Yimin Xuan
e-mail: ymxuan@mail.njust.edu.cn

Qiang Li

School of Power Engineering,
Nanjing University of Science & Technology,
Nanjing 210094, China

Investigation on Convective Heat Transfer and Flow Features of Nanofluids

An experimental system was built to investigate convective heat transfer and flow features of the nanofluid in a tube. Both the convective heat transfer coefficient and friction factor of the sample nanofluids for the turbulent flow are measured, respectively. The effects of such factors as the volume fraction of suspended nanoparticles and the Reynolds number on the heat transfer and flow features are discussed in detail. A new type of convective heat transfer correlation is proposed to correlate experimental data of heat transfer for nanofluids. [DOI: 10.1115/1.1532008]

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1 Introduction

With progresses of thermoscience and thermal engineering, many efforts have been devoted to heat transfer enhancement. Among them, application of additives to liquids is often involved. Since the flow media themselves may be the controlling factor of limiting heat transfer performance, solid additives are suspended in the base liquids in order to change transport properties, flow and heat transfer features of the liquids [1,2]. Besides describing chronological development in this field, Hetsroni and Rozenblit [3] investigated the thermal interaction between the particles-laden turbulent flow and a heated plate for a liquid-solid mixture consisting of water and polystyrene particles. Traditionally, solid particles of micrometer or millimeter magnitudes are mixed in the base liquid. Although the solid additives may improve heat transfer coefficient, practical application are limited because of the fact that the micrometer and/or millimeter-sized particles settle rapidly, clog flow channels, erode pipelines and cause severe pressure drops. The concept of nanofluids refers to a new kind of heat transport fluids by suspending nanoscaled metallic or nonmetallic particles in base fluids. Energy transport of the nanofluid is affected by the properties and dimension of nanoparticles as well as the solid volume fraction. Some experimental investigations have revealed that the nanofluids have remarkably higher thermal conductivities than those of conventional pure fluids and shown that the nanofluids have great potential for heat transfer enhancement [4–7]. Compared with the existing techniques for enhancing heat transfer by adding millimeter and/or micrometer-sized particles in fluids, nanofluids are expected to be ideally suited for practical application with incurring little or no penalty in pressure drop because the nanoparticles are so small that the nanofluid behaves like a pure fluid.

To apply the nanofluid to practical heat transfer processes, more studies on its flow and heat transfer feature are needed. Pak and Cho [8] performed experiments on turbulent friction and heat transfer behaviors of two kinds of the nanofluids. In their study, γ-Al_2O_3 and TiO_2 were dispersed in water, and the experimental results showed that the Nusselt number of the dispersed fluids increases with increasing the volume fraction of the suspended solid particles and the Reynolds number. Lee and Choi [9] applied the nanofluid as the coolant to a microchannel heat exchanger for cooling crystal silicon mirrors used in high-intensity X-ray sources and pointed out that the nanofluid dramatically enhances cooling rates compared with the conventional water-cooled and liquid-nitrogen-cooled microchannel heat exchangers.

It is expected that the main reasons of heat transfer enhancement of the nanofluids may be from intensification of turbulence or eddy, suppression or interruption of the boundary layer as well as dispersion or backmixing of the suspended nanoparticles, besides substantial augmentation of the thermal conductivity and the heat capacity of the fluid. Therefore, the convective heat transfer coefficient of the nanofluids is a function of properties, dimension and volume fraction of suspended nanoparticles as well as the flow velocity. The conventional convective heat transfer correlation of the pure fluid isn't applicable to the nanofluid. To understand the mechanism of heat transfer enhancement of nanofluids and to accelerate practical applications of the nanofluids, more investigations are needed on fundamental features of convective heat transfer and flow performance of the nanofluids. Based on one of the previous papers [10], this paper is aimed at studying the single-phase flow and heat transfer performance of the nanofluid in tubes for the turbulent flow and developing heat transfer correlation for the experimental data.

2 Experimental System

An experimental rig is built to study the flow and convective heat transfer feature of the nanofluid flowing in a tube. As shown schematically in Fig. 1, the experimental system mainly includes a reservoir tank, a pump, a pipe line, a test section, a cooler, and a fluid collection tank.

The reservoir tank of 5 Liter is manufactured of polymethylmethacrylate to reserve the nanofluid and to monitor the dispersion behavior and stability of the nanofluid. The cooler of 5.2 KW cooling capacity is used to keep a constant temperature of the nanofluid at the inlet of the test section. The flow rate is controlled with two adjusting valves, one at the main flow loop and the other at the by-bass line. A three-way valve is installed at the end of the main flow loop and it provides an access for the nanofluid from the reservoir tank into the fluid collection tank to measure the mass flow rate of the nanofluid. The test section is a straight brass tube of the inner diameter of 10 mm and the length of 800 mm. Eight thermocouples (K-type) are mounted at different places of the heat transfer test section to measure the wall temperatures and other two thermocouples are respectively located at the entrance and exit of the test section to read the bulk temperatures of the nanofluid. To obtain a constant-heat-flux boundary condition, the heat transfer test section is heated electrically by a DC power supply capable of delivering a maximum of 3.5 KW. The test section is thermally isolated from its upstream and downstream sections by plastic bushings to minimize the heat loss resulting

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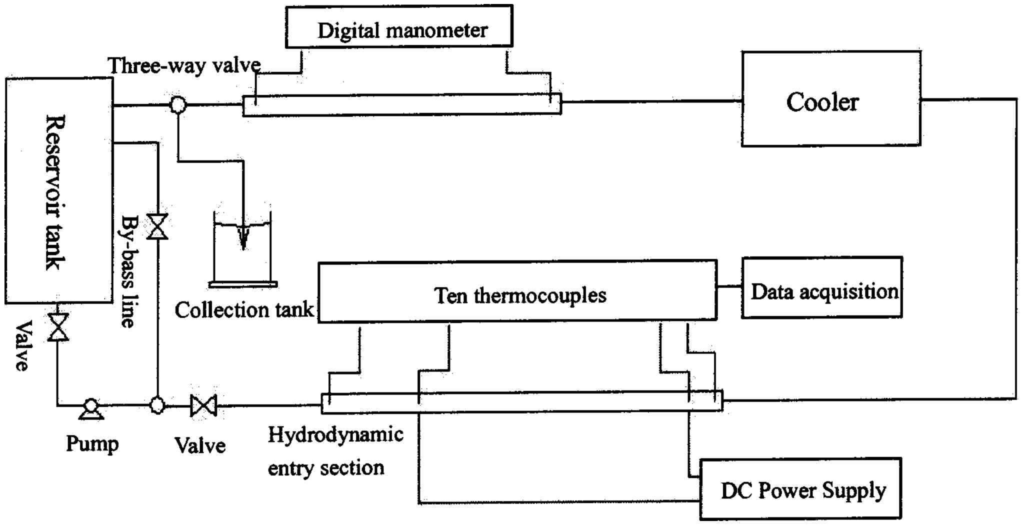

Fig. 1 The experimental system of the convective heat transfer feature for the nanofluid

from axial heat conduction. In order to minimize the heat loss from the test section to the ambient, the whole test section is thermally isolated on the outside with a layer of expanded pearl powder and a vacuum casing tube. The hydrodynamic entry section is long enough to accomplish fully developed flow at the entrance of the heat transfer test section. Two pressure transducers are installed at the both ends of the test section to measure the pressure drop of nanofluids.

During experimental runs, the tube wall temperatures, inlet and outlet temperatures of the sample nanofluid, mass flow rates and electric power inputs as well as the static pressures are measured. The measured Nusselt numbers of the nanofluid within the fully developed turbulent flow region are obtained from the following correlation

$$h_{nf}=\frac{q}{T_w-T_f} \tag{1}$$

$$\mathrm{Nu}_{nf}=\frac{h_{nf}D}{k_{nf}} \tag{2}$$

The transport properties of the nanofluid are calculated by using the mean value of the temperatures of the nanofluid at the inlet and outlet. Before measuring the convective heat transfer coefficient of nanofluids, the reliability and accuracy of the experimental system are estimated by using water as the working fluid. The results of estimation experiment are compared with the calculated values obtained from the well-known Dittus-Boelter equation [11]

$$\mathrm{Nu}=0.023\,\mathrm{Re}^{0.8}\mathrm{Pr}^{0.4} \tag{3}$$

As shown in Fig. 2, the good coincidence between the experimental results and the calculated values for water reveals that the precision of the experimental system is considerably high. The uncertainty of the experimental system is less than 4 percent.

Uncertainty of the experimental data may result from the measuring errors of parameters such as heat flux, temperature as well as flow rate. It can be defined as:

$$\left(\frac{\delta \mathrm{Nu}}{\mathrm{Nu}}\right)=\sqrt{\left(\frac{\Delta q}{q}\right)^2+\left(\frac{\Delta T}{T}\right)^2+\left(\frac{\Delta u}{u}\right)^2} \tag{4}$$

The error term $(\Delta q/q)$ arises from reading errors of direct electric current and voltage as well as heat loss from the experimental system to the ambient. The calibration results of energy balance revealed that experimental uncertainty of the heat flux is less than 3 percent in this experiment. The thermocouple is calibrated by using the thermostat and the measuring precision is of 0.1 percent. Flow rates are measured directly by weighing fluid and the measuring error of the flow rate is less than 1 percent. Therefore, uncertainty of the experimental data is less than 4 percent.

3 Convective Heat Transfer Experiment and Correlation

The sample nanofluids are prepared by mixing the nanostructured Cu particles below 100 nm diameter and deionized water. The details of preparation procedure of nanofluids are referred to the previous paper [7]. The nanofluids with different particle volume fractions are used in the experiment to investigate the effect of the nanoparticle concentration on the enhanced heat transfer performances of nanofluids, in which 0.3 percent, 0.5 percent, 0.8 percent, 1.0 percent, 1.2 percent, 1.5 percent, and 2.0 percent volume fraction Cu-water nanofluids are involved. Nanofluids with higher volume fractions of nanoparticles may be limited in practical application and consume much more solid particles. The Reynolds number Re varies between the range 10,000∼25,000.

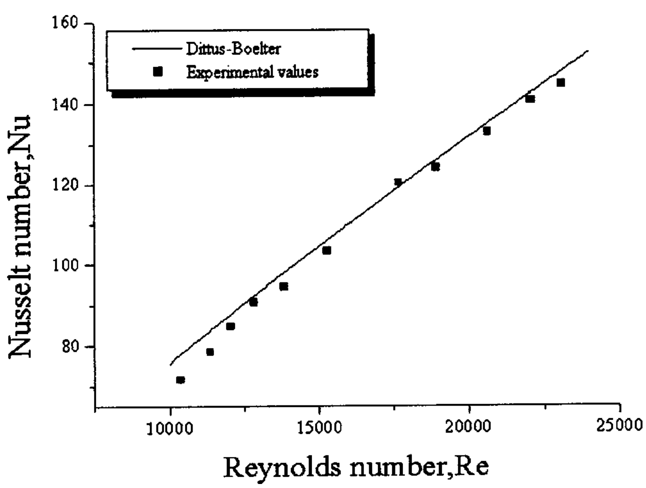

Fig. 2 Comparison between the measured Nusselt numbers of water and the calculated values with Eq. (3)

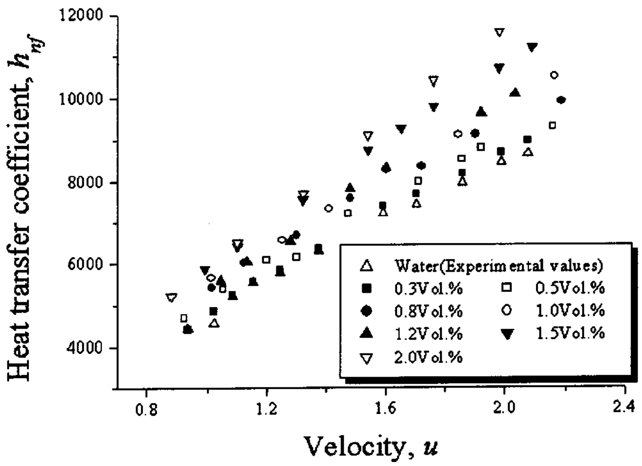

Fig. 3 Variation of heat transfer coefficient with velocity in the turbulent flow

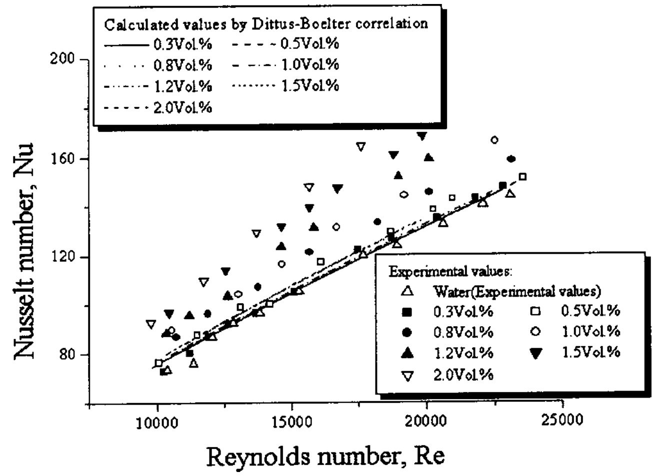

Fig. 4 The Nusselt numbers of nanofluids with the Reynolds numbers and the predicated values from the Dittus-Boelter correlation

As shown in Fig. 3, the convective heat transfer coefficient of the nanofluid increases with the flow velocity as well as the volume fraction of nanoparticles and it is larger than that of the base liquid (water) under the same flow velocity. It may be instructive to point out that proper selection of the particle volume fraction and of the couple pair of solid particles and base liquid are important for applying nanoparticles to heat transfer enhancement. In some cases, the viscosity of the dispersed fluid sharply increases with increasing the particle volume fraction and becomes much higher than that of the base liquid, so that higher volume fraction of the solid particles may suppress heat transfer enhancement of the suspension. Pak and Cho [8] studied heat transfer process of the suspensions consisting of water and metallic oxide particles such as γ-alumina (Al_2O_3) with mean diameter of 13 nm and titanium dioxide (TiO_2) with mean diameter of 27 nm and found that the convective heat transfer coefficient of the suspensions at a volume concentration of 3 percent was 12 percent smaller than that of pure water when compared under the condition of constant average velocity. The reason may be that both the suspensions have much higher viscosities than that of water, which suppresses flow turbulence. While preparing the nanofluid, therefore, it may be of importance to select the volume fraction, dimensions and material properties of the nanoparticles suspended in the base liquid.

Figure 4 illustrates variation tendency of the Nusselt numbers of the sample nanofluid with the volume fraction of nanoparticles and the Reynolds number Re. The experimental results indicate that the suspended nanoparticles remarkably improve heat transfer performance of the base fluid. Compared with water, the Nusselt number of the nanofluid is increased more than 39 percent for the nanofluid with the volume fraction 2.0 percent of Cu nanoparticles. The experimental results also indicate that the heat transfer feature of a nanofluid remarkably increases with the volume fraction of nanoparticles. The particle volume fraction is one of the main factors affecting the Nusselt numbers of the nanofluid. While the volume fraction of the Cu nanoparticles increases from 0.5 percent to 2.0 percent, for example, the ratio of the Nusselt number of the Cu-water nanofluid to that of water varies from 1.06 to 1.39 under the same Reynolds number.

As mentioned before, the nanofluid behaves more like a fluid than the conventional solid-fluid mixtures in which relatively larger particles with micrometer or millimeter orders are suspended. But the nanofluid is a two-phase fluid in nature and has some common features of the solid-fluid mixtures. The effects of several factors such as gravity, Brownian force, and friction force between the fluid and ultrafine solid particles, the phenomena of Brownian diffusion, sedimentation, and dispersion may coexist in the main flow of a nanofluid. This means that the slip velocity between the fluid and the suspended nanoparticles may not be zero, although the particles are ultrafine. Random movement of the suspended nanoparticles increases energy exchange rates in the fluid. The dispersion will flatten temperature distribution and makes the temperature gradient between the fluid and wall steeper, which augments heat transfer rate between the fluid and the wall. The enhanced heat transfer by the nanofluid may result from the following two aspects [10]: One is that the suspended particles increase the thermal conductivity of the two-phase mixture; another is that chaotic movement of ultrafine particles accelerates energy exchange process in the fluid. The comparison between the experimental results and the predicated values from the Dittus-Boelter correlation is shown in Fig. 4. The predicted results are obtained in such a way that all the transport properties involved in the Dittus-Boelter correlation are respectively replaced by the effective conductivity, viscosity and diffusivity of the nanofluid. Clearly, greater deviation between the experimental results and the predicted ones from the conventional correlation exists and such deviation increases with increasing the volume fraction of the suspended nanoparticles and the Reynolds number. For example, the discrepancy between both them is over 30 percent for $\mathrm{Re}=17{,}600$ and $\phi=2.0$ percent. The Dittus-Boelter correlation failed to predict the dependency of the Nusselt number of the nanofluid on the volume fraction of nanoparticles because this correlation is valid only for the single-phase flow. It is improper to apply the Dittus-Boelter correlation to prediction the Nusselt number for the nanofluid even if the effective transport properties of the nanofluid are used, especially for the case that the volume fraction of the nanoparticles is larger than 0.5 percent. The values of the Nu number calculated from the Dittus-Boelter correlation just slightly change (as illustrated by the solid and dotted lines in Fig. 4), although the volume fraction of the nanoparticles varies from 0.3 percent to 2.0 percent. In fact, the experimental data of the Nu number do remarkably vary with the volume fraction of the nanoparticles (as shown by all the scattered points in Fig. 4).

A new approach needs to be tried to develop the heat transfer correlation for the nanofluid. In general, the Nusselt number Nu of a nanofluid may be expressed as follows:

$$\mathrm{Nu}_{nf}=f\left(\mathrm{Re},\mathrm{Pr},\frac{k_d}{k_f},\frac{(\rho c_p)_d}{(\rho c_p)_f},\phi,\text{dimensions and shape of particles}\right) \quad (5)$$

Table 1 The effective thermal conductivity and viscosity of the sample nanofluid

Transport properties	Volume fraction of nanoparticles						
	0.3%	0.5%	0.8%	1.0%	1.2%	1.5%	2.0%
k_{nf} (W/m·°C)	0.6054	0.615	0.6252	0.6306	0.633	0.663	0.702
$\upsilon_{nf}\times 10^6$ (m^2/s)	0.91	0.915	0.945	0.96	1.012	1.044	1.125

In the light of analysis and derivation presented in the previous paper [10], the following formula is proposed to correlate the experimental data for the nanofluid:

$$\mathrm{Nu}_{nf}=c_1(1.0+c_2\phi^{m_1}\mathrm{Pe}_d^{m_2})\mathrm{Re}_{nf}^{m_3}\mathrm{Pr}_{nf}^{0.4} \tag{6}$$

Compared with the heat transfer correlation for conventional single-phase flow, the volume fraction ϕ of suspended nanoparticles and the Peclet number are involved in the above expression. The Peclet number Pe describes the effect of thermal dispersion caused by microconvective and microdiffusion of the suspended nanoparticles. The case $c_2=0$ refers to zero thermal dispersion, which namely corresponds to the case of the pure base fluid. The particle Peclet number Pe_d in expression (6) is defined as

$$\mathrm{Pe}_d=\frac{u_m d_p}{\alpha_{nf}} \tag{7}$$

The Reynolds number of the nanofluid is defined as

$$\mathrm{Re}_{nf}=\frac{u_m D}{\upsilon_{nf}} \tag{8}$$

The Prandlt number Pr of the nanofluid is defined as

$$\mathrm{Pr}_{nf}=\frac{\upsilon_{nf}}{\alpha_{nf}} \tag{9}$$

To calculate this parameter, the thermal diffusivity of the nanofluid α_{nf} is defined as

$$\alpha_{nf}=\frac{k_{nf}}{(\rho c_p)_{nf}}=\frac{k_{nf}}{(1-\phi)(\rho c_p)_f+\phi(\rho c_p)_d} \tag{10}$$

Where, the thermal conductivity k_{nf} and the viscosity υ_{nf} are experimentally obtained and the experimental procedure is described in detail in reference [6]. Some experimental data are listed in Table 1.

From the whole ensemble of experimental data, the coefficient c_1 and c_2 as well as the exponent m_1, m_2, and m_3 in expression (6) can be determined by a proper data-reduction procedure. For a given flow velocity, a set of all these coefficients and exponents is fit for predicting heat transfer performance of nanofluids with different volume fractions of suspended nanoparticles. For a variety of volume fraction of suspended nanoparticles, for example, the Nusselt number Nu for the turbulent flow of nanofluids inside a tube are obtained as follows:

$$\mathrm{Nu}_{nf}=0.0059(1.0+7.6286\phi^{0.6886}\mathrm{Pe}_d^{0.001})\mathrm{Re}_{nf}^{0.9238}\mathrm{Pr}_{nf}^{0.4} \tag{11}$$

Figure 5 depicts the curves of the theoretical predictions of convective heat transfer coefficients of nanofluids from correlation (11). Obviously, there exists good coincidence between the results calculated from this correlation and the experimental ones. The discrepancy between both them falls below 8 percent. It reveals that formula (6) correctly incorporates the main factors of affecting heat transfer process of the nanofluid into in a simpler form and can be used to correlate experimental data of heat transfer coefficient of the nanofluid. Once all the coefficients and exponents in formula (6) have been determined by correlating experimental data of heat transfer for nanofluids, therefore, this formula can be used to predict the convective heat transfer coefficient for such suspensions with different volume fractions of nanoparticles in the turbulent flow.

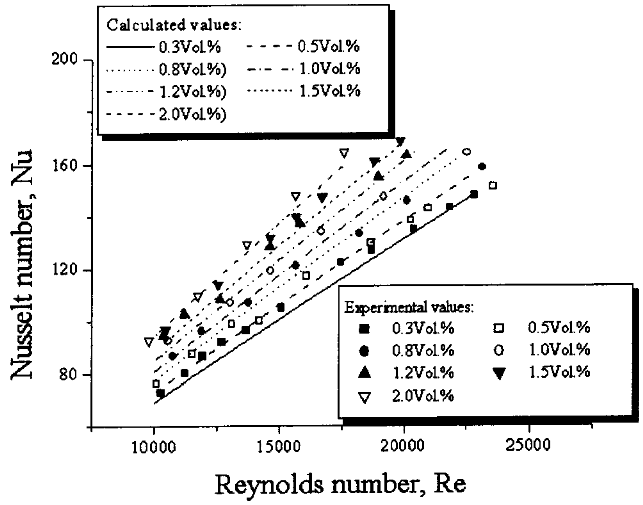

Fig. 5 Comparison between the measured data and the calculated values from correlation (11) for turbulent flow

4 Pressure Drop Experiment

It is necessary to learn the flow resistance of nanofluids besides the heat transfer enhancement feature in order to apply the nanofluid to practical cases. The pressure drops of the dilute suspensions consisting of water and Cu-nanoparticles in a tube are experimentally measured for the turbulent flow.

Four sample nanofluids with the volume fractions of nanoparticles 1.0 percent, 1.2 percent, 1.5 percent, and 2.0 percent are used in pressure drop test. Figure 6 illustrates the friction factors as a function of the Reynolds number for the turbulent flow. The friction factor of the pure water is also shown as a solid line in the figures. The friction factor is defined as

$$\lambda_{nf}=\frac{P_{nf}D}{L}\frac{2g}{u_m^2} \tag{12}$$

Obviously, the friction factors of the dilute nanofluids are almost equal to those of water under the same Reynolds number. Compared with water, no significant augmentation in pressure drop for the nanofluid is found in all runs of the experiment,

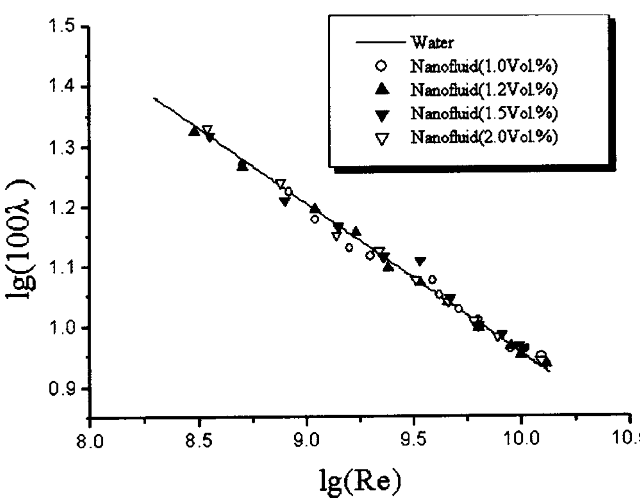

Fig. 6 The friction factors of nanofluids for the turbulent flow

which reveals that dilute nanofluids will not cause extra penalty in pump power. It implies that the friction factor correlation for the single-phase flow can be extended to the dilute nanofluids.

5 Conclusions

The convective heat transfer feature and flow performance of Cu-water nanofluids in a tube have experimentally been investigated. The suspended nanoparticles remarkably enhance heat transfer process and the nanofluid has larger heat transfer coefficient than that of the original base liquid under the same Reynolds number. The heat transfer feature of a nanofluid increases with the volume fraction of nanoparticles.

By considering the microconvection and microdiffusion effects of the suspended nanoparticles, a new type of the convective heat transfer correlation for nanofluids in a tube has been proposed as $Nu_{nf}=c_1(1.0+c_2\phi^{m_1}Pe_d^{m_2})Re_{nf}^{m_3}Pr_{nf}^{0.4}$. This correlation correctly takes the main factors of affecting heat transfer of the nanofluid into account.

On the other hand, the friction factor for the dilute nanofluids consisting of water and Cu-nanoparticles is approximately the same as that of water. The nanofluid with the low volume fraction of the suspended nanoparticles incurs almost no extra penalty of pump power.

Acknowledgment

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Nomenclature

d = nanoparticle diameter, m
D = inner diameter of the tube, m
g = acceleration of gravity, m/s^2
h = heat transfer coefficient, W/m^{2}K
k = thermal conductivity of the sample nanofluid, W/mK
L = length of the pressure drop test tube, m
Nu = Nusselt number
P = pressure drop, Pa
Pe = Peclet number
Pr = Prandtl number
q = heat flux, W/m^2
Re = Reynolds number
T = temperature, °C
u = mean velocity, m/s
ϕ = volume fraction
α = thermal diffusivity, m^2/s
υ = viscosity, m^2/s

Subscript

d = particle
f = fluid
in = inlet
m = mean
out = outlet
nf = nanofluid
w = tube wall

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