Convective heat transfer and flow characteristics of Cu-water nanofluid

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Abstract An experimental system is built to investigate convective heat transfer and flow characteristics of the nanofluid in a tube. Both the convective heat transfer coefficient and friction factor of Cu-water nanofluid for the laminar and turbulent flow are measured. The effects of such factors as the volume fraction of suspended nanoparticles and the Reynolds number on the heat transfer and flow characteristics are discussed in detail. The experimental results show that the suspended nanoparticles remarkably increase the convective heat transfer coefficient of the base fluid and show that the friction factor of the sample nanofluid with the low volume fraction of nanoparticles is almost not changed. Compared with the base fluid, for example, the convective heat transfer coefficient is increased about 60% for the nanofluid with 2.0 vol% Cu nanoparticles at the same Reynolds number. Considering the factors affecting the convective heat transfer coefficient of the nanofluid, a new convective heat transfer correlation for nanofluid under single-phase flows in tubes is established. Comparison between the experimental data and the calculated results indicate that the correlation describes correctly the energy transport of the nanofluid.

Keywords: nanofluid, convective heat transfer, correlation, friction factor.

With increasing heat transfer rate of the heat exchange equipment, the conventional process fluid with low thermal conductivity can no longer meet the requirements of high-intensity heat transfer. Low thermal property of heat transfer fluid is a primary limitation to the development of high compactness and effectiveness of heat exchangers. An effective way of improving the thermal conductivity of fluids is to suspend small solid particles in the fluids[1]. Traditionally, solid particles of micrometer or millimeter magnitudes were mixed in the base liquid. Although the solid additives may improve heat transfer coefficient, practical uses are limited because 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 nano-sized metallic or nonmetallic particles in base fluids. Some experimental investigations[2–5] 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. Nanofluids are more suited for practical application than the existing techniques for enhancing heat transfer by adding millimeter and/or micrometer-sized particles in fluids. It incurs 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\cite{6} performed experiments on turbulent friction and heat transfer behavior of two kinds of the nanofluids. In their study, $\gamma$-Al$_2$O$_3$ with mean diameter of 13 nm and TiO$_2$ with mean diameter of 27 nm were dispersed in water, and the experimental results showed that the suspended nanoparticles increased convective heat transfer coefficient of the fluid. The Nusselt number of the nanofluids was found to increase with the increasing volume fraction and Reynolds number. Lee and Choi\cite{7} 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 enhanced cooling rates compared with the conventional water-cooled and liquid-nitrogen-cooled microchannel heat exchangers. The cooling capability of 30MW·m$^{-2}$ was achieved.

It is expected that the heat transfer enhancement of the nanofluids may result from intensification of turbulence or eddy, suppression of the boundary layer as well as dispersion or backmixing of the suspended nanoparticles, besides substantial augmentation of the thermal conductivity of the fluid. Therefore, the convective heat transfer coefficient of the nanofluids is a function of properties, dimension and volume fraction of suspended nanoparticles, and the flow velocity. The conventional convective heat transfer correlation of the pure fluid is not applicable to the nanofluid. This paper is aimed at investigating convective heat transfer performances of the nanofluid and establishing heat transfer correlation for single-phase flows in tubes.

1 Experimental system of convective heat transfer and flow characteristics for nanofluids

An experimental rig is built to study the convective heat transfer and flow characteristics of the nanofluid flowing in a tube. As shown schematically in fig. 1, the experimental system mainly includes a reservoir tank, a pump, a bypass line, a heat transfer test section, a cooler, a pressure drop test section, a fluid collection tank and so on.

![Fig. 1. The experimental system of the convective heat transfer and flow characteristics for the nanofluid.](image)
The reservoir tank of 5 liter volume is manufactured using polymethylmethacrylate to deposit the nanofluid and monitor the dispersion behavior and stability of the nanofluid. The cooler with 5.2 kW of cooling capacity is used to keep constant temperature 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 bass line. A three-way valve is installed at the end of the main flow loop, which enables the flow to be diverted from the reservoir tank into the fluid collection tank in order to measure the mass flow rate of the nanofluid. Two same flexible vinyl tubes are respectively connected with the reservoir tank and the collection tank in such way that the flow rate measurement procedure would not change the flow rate itself. Two pressure taps were mounted at the end of the pressure drop test section with 12 mm inner diameter and 1m length to measure the pressure drop of nanofluids.

The heat transfer test section is a straight brass tube with an inner diameter of 10mm and a length of 800 mm. Ten thermocouples are mounted at different places of the heat transfer test section to measure the wall temperatures and the fluid bulk temperatures. 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 isolated thermally from its upstream and downstream sections by plastic bushings to minimize the heat loss resulting 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.

During experimental runs, the tube wall temperatures, inlet and outlet temperatures of the sample nanofluid, mass flow rates and electric power inputs are measured. From these data, the convective heat transfer coefficients can be determined

\[ h_{nt} = \frac{q}{T_w - T_f}, \]

where \( q \) is the heat flux of the heat transfer test section, \( T_w \) is the mean tube wall temperature. \( T_f \) is the mean bulk temperature of the fluid, amounting to half the sum of inlet and outlet temperatures, because the difference between inlet and outlet temperatures is small.

Before measuring the convective heat transfer coefficient of nanofluids, the experimental system was calibrated by comparing the measured Nusselt numbers of water at the fully developed turbulent flow with the calculated values using the Dittus-Boelter equation\[8]\]

\[ Nu = 0.023Re^{0.8}Pr^{0.4}. \]

As illustrated in fig. 2, the good coincidence between the experimental results and the calculated values shows that the precision of the experimental system is relatively high. The precision of the experimental system is less than 4%.

2 Convective heat transfer experiment and correlation

The sample nanofluids are prepared by direct mixing of the nanostructured Cu particles with
diameters below 100 nm and deionized water. To prevent the aggregation among nanoparticles, a small amount of fatty acid salt is selected as the dispersant to cover the nanoparticles. 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%, 0.5%, 0.8%, 1.0%, 1.2%, 1.5% and 2.0% 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 in the range of 800—25000. In the present study, the volume fraction of nanoparticles is determined with the mass fraction of nanoparticles.

Fig. 3 and 4 give the convective heat transfer coefficient of the Cu-water nanofluid with the different Reynolds numbers under the laminar and the turbulent flow, respectively. The experiments show that the suspended nanoparticles remarkably increase heat transfer performance of the base fluid, and the Cu-water nanofluid has larger heat transfer coefficient than pure water under the same Reynolds number. Compared with water, for example, the convective heat transfer coefficient of nanofluid is increased about 60% for the nanofluid with 2.0 vol% Cu nanoparticles the same Reynolds number.

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 factors affecting the convective heat transfer coefficient of the nanofluid. For example, the ratio of
the Nusselt number of the nanofluid to that of water at the same Reynolds number varies from 1.06 to 1.6 if the volume fraction of the nanoparticles increases from 0.5% to 2.0%.

As shown in fig. 5, the convective heat transfer coefficient of the nanofluid increases with the flow velocity as well as the volume fraction of nanoparticles and is larger than that of the base liquid (water) at the same flow velocity. But Pak and Cho\[^6\] found that the convective heat transfer coefficient of the suspensions at a volume concentration of 3% was 12% smaller than that of pure water 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. It may be instructive to point out that a proper selection of the particle volume fraction and of the couple pair of solid particles and base liquid is important for applying nanoparticles to heat transfer enhancement. In some cases, the viscosity of the dispersed fluid sharply increases with the increasing 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, 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. Selecting nanoparticles with higher thermal conductivity and larger size is a crucial point.

As mentioned before, the enhanced heat transfer by the nanofluid may result from the following two aspects: One is that the suspended particles increase the thermal conductivity of the two-phase mixture; the other is that chaotic movement of ultrafine particles accelerates energy exchange process in the fluid. The nanofluid behaves more like a fluid than the conventional solid-fluid mixtures in which relatively large 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 particles may not be zero, although the particles are ultrafine. Irregular and random movement of the particles increase energy exchange rates in the fluid, i.e. thermal dispersion takes place in the flow of the nanofluid. The thermal dispersion will flatten temperature distribution and make the temperature gradient between the fluid and wall steeper, augmenting heat transfer rate between the fluid and the wall. Evidently the thermal dispersion plays an important role in heat transfer enhancement. Therefore, it may be improper to correlate
the experimental data of heat transfer for nanofluids with the conventional forms for single-phase fluids.

In general, the Nusselt number $Nu$ of a nanofluid may be expressed as follows:

$$Nu_{nf} = f \left( Re_{nf}, Pr_{nf}, \frac{k_d}{k_f}, \frac{(\rho c_p)_d}{(\rho c_p)_f}, \phi, \text{dimensions and shape of particles} \right),$$

(3)

where $Re_{nf}$ is the Reynolds number of the nanofluid, $Pr_{nf}$ is the Prandtl number of the nanofluid, $k_d$ is the thermal conductivity of the nanoparticle, $k_f$ is the thermal conductivity of the base fluid, $(\rho c_p)_d$ is the heat capacity of the nanoparticle, $(\rho c_p)_f$ is the heat capacity of the base fluid, and $\phi$ is the volume fraction of the nanoparticle.

In the light of analysis and derivation presented by Xuan\textsuperscript{[9]}, the following formula is proposed to correlate the experimental data for the nanofluid:

$$Nu_{nf} = c_1 \left( 1.0 + c_2 \phi^m Pe_d^{m_2} \right) Re_{nf}^{m_1} Pr_{nf}^{0.4},$$

(4)

where the particle Peclet number $Pe_d$ is defined as

$$Pe_d = \frac{u_m d_p}{\alpha_{nf}}.$$

(5)

The Reynolds number and the Prandtl number of the nanofluid are defined respectively as

$$Re_{nf} = \frac{u_m D}{\nu_{nf}},$$

(6)

$$Pr_{nf} = \frac{\nu_{nf}}{\alpha_{nf}}.$$

(7)

The thermal diffusivity of the nanofluid $\alpha_{nf}$ in expressions (5) and (7) is defined as

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

(8)

where $u_m$ is the mean velocity, $D$ is inner diameter of the tube, $\nu_{nf}$ is the viscosity of the nanofluid, $d_p$ is the mean diameter of the nanoparticle. $k_{nf}$ is the thermal conductivity of the nanofluid. The thermal conductivity $k_{nf}$ and the viscosity $\nu_{nf}$ can be found from the previous paper\textsuperscript{[4]}.

Compared with the heat transfer correlation for conventional single-phase flow, the volume fraction $\phi$ 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 just corresponds to the case of the pure base fluid.

With an ensemble of experimental data of $Nu_{nf}$ vs $\phi$, $Pe_d$, $Re_{nf}$ and $Pr_{nf}$, the coefficients $c_1$ and $c_2$ as well as the exponents $m_1$, $m_2$ and $m_3$ in the above formula are found for either the laminar or turbulent flow by data reduction.
\[ Nu_{\text{nf}} = 0.4328 \left(1.0 + 11.285 \phi^{0.754} P_{\text{e_d}}^{0.218}\right) R_{\text{e_nf}}^{0.333} P_{\text{r_nf}}^{0.4} \quad \text{(for the laminar flow),} \tag{9} \]

\[ Nu_{\text{nf}} = 0.0059 \left(1.0 + 7.6286 \phi^{0.6886} P_{\text{e_d}}^{0.001}\right) R_{\text{e_nf}}^{0.9238} P_{\text{r_nf}}^{0.4} \quad \text{(for the turbulent flow).} \tag{10} \]

Figs. 6 and 7 give the calculated results of the sample nanofluids with expressions (9) and (10), respectively. Comparison shows that the calculated results are in good coincidence with the experimental results with only 8% discrepancy revealing that expression (4) correctly takes into account the main factors affecting heat transfer of the nanofluid and can be used to predict heat transfer coefficient of the nanofluid.

Fig. 6. Comparison between the measured data and the calculated values for laminar flow. Calculated values: 1, 0.3 vol%; 2, 0.5 vol%; 3, 0.8 vol%; 4, 1.0 vol%; 5, 1.2 vol%; 6, 1.5 vol%; 7, 2.0 vol%. Experimental values: 1, 0.3 vol%; 2, 0.5 vol%; 3, 0.8 vol%; 4, 1.0 vol%; 5, 1.2 vol%; 6, 1.5 vol%; 7, 2.0 vol%.

Fig. 7. Comparison between the measured data and the calculated values for turbulent flow. Calculated values: 1, 0.3 vol%; 2, 0.5 vol%; 3, 0.8 vol%; 4, 1.0 vol%; 5, 1.2 vol%; 6, 1.5 vol%; 7, 2.0 vol%. Experimental values: 1, 0.3 vol%; 2, 0.5 vol%; 3, 0.8 vol%; 4, 1.0 vol%; 5, 1.2 vol%; 6, 1.5 vol%; 7, 2.0 vol%.

3 Pressure drop experiment of nanofluids

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 both the laminar and turbulent flow.

Four sample nanofluids with the volume fractions of nanoparticles 1.0%, 1.2%, 1.5% and 2.0% are used in pressure drop tests. Figs. 8 and 9 illustrate the friction factors as a function of the Reynolds number for the laminar and turbulent flow, respectively. The friction factor of the pure water is also shown as a solid line in the figures. The friction factor \( \lambda_{\text{nf}} \) is defined as

\[ \lambda_{\text{nf}} = \frac{P_{\text{nf}} D}{L \ u_m^2}, \tag{11} \]

where \( P_{\text{nf}} \) is the pressure drop of the pressure drop test section, \( L \) is the length of the pressure drop test section, and \( g \) is the acceleration of gravity.
These figures show that the friction factors of the dilute nanofluids are almost equal to those of water under the same Reynolds number and do not increase with the volume fraction of nanoparticles. Compared with water, no significant augmentation in pressure drop for the nanofluid is found in all runs of the experiment, which reveals that dilute nanofluids will not cause extra penalty in pump power.

4 Conclusion

The convective heat transfer feature of Cu-water nanofluids were investigated experimentally in a tube. The experimental results show that the suspended nanoparticles remarkably increase heat transfer performance of the base fluid and the nanofluid has larger heat transfer coefficient than pure water under the same Reynolds number. The heat transfer feature of a nanofluid increases with the volume fraction of nanoparticles. The friction factors of the nanofluids coincide well with those of the water in the pressure drop test, because the nanoparticles are so small that a suspension with nanoparticles behaves like a pure fluid. The nanofluid with the low volume fraction incurs almost no augmentation of pressure drop.

Considering some factors affecting convective heat transfer characteristics of nanofluids, such as the flow velocity, the transport properties, the volume fraction of nanoparticle, the microconvective and microdiffusion of the nanoparticles, a new convective heat transfer correlation for nanofluid suspending the metal nanoparticles under single-phase flows in tubes has been established as

\[ Nu_{nf} = c_1 \left( 1.0 + c_2 \phi^{m_1} Pe_d^{m_2} Re_{nf}^{m_3} Pr_{nf}^{0.4} \right) \]

Comparison between the experimental data and the calculated results indicates that the correlation correctly takes into account the main factors that affect heat transfer of the nanofluid and can be used to predict heat transfer coefficient of the nanofluid.

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References