p	Packed bed Reynolds number, $\rho V_s D_P/\mu(1-\varepsilon)$
R_m	differential height in manometer fluid
T	Temperature, K
$\overline{T}$	mean temperature, K
Vol	volume, m^3
V	velocity, m/s
X	x/L_B

Subscripts

A	mercury
B	bed
bf	bulk fluid
Ex	experimental
I	inlet

lam	laminar flow
o	outlet
f	fluid
L	liquid
P	particle
0	superficial
S	surface
Th	theoretical
tub	turbulent flow
x	local values
$Nano$	nano fluid

Greek Symbols

θ	non-dimensional fluid temperature
ρ	density of the fluid, kg/m^3
μ	dynamic viscosity, $N-s/m$
ε	void fraction
ϕ	volume concentration