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Natural Convection Heat Transfer of Al_2O_3 Nanofluid Through Packed Beds

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***Abstract*:- Natural convection heat transfer through packed beds has been presented in this paper. The heat transfer experimental data was presented and compared with data from general correlations proposed in the literature. The unsteady and steady state natural convection heat transfer of a nanofluid in vertical packed column is investigated. The Darcy model is used to formulate the problem. The nonlinear governing equations are solved using an unconditional stable finite difference scheme. The heat transfer enhancement and fluid temperature at the bed outlet for Al_2O_3 nanofluid is compared with that of water for different porosities of the bed. It is observed that heat transfer rate increases with a decrease in bed particles in the range of 6mm to 14.6mm diameter for the nanofluid concentration range of 0.02% to 0.5% and increases with concentration and flow rate.**

Keywords: Darcy model, heat transfer, natural convection, packed bed, nanofluid

1. INTRODUCTION:

The analysis of natural convection heat transfer in fluid saturated porous media plays an important role in many applications. These include geothermal engineering, thermal insulation systems, packed bed chemical reactors, porous heat exchangers, oil separation from sand by steam, underground disposal of nuclear waste materials, food storage, and electronic device cooling to name a few applications. Due to the characteristic qualities of high surface area, fluid mixing qualities, high thermal conductivity and applicability in a wide range of industrial applications, natural convection through porous media has gained considerable attention of various researchers in the past few decades.

Nield and Bejan [1992] and Kavinay [1995] have presented the main principle of heat transfer through porous medium. With the porosities in the range of 0.4 - 0.6 most of the researchers studied free convection through porous medium, packed beds and granular materials.

Prasad and Kulacki [1984], Prasad [1986] carried out studies on natural convection in a vertical porous annulus for isothermal heating by applying a constant heat flux at the inner wall for a much wider range of Rayleigh numbers, aspect ratios and radius ratios than those considered by previous authors.

The natural convection, in a vertical annulus without porous media, has been extensively investigated in the literature for uniform or discrete heating. Natural convection in a porous square cavity with an isoflux and isothermal discrete heater placed at the walls has been numerically studied by Saeid and Pop [2005] using the Darcy model. They found that maximum heat transfer can be achieved when the heater is placed near the bottom of the cavity. Saeid [2006] numerically studied the natural convective flow induced by two isothermal heat sources on a vertical plate channel filled with a porous layer. Natural convection heat transfer in a square porous enclosure due to non-uniformly heated walls has been investigated in the literature by Basak et al. [2007] and Sathiyamoorthy et al. [2007]. Using Bejan’s heat lines method, Kaluri et al. [2009] analyzed the optimal heating in a square cavity filled with a fluid saturated porous medium for three different thermal conditions.

Natural or free convection in packed beds was carried out by Beckman [1931] and later extended by Kraussold [1934]. The experimental study of natural convection in packed beds was undertaken by Elder [1967], who proposed a correlation for the estimation of mean Nusselt number, developed as a function of Rayleigh number. Kuehn and Goldstein [1978] have undertaken experimental studies using Mach-Zehnder interferometer and conducted numerical simulations using the finite elements technique. Charrier-Mojtabi et al. [1979], Farouk and Guceri [1981] solved for natural convection under turbulent flow using a two-equation model to predict the Nusselt number leading to improved accuracy in predicting the thermal performance of storage systems. Garon and Goldstein [1973], Caltagirone [1976b], Poulikakos [1984], Bejan [1987], Oosthuizen and Naylor [1989] proposed correlations for the estimation of natural convection heat transfer in a porous medium.

The combined effect of heat and mass transfer with porous medium have been reported by Nield [1968], Khan and Zebib [1981]. Calmidi and Mahajan [1999], Zhao et al. [2004, 2005] observed natural convection using steel alloy foam with air as the interstitial fluid. Other researchers like Phanikumar et al. [2002], Krishnan et al. [2004], Zhao et al. [2004, 2005], and Edimilson et al. [2006] have reported studies on natural convection using metal foams for attaining improved thermal storage performance.

The conventional heat transfer fluids, such as oil, water and ethylene glycol mixture are poor heat transfer fluids. The thermal conductivity of fluids between the heat transfer medium and the heat transfer surface, play an important role in determining the heat transfer coefficient. An innovative technique for improving heat transfer, by using ultra fine solid particles in the fluids, has been developed and used extensively during the last years. Certain investigations for the estimation of heat transfer coefficient of Al_2O_3 and CuO nanofluids in a circular tube under laminar and turbulent flow conditions have been reported by several researchers. Compared with conventional fluids, nanofluids have been reported to have higher heat transfer rates.

Sundar et al. [2007] undertook the experimental determination of heat transfer coefficients by measuring the viscosity and thermal conductivity, using Al_2O_3 nanofluid, for flow in a plain tube and with twisted tape insert. They concluded that nanofluids enhance heat transfer coefficients which can reduce the size of thermal storage system. Rao et.al. [2011], [2012] have estimated pressure drop and the heat transfer coefficient in packed beds with the same nanofluid and concluded that nanofluids performed better than conventional fluids such as air and water.

The objective of the work is to determine heat transfer coefficient and pressure drop under natural convection flow of fluid in a packed bed filled with glass beads and pebbles from unsteady to steady state of temperature. The governing equations of continuity, momentum and energy in non dimensional form are solved subjected to relevant boundary conditions. The experiments values at various flow rates, inlet temperatures and nanofluid concentrations are compared with the numerical results. The properties of nanofluids are adopted from the generalized equations of Sharma et.al [2012]. The results of unsteady temperatures are compared with the numerical solutions of Burmeister [1993], Bejan [1995], Hornung [1997] and those of steady state with that of Oosthuizen [1989].

2. MATHEMATICAL MODEL

Mathematical models are employed by Burmeister [1993], Bejan [1995], Hornung [1997] for the determination of transient temperatures in a fluid–saturated packed bed storage system under natural convection at low flow rate of the fluid.

The assumptions made by these investigators include:

i. Isotropic porous medium
ii. Thermal equilibrium between the phases
iii. Negligible internal heat generation
iv. Negligible temperature gradient in radial direction

The governing equations are

$$\frac{\partial\rho}{\partial t}+\frac{1}{r}\frac{\partial}{\partial r}(\rho r v)+\frac{1}{r}\frac{\partial}{\partial\theta}(\rho\omega)+\frac{\partial}{\partial z}(\rho u)=0 \tag{1}$$

z- component

$$\rho\left(\frac{\partial u}{\partial t}+v\frac{\partial u}{\partial r}+u\frac{\partial u}{\partial z}\right)=-\frac{\partial p}{\partial z}+\mu\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial u}{\partial r}\right)+\frac{\partial^2 u}{\partial z^2}\right] \tag{2}$$

r- component

$$\rho\left(\frac{\partial v}{\partial t}+v\frac{\partial v}{\partial r}+u\frac{\partial v}{\partial z}\right)=-\frac{\partial p}{\partial r}+\mu\left[\frac{1}{r}\frac{\partial}{\partial r}\left(r\frac{\partial v}{\partial r}\right)+\frac{\partial^2 v}{\partial r^2}\right] \tag{3}$$

Energy Equation

$$\rho C_p\left(\frac{\partial T}{\partial t}+v\frac{\partial T}{\partial r}+u\frac{\partial T}{\partial z}\right)=k\left(\frac{1}{r}\frac{\partial T}{\partial r}+\frac{\partial^2 T}{\partial z^2}\right) \tag{4}$$

where the non-dimensional terms are:

$$\overset{*}{R}=\frac{r}{R},Z=\frac{z}{L_b},\theta_f=\frac{T-T_\alpha}{T_{fi}-T_\alpha},$$

$$\overset{*}{u}=\frac{u}{U_0},\overset{*}{v}=\frac{v}{U_0}\frac{L_b}{R},\tau=\frac{\alpha t}{\sigma_b R^2},Ra_w=\frac{Kg\rho_f\beta L_b(T_{fi}-T_{fo})}{\mu\alpha} \tag{5}$$

and K,ρ_f,α,β and μ represents the permeability of bed, density, thermal diffusivity, coefficient of thermal expansion, and absolute viscosity of the fluid respectively.

In the present experimental setup, as the diameter is small compared to length of the bed, the variation of temperature in the radial direction is ignored. The Eqs. (1), (2), (3) and (4) are transformed into non dimensional terms as

$$\frac{\partial \overset{*}{u}}{\partial Z} = 0 \tag{6}$$

$$\left(\frac{\overset{*}{R}}{L_b}\right)\frac{\partial \overset{*}{u}}{\partial Z} = 0 \tag{7}$$

$$\frac{\partial \theta_f}{\partial \tau} + Ra_w\left(\frac{\overset{*}{R}}{L_b}\right)^2\left(\overset{*}{u}\frac{\partial \theta_f}{\partial Z}\right) = \left(\frac{\overset{*}{R}}{L_b}\right)^2\frac{\partial^2 \theta_f}{\partial Z^2} \tag{8}$$

Equations (6) and (7) are constrained by the following boundary conditions along axial direction:

At $Z = 0,\ \overset{*}{u} = 0,\ \theta_f = 1$

$$\text{At } Z = 1,\quad \overset{*}{u} = 0,\ \theta_f = 0 \tag{9}$$

The initial condition is given by

$$\tau = 0,\ \theta_f = 1,\ \overset{*}{u} = 0 \tag{10}$$

The equations (6) - (8) are solved numerically with explicit finite difference approximate analysis subjected to boundary conditions (9) and (10) for the determination of non-dimensional temperature profiles. The experimental values are shown in comparison with the theoretical results and plotted through Figs. (3) - (6).

3. EXPERIMENTAL SETUP

The experimental setup consists of 4cm diameter and 50 cm length of a bed column. Figures 1 and 2 show the experimental flow process and instrumentation diagram. An immersion heater heats the water, which is connected to feed water storage tank of 50 liters capacity. The tank is setup at a height of 2m from the ground level to maintain sufficient head to conduct the experiment under free convection mode. The working fluid (water/nanofluid) is heated in the elevated tank and enters the test section due to gravity at a predetermined temperature. The interaction between the cold bed and the hot fluid takes place. As a result, the fluid temperature decreases at the bed outlet. The working fluid is pumped to the storage tank for recirculation in a closed circuit. The working fluid flows through a helical coil immersed in the hot water tank with the aid of a pump, having flow control and bypass valves, entering the test section under gravitational force. The flow rate of working fluid and the variation of axial temperature are measured and recorded using suitable instrumentation. It achieves the desired temperature before it enters the test section. When the bed reaches a steady state, the temperatures along the bed length are obtained from personal computer through a data logger for two different glass beads of sizes 6 mm and 14.6 mm diameter. Experiments are undertaken with two bed particle sizes of 6mm and 14.6 mm with glass beads using water and nanofluid for flow rates of 15 and 30LPH with Rayleigh numbers in the range of Ra_f = 6.0E6 to 22.0E6 for both 6mm and 14.6mm glass beads for water and nanofluid at different concentrations. Experimental investigations are undertaken for inlet temperatures of 40 to 55^{0}C in steps of 5^{0}C. The working fluid water and nanofluid at 0.02, 0.1 and 0.5% volume concentrations are employed in the investigations with lower flow rates at 15 and 30 LPH considered for better understanding of flow phenomena. The experiments are conducted till the steady state is reached. The experimental Nusselt numbers are compared with the values obtained from equations available in the literature.

4. THERMO PHYSICAL PROPERTIES OF NANOFLUID

The density and specific heat of nanofluid at different volume concentrations are estimated using the following relations valid for homogeneous mixture given by

$$\rho_{nf} = \phi\,\rho_p + (1-\phi)\rho_{bf} \tag{11}$$

$$C_{p_{nf}} = \frac{(1-\phi)(\rho\,C_p)_{bf} + \phi\,(\rho C_p)_p}{(1-\phi)\rho_{bf} + \phi\rho_p} \tag{12}$$

The experimental data of viscosity available in the literature for Cu, CuO, Al_2O_3, TiO_2, SiO_2, ZrO_2, ZnO nanoparticles dispersed in water is subjected to regression by Sharma et al. [2012] for $0 \le \phi \le 3.7$, $20 \le T_b \le 70$, $20 \le d_p \le 170$. The equations are obtained with a deviation of less than 10% where ϕ is in percent, T_b in ^{0}C and d_p in nanometer is given by

$$\frac{\mu_{nf}}{\mu_w} = C_1 \left(1+\frac{\phi}{100}\right)^{11.3} \left(1+\frac{T_{nf}}{70}\right)^{-0.038} \left(1+\frac{d_p}{170}\right)^{-0.061} \tag{13}$$

Sharma et al. [2012] developed an equation for the determination of thermal conductivity of Cu, CuO, Al_2O_3, TiO_2, SiO_2, ZrO_2, ZnO nanoparticles dispersed in water valid in the range $0 \le \phi \le 3.7$, $20 \le T_b \le 70$, $20 \le d_p \le 170$. The equation presented by them is used to determine the thermal conductivity of nanofluid for the experimental conditions given by

$$\frac{k_{nf}}{k_w} = \left[0.8938 \left(1+\frac{\phi}{100}\right)^{1.37} \left(1+\frac{T_{nf}}{70}\right)^{0.2777} \left(1+\frac{d_p}{150}\right)^{-0.0336} \left(\frac{\alpha_p}{\alpha_w}\right)^{0.01737}\right] \tag{14}$$

5. EXPERIMENTATION

The data of temperature variation of the bed for different concentrations of the nanofluid and water was noted at the 60th minute and the values were compared with those obtained using the Eqs. 6 and 7 shown through Figs. 3 to 5. The temperature variation of the water and bed along its length for four inlet temperatures of 40, 45, 50, and 55^0 C is determined at different flow rates. Temperature profiles for inlet conditions of 40 and 55^0 C, for nanofluid concentration of 0.02 and 0.5 are presented. The variation of bed temperature along the length for 14.6 mm particles is presented in Fig. 2. The temperature of the bed at 15LPH flow rate is lower compared with the value obtained at a flow rate of 30LPH as shown in Fig 2. This is due to lower heat transfer exchange between the fluid and the bed. The effect of concentration on temperature profiles are shown in Fig.3. The temperature gradients with nanofluid concentration of 0.5% at 30LPH flow rate are greater than water for an identical inlet temperature of 55^{0}C. It can be due to greater values of nanofluid thermal conductivity compared to water at the same temperature. Under similar operating conditions for both particles as presented in Fig. 4, the gradients with small size bed particles compared with larger diameters are greater due to the greater surface area. The temperature gradients estimated with the theoretical model are observed to be in satisfactory agreement with the experimental values. The steady state values of experimental Nusselt numbers are compared with numerical results employing Eqs. (15) to (17). Garon and Goldstein [1973] proposed an equation for the estimation of Nusselt number for water saturated in metal foam for free convection given by Eq. (15).

$$Nu_G = 0.13 Ra_f^{0.293} \tag{15}$$

Oosthuizen and Naylor [1989] proposed a generalized correlation for free convection in rectangular enclosures with porous medium given by Eq. (16)

$$Nu_O = 0.508 Ra_w^{0.5} / AR^{0.5} \tag{16}$$

Equation (17) is developed by Cheng [1980] for the determination of free convection heat transfer in porous medium. It is given as

$$Nu_C = 0.362 Ra_f^{0.5} \tag{17}$$

The experimentation values are estimated with the energy balance relation

$$Q = m_f Cp_f (Tf_i - Tf_o) \tag{18}$$

$$h_{\exp} = Q / A_s (T_s - T_b) \tag{19}$$

where T_s and T_b are the average surface and bulk temperatures. The values of experimental Nusselt numbers are in good agreement with the values estimated with Eqs. (15) to (17).

6 RESULTS AND DISCUSSION

Experiments are undertaken with two bed particle sizes of 6mm and 14.6mm with glass beads using water and nanofluid at flow rates of 15 and 30LPH. The temperature variation of the bed for different concentrations of the nanofluid and water flow determined at the 60th minute is shown through Figures 3 to 5. The temperature variation of the water and bed along its length for two inlet temperatures of 40^{0}C and 55^{0}C is presented in Figure 3. Higher inlet temperatures show greater non-dimensional temperatures for the fluid and the bed for 30LPH. The effect of concentration on the temperature profiles are shown in Figure 4. At 0.5% nanofluid concentration, for a flow rate of 30LPH, an inlet temperature of 55^{0}C, the temperature gradients for nanofluids are greater than water by a 5% to 7%; this is due to larger values of thermal conductivity of nanofluid compared to water at the same temperature. Under similar operating conditions for two particle sizes presented in Figure 5, the gradients for smaller size bed particles are higher due to the greater surface area compared to particles of larger diameter. The temperature gradients estimated with theoretical models are observed to be in satisfactory agreement with the experimental values.

Fig 6 reveals the effect of flow rates on Nusselt number compared with Darcy-modified Rayleigh number, Ra_w for 14.6mm glass beads using water as the working fluid. The experimental values of Nusselt number at an aspect ratio AR =12.2 for different values of Ra_w are determined. The Nusselt number is observed to increase with the flow rate of the working fluid. The variation of temperature and nanofluid concentration on the values of Nusselt number is shown in Figure 7. The experiments are undertaken at various temperatures in the range of 40 to 55^{0}C at a constant flow rate of 15LPH with 14.6mm glass beads as bed particles. The Nusselt numbers obtained are compared with Eq.(16) of Oosthuizen [1989]. The experimental values of water obtained with 6mm and 14.6mm glass beads are plotted and compared with the regression Equations of Chang [1980], Oosthuizen [1989] and Garon & Goldstein [1973] for steady state natural convection are shown in Figure 7. Experiments are undertaken with nanofluid at different mass flow rates and concentration with glass beads of 14.6mm diameter are presented in Figure 9. The Nusselt number increases with flow rate and concentration of the nanofluid. The experimental heat transfer coefficients are estimated with the Eqs.(18) and (19) and compared with the values from theory given by Eq. (16).

It can be observed from Figures (6) to (9) that the experimental Nusselt numbers are in satisfactory agreement with the values obtained with Oosthuizen correlation (1989) Eq. (16). A regression equation applicable for Raleigh number in the range of 13.3 x 10^6 to 22.1 x 10^6, bulk temperature between 40^{0}C and 55^{0}C for bed particle diameter applicable between 6.0 and 14.56mm is given by

$$Nu = 0.011251 Ra_w^{0.8811}(1+\phi)^{0.2493} \tag{20}$$

valid in the volume concentration range of 0.02< ϕ< 0.5%

7. CONCLUSIONS

The experimental results are in good agreement compared with the numerical results for the base liquid water. The enhancement in Nusselt number with nanofluid is greater by 10 % with 0.5% nanofluid concentration. The following conclusions are made.

1. The temperature gradient increases with time and concentration of the nanofluid.
2. The temperature gradient is predicted both experimentally and theoretically. Mathematical models are employed by Burmeister [1993], Bejan [1995], Hornung [1997] for the determination of transient temperatures in a fluid–saturated packed bed storage system under natural convection associated with low flow rates. The experimental gradients are in good agreement with theoretical models with 10 to 20% of deviation. The temperature gradient increases with time and concentration of the nanofluid.
3. The nanofluid heat transfer coefficient is greater with 6mm compared to 14.56mm bed particles, at the same flow rate. The heat transfer coefficient for 0.5 % nanofluid with 6mm particles is 1.72 times higher at lower temperatures and 1.85 times higher at 55^{0}C with Raleigh number from 13.29x10^6 to 22.12x10^6 from lower to higher temperatures.
4. The thermal energy storage capacity of the packed bed system is enhanced with increasing concentration of nanofluid by 10% to 15% by using nanofluid at 0.5% of volume concentration.

NOMENCLATURE

C_p specific heat, J/KgK

d diameter

D_a Darcy number, $\frac{K}{D_t^2}$

h convective heat transfer coefficient, W/m^2K

$h_{\exp}$ Experimental heat transfer coefficient

k thermal conductivity, W/mK

K Permeability

L length of the tube, m

Nu Nusselt number based on the fluid properties,

$\Pr$ Prandtl Number, $\mu_f Cp_f / k_f$

r Radius

m mass flow rate

$\overset{*}{R}$ Nondimensional radius

Ra_f Rayleigh number based on the fluid properties

$$\frac{g\beta_f D_t^3 (T_h - T_c)}{(\alpha\nu)_f}$$

Ra_w Darcy modified Rayleigh number,

$$\frac{g\beta_f K D_t (T_h - T_c)}{(\alpha\nu)_f}$$

T Temperature, K

Z non dimensional axial length $\left(\frac{z}{L_b}\right)$

$u, \upsilon,$ Velocity components in axial, radial directions

U_0 Reference velocity /superficial velocity of fluid

Greek symbols

α Thermal diffusivity, $\left(\frac{k}{\rho C_p}\right)$

ε Void fraction

θ_f Dimensionless fluid temperature, $\left(\frac{T - T_\alpha}{T_{fi} - T_\alpha}\right)$

β Thermal expansion coefficient

μ dynamic viscosity, kg/m^2 s

ρ density, kg/m^3

τ non dimensional time

ϕ volume concentration of nanoparticles, %

Subscripts

bf base fluid

eff effective

p nanoparticle

th theoretical

w water

f fluid

nf Nanofluid

i Inlet

0 Outlet

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Figure. 1 Photograph of the experimental setup

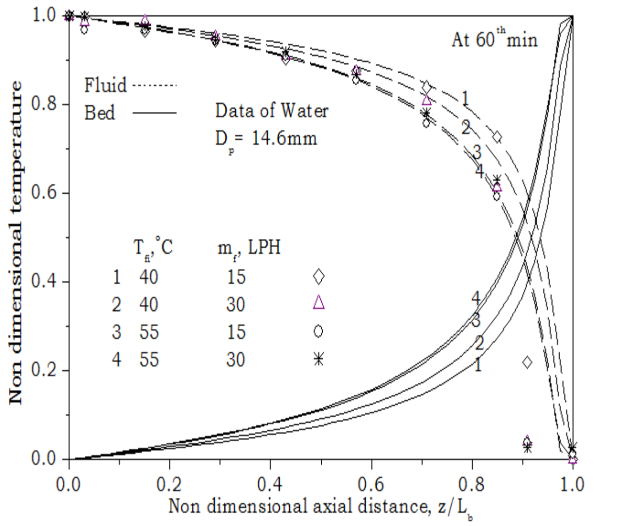
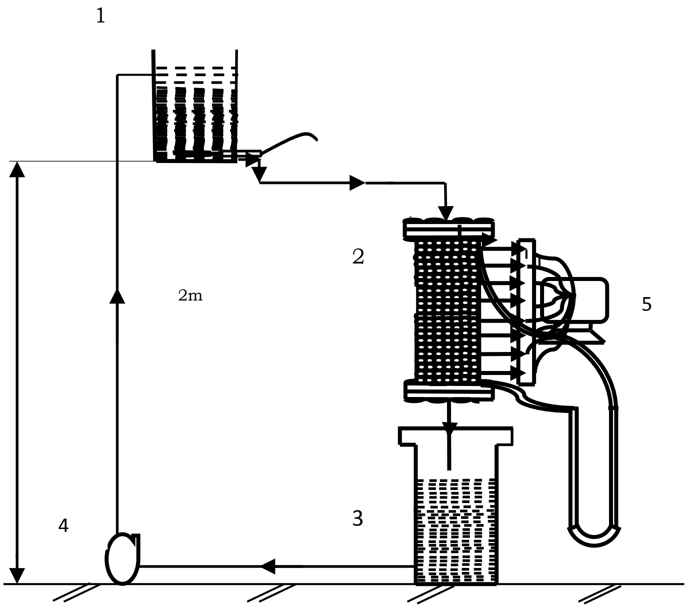


Figure 3 Effect of mass flow rate and inlet temperature on temperature distribution for solid and fluid.



Part list: 1. Supply tank. 2. Test section 3. Collecting tank 4.Pump 5.Data Acquisition System
Figure. 2 Flow diagram for natural convection

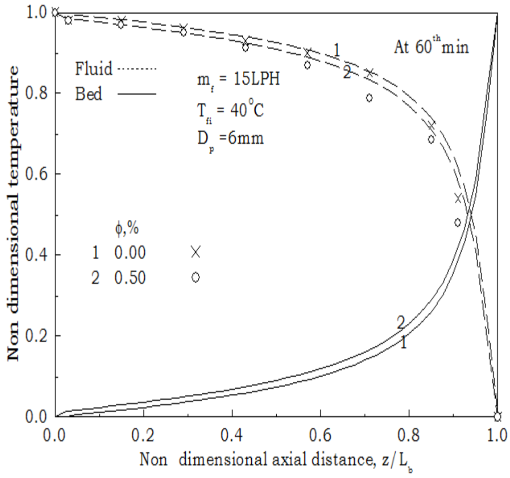


Figure4: Effect of nanofluid concentration on temperature distribution for fluid and solid

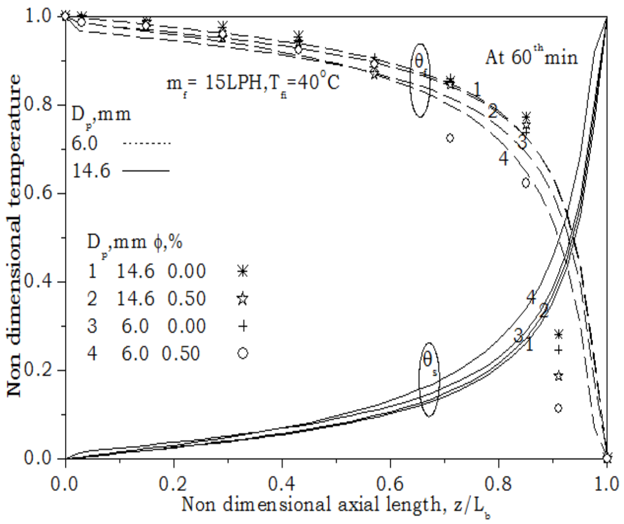


Fig. 5: Effect particle size of particle and concentration of nanofluid on nondimensional temperature distribution for solid and fluid

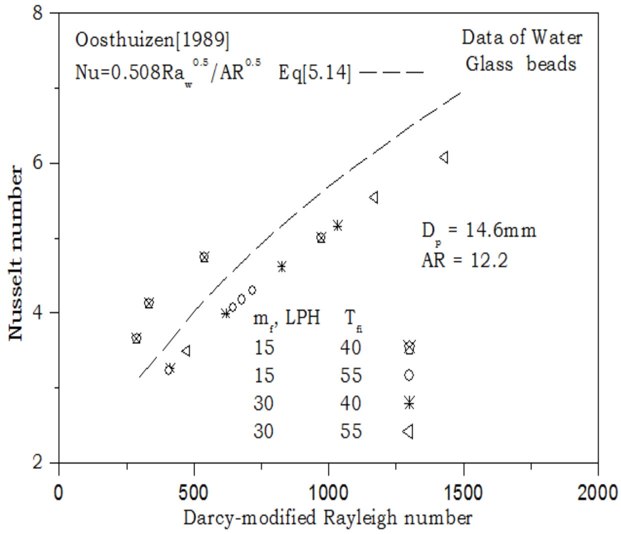


Figure . 6 Effect of flow rate and temperature on Nusselt number for validation of experimental setup with free convection

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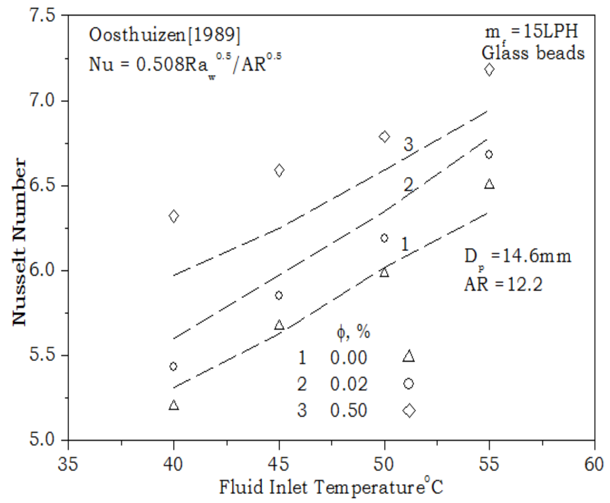

Fig.7 Effect of fluid inlet temperature and concentration on Nusselt number at lower flow rates

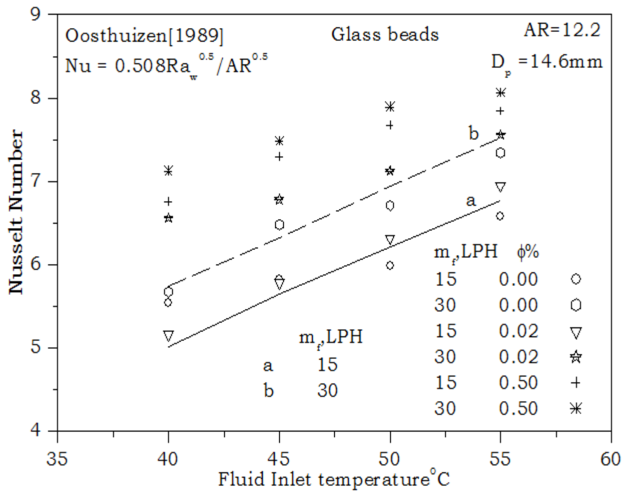

Fig. 9 Effect of flow rate and concentration of nanofluid on Nusselt numbers

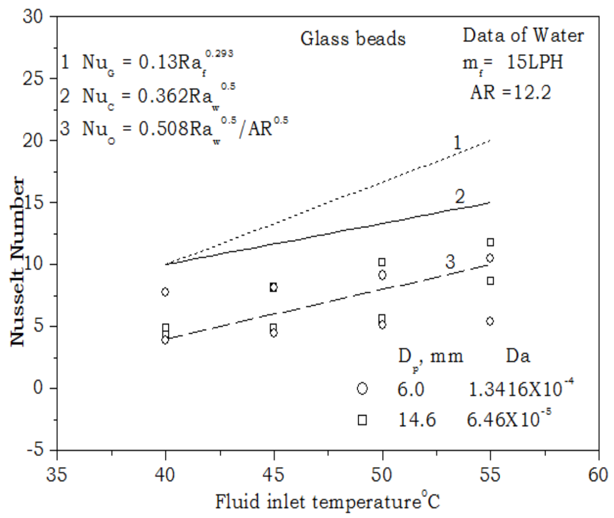

Figure 8: Effect of particle size and fluid inlet temperature on Nusselt numbers