RESEARCH ON HEAT TRANSFER CHARACTERISTICS OF FLOW IN TUBE OF WATER-BASED NANOFLUIDS

by

Pengfei CHEN, Zhuangzhuang JIA, Zhumei LUO*, Shan Qing*, and Xiaoyan HUANG

Faculty of Metallurgical and Energy Engineering,
Kunming University of Science and Technology, Kunming, China

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In this paper, the characteristics of forced convection heat transfer in water-based nanofluids are studied by means of experimental and theoretical analysis. Nusselt number of nanofluids were calculated by changing the volume fraction and the type of nanoparticles in the tube. The effects of Reynolds number and the volume fraction of nanoparticles on the forced convection heat transfer were studied. An exergy analytical model was established for the laminar heat transfer of nanofluid under the condition of constant heat flow. At the same Reynolds condition, the friction entropy production of the flow and heat transfer process in the tube increases with the addition of nanoparticles, and the heat transfer entropy production decreases at the same time. However, the magnitude of friction entropy production is only 10^{-6}, which is negligible compared with the heat transfer entropy production. Therefore, in general, the loss of nanofluids is lower than that of pure water and for nanofluids, the exergy loss of hybrid nanofluid is lower than that of single nanofluid at the same volume fraction.

Key words*: nanofluids, enhanced heat transfer, flow and heat transfer in tube, exergy analysis*

Introduction

Nanofluids refer to a new type of heat transfer working medium formed by adding nanoscale metal or non-metal particles to the basic fluid in a certain way and proportion [1]. According to Maxwell's [2] theory, the thermal conductivity of the solid-liquid two-phase mixture has a great relationship with the thermal conductivity of the solid particles, and the solid particles with larger thermal conductivity will improve the thermal conductivity of the mixture. Generally, adding nanoparticles into the liquid can significantly improve the thermal conductivity of the working fluid. The poor stability of millimeter and micrometer suspensions is easy to deposit on the surface of heat exchanger and cause blockage, and the heat exchange channel will be worn during the flow process, while the unique small size effect of nanofluids significantly improves this problem [3]. Therefore, in order to make nanofluids play an important role in industrial heat exchange equipment successfully, it is necessary to carry out experiments and study its heat transfer characteristics.

Forced convective heat transfer in tube is a widely used heat transfer method. It is necessary to study the heat transfer characteristics of nanofluids in this heat transfer mode. The

* Corresponding authors, e-mail: luozhumei@163.com; 326720228@qq.com

heat transfer effect of nanofluids is the result of Brownian motion [4, 5], solid-liquid interface properties [6], particle agglomeration [7] and many other factors. At present, researchers have done a lot of research on forced convection and heat transfer in nanofluid tubes, mainly including the influence of the types of nanoparticles, the proportion of particles in the total volume, the temperature of working medium, Reznolds number, heat flow density and other factors on the convective heat transfer coefficient and Nusselt number [8-13].

Hybrid nanofluid is composed of two or more nanoparticles dispersed in water, ethanol and other conventional fluids. The combination of different particles and basic fluids is expected to show better performance than single nanofluid, but the conclusions are not uniform: some researchers think that the thermophysical properties of hybrid nanofluids are better than that of single particle nanofluids [14, 15], and some scholars' experiments show that the comprehensive performance of hybrid nanofluids is not as good as that of single particle nanofluids [16]. At present, the research of hybrid nanofluids mainly focuses on the thermal conductivity, viscosity, density and specific heat of the working fluid [17-19]. Some scholars have also conducted heat transfer experiments on hybrid nanofluids [20-22].

However, the current research mainly focuses on the heat transfer performance of single nanofluids. Jasim *et al*. [23] investigated the improvement of thermal performance of heat exchanger by Al_2O_3-water nanofluid. The experimental results show that the heat transfer and heat transfer efficiency increase with the increase of the volume concentration and flow rate of nanofluids. Some researchers have calculated the exergy efficiency and analyzed the entropy generation of nanofluids. Selimefendigil and Oztop [24] studied the exergy performance of VCR system with TiO_2 nanoparticles. When nanoparticles added, total irreversibility was reduced and it was significant for higher particle volume fraction. Khan *et al.* [25] explored the radiative nanomaterial flow of Ree-Eyring fluid subject to stretchable surface with entropy generation.

In this paper, different volume fractions of Al_2O_3-water nanofluid, CuO-water nanofluids, and Al_2O_3-CuO-water hybrid nanofluid are prepared. The characteristics of forced convection heat transfer in water-based nanofluids are studied by means of experimental and theoretical analysis. The forced convection heat transfer experiment platform of nanofluid in tube was built and the experiment was carried out. The effects of Reynolds number and nanoparticles volume fraction on the forced convection heat transfer in tube were studied. Adding nanoparticles into pure water can enhance the effect of fluid-flow and heat transfer. Based on the experimental data, calculation and analysis are carried out to explore the effect of fluid heat transfer and different parameters on the heat transfer effect. For example, Nusselt number, Reynolds number, the volume fraction of nanoparticles on the forced convection heat transfer, the exergy analytical model and the exergy loss of hybrid nanofluid were studied. Which revealed the nanoparticles can effectively reduce the exergy loss.

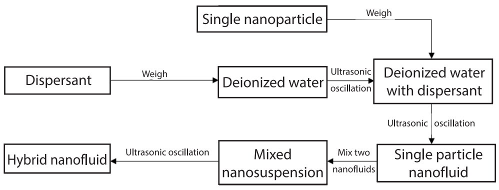


Figure 1. Two-step preparation of hybrid nanofluids

Experimental device and method

At present, the preparation of nanofluids includes *one-step* and *two-step* [26]. *Two-step* method was used to prepare the nanofluids needed in the experiment. The SDBS (sodium dodecyl benzene sulfonate, common anionic surfactants) was used as dispersant to improve the stability of nanofluids. In this paper, the mass fraction of dispersant added to nanofluids is 0.05% [27]. The preparation method of the hybrid nanofluids is shown in fig. 1.

A nanofluidic convection heat transfer experimental platform was built to study the nanofluidic convective heat transfer mechanism and influencing factors by changing the nanofluid volume fraction and flow rate parameters. The experimental table of flow heat transfer in water-based nanofluidic tubes is shown in fig. 2. The P, Q, and T are the pressure, heating power and temperature of the experimental section, respectively. The research content and parameter settings are shown in tab. 1.

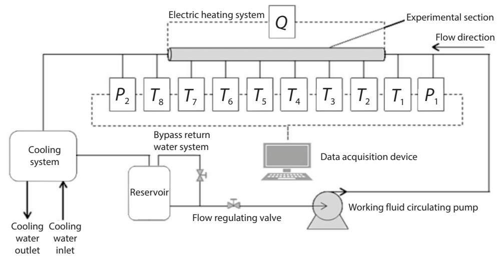


Figure 2. Schematic diagram of the flow heat transfer experimental platform in the tube for nanofluids

Table 1. Parameter setting

Research contents	Parameter setting
Nanoparticles	Al_2O_3,CuO, and the mixture of the two
Volume fraction	0.1%, 0.2%,0.3%
Reynolds number	1100-2300
Inlet temperature	16 °C
Heat flow	500 W, 1000 W, 2000 W

In the experimental section, constant heat flux can be obtained by electric heating. The effective heat flux per unit area can be expressed:

$$q = \frac{Q}{\pi D_\mathrm{i} L} = \frac{Q_\mathrm{total} - Q_\mathrm{loss}}{\pi D_\mathrm{i} L} \tag{1}$$

where D_i [m] is the inner diameter of the pipe, L [m] – the length of the pipe, Q [W] – the effective power, and Q_total [W] – the heating power, adjusted by the voltage regulator:

$$Q_\mathrm{total} = UI \tag{2}$$

where U [V] is the voltage at both ends of the electric heating rod, I [A] – the current of the electric heating rod, and Q_loss [W] is the convection and radiation heat loss between the insulation layer and the environment, calculated as 3% of the total power.

The local flow heat transfer coefficient at a certain position in the pipe is calculated:

$$h(x) = \frac{q}{T_\mathrm{w,i}(x) - T_\mathrm{f}(x)} \tag{3}$$

where $T_\mathrm{w,\,i}(x)$ [K] is the temperature of a certain inner wall in the pipe and $T_\mathrm{f}(x)$ [K] – the temperature of a certain fluid in the pipe.

The temperature of the inner wall of a pipe can be calculated by the temperature of the outer wall:

$$T_\mathrm{w,i}(x) = T_\mathrm{w,o}(x) - q\frac{\ln\dfrac{D_\mathrm{o}}{D_\mathrm{i}}}{2\pi Lk} \tag{4}$$

where $T_\mathrm{w,o}(x)$ [K] is the outer wall temperature of the tube at the corresponding position, measured by the thermocouple, D_o [m] – the outer diameter of the tube, and k – the thermal conductivity of the tube. The fluid temperature in the pipe is calculated:

$$T_f(x) = T_{nf,in} + \frac{q\pi D_i x}{\rho_{nf} c_{p,nf} u A} \tag{5}$$

where x [m] is the distance between the pipe flow direction and the inlet, u [ms^{-1}] – the fluid velocity, and A [m^2] – the cross-sectional area of the pipe. According to the aforementioned formula, the local Nusselt number at a certain position in the pipe can be obtained:

$$\text{Nu}(x) = \frac{h(x)D_i}{k_{nf}} \tag{6}$$

The average heat transfer coefficient in the tube can be calculated:

$$h = \frac{q}{T_{w,i} - T_f} \tag{7}$$

where $T_{w,i}$ [K] is the average temperature of the inner wall of the pipe:

$$T_{w,i} = T_{w,o} - q\frac{\ln\frac{D_o}{D_i}}{2\pi Lk} \tag{8}$$

where $T_{w,o}$ [K] is the average temperature of the outer wall of the tube and the average value of the temperature measurement point is taken:

$$T_{w,o} = \frac{\sum T_{w,o}(x)}{6} \tag{9}$$

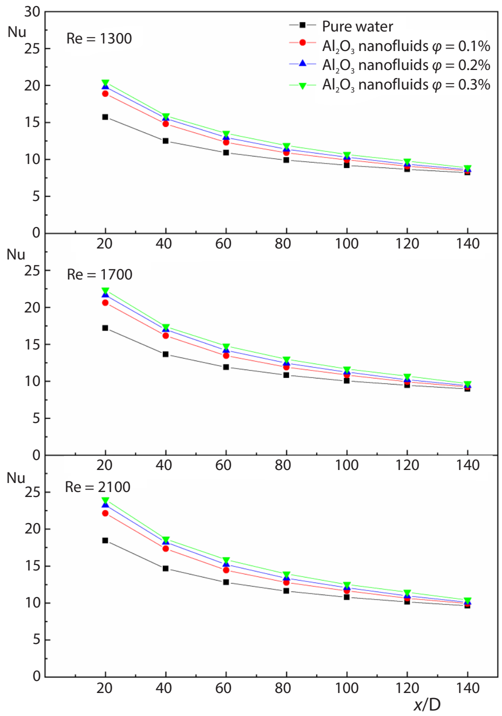


Figure 3. Change trend curve of Nu with *x*/*D* for Al_2O_3 nanofluids with different volume fractions

where T_f [K] is the characteristic temperature of the working medium:

$$T_f = \frac{T_{out} - T_{in}}{2} \tag{10}$$

Therefore, the average Nusselt number in the tube can be expressed:

$$\text{Nu} = \frac{hD_i}{k_{nf}} \tag{11}$$

In addition, the Reynolds number is obtained:

$$\text{Re} = \frac{GD_i}{A\mu_f} \tag{12}$$

where G [kgm^{-2}s^{-1}] is the mass-flow density of the fluid in the tube.

Experimental results and analysis

Experimental results of flow and heat transfer in nanofluid tubes

Figure 3 shows the change trend curve of Nusselt number with x/D for Al_2O_3 nanofluids with different volume fractions.

It can be seen that under the condition of laminar heat transfer in the tube, the change trend of local Nusselt number of Al_2O_3 nanofluids with x/D is consistent with that of pure water, and the local Nusselt number of all working fluids gradually decreases along the tube length direction, because the thickness of boundary-layer will increase with the increase of axial distance. At the same Reynolds number, the Nusselt number increases with the increase of the volume fraction of nanofluids, that is, the heat transfer effect is positively correlated with the volume fraction. This is consistent with the conclusion trend of relevant scholars [28, 29].

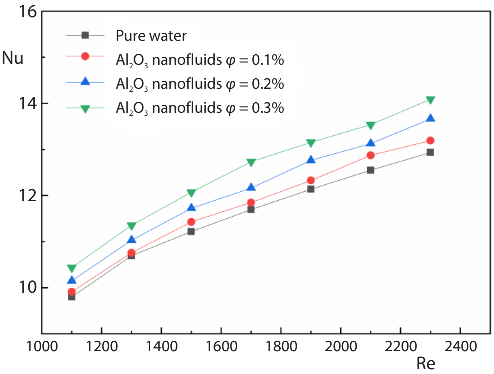


Figure 4. Average Nu of Al_2O_3 nanofluids with different volume fractions *vs.* Reynolds number

Figure 4 shows that under the same Reynolds, the thermal conductivity of Al_2O_3 nanofluid is always better than that of pure water, and the average Nusselt number of flow and heat transfer in the tube increases with the volume fraction of nanofluid.

It can be seen from fig. 5 that for pure water and Al_2O_3 nanofluids with different volume fractions, the inner wall temperature of the test pipe gradually increases along the axial direction. Under the same Reynolds number, at the same temperature measuring point, the wall temperature of pure water is always higher than that of nanofluids, and the higher the concentration of nanofluids, the lower the wall temperature of the same temperature measuring point.

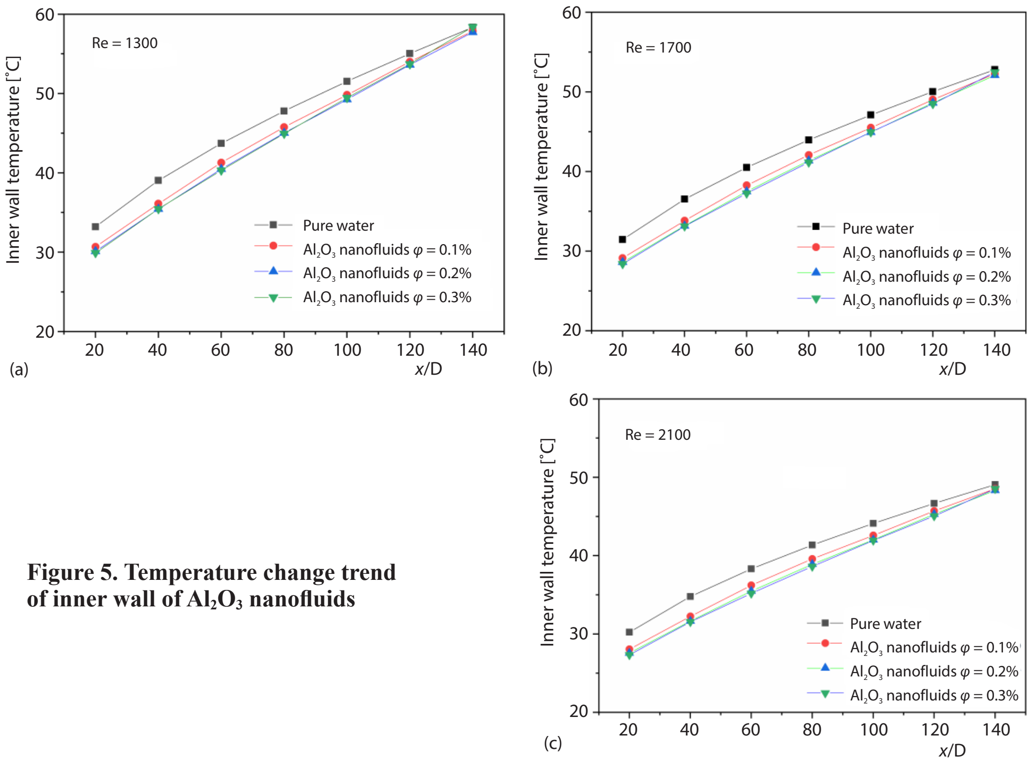


Figure 5. Temperature change trend of inner wall of Al_2O_3 nanofluids

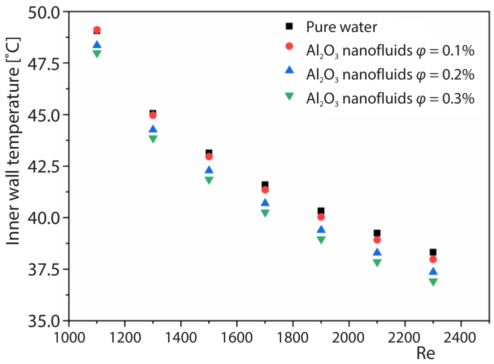
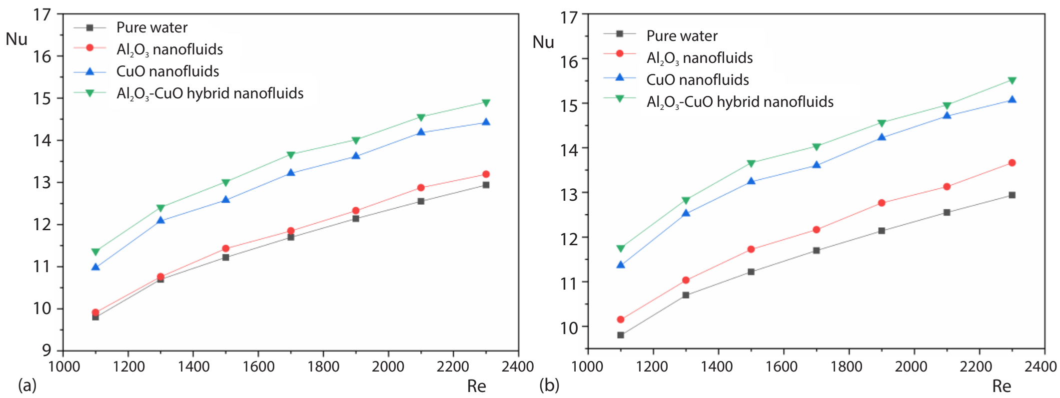


Figure 6. Average temperature of inner wall at different Reynolds number

Figure 6 indicates that with the increase of Reynolds number, the average temperature of the inner wall of the tube decreases gradually, and the higher the volume concentration of nanofluids, the lower the average temperature of the inner wall. With the increase of Reynolds number, the boundary-layer between the fluid and the inner wall of the tube becomes thinner and the heat transfer resistance decreases, thus enhancing the heat transfer effect. With the addition of nanoparticles, the disturbance of the fluid boundary-layer is enhanced, which is more significant with the increase of Reynolds number.

Figure 7 indicates that at the same volume fraction, the flow average Nusselt number in the tube is Al_2O_3-CuO nanofluids, CuO nanofluids and Al_2O_3 nanofluids from large to small. The natural settling speed of Al_2O_3 particles and CuO particles are different in the fluid. When the CuO nanoparticles close to the pipe wall first carry part of the heat and then return to the mainstream, most of the Al_2O_3 particles may still exist in the main fluid, which will participate in the exothermic process of copper oxide particles and act as a medium, accelerating energy transfer of this process.



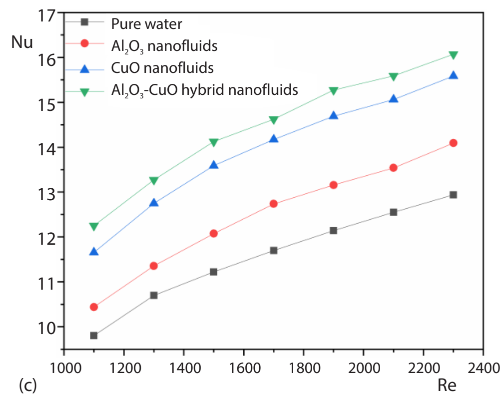


Figure 7. Variation of mean Nusselt number with Reynolds number for three nanofluids at the same volume fraction; (a) 0.1% by volume, (b) 0.2% by volume, and (c) 0.3% by volume

Exergy analysis of heat transfer in nanofluid tubes

Exergy analytical method reveals the direction of energy in the thermal process, and highlights the quality and level of energy. It can improve the scientificity and rationality of energy use better to evaluate the thermal process by means of exergy analytical method.

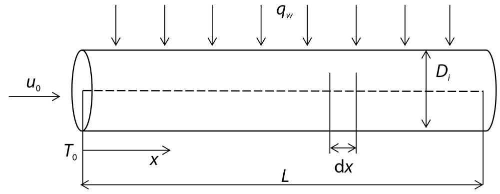


Figure. 8. Convective heat transfer model

In order to calculate the exergy loss of forced convection heat transfer in the nanofluidic round tube, a convection heat transfer model as shown in fig. 8 was established.

For the micro element control body in the convective heat transfer model shown in fig. 8, there is the relationship [30]:

$$\frac{\mathrm{d}S_\mathrm{g}}{\mathrm{d}x} = \dot{m}C_p\left(\frac{1}{T_\mathrm{f}} - \frac{1}{T_\mathrm{w}}\right)\frac{\mathrm{d}T_\mathrm{f}}{\mathrm{d}x} + \frac{\dot{m}}{\rho T_\mathrm{f}}\left(-\frac{\mathrm{d}p}{\mathrm{d}x}\right) \tag{13}$$

According to the conservation of energy, the relationship is:

$$\dot{m}C_p\mathrm{d}T_\mathrm{f} = q_\mathrm{w}\pi D_\mathrm{i}\mathrm{d}x \tag{14}$$

Substituting eq. (14) into eq. (13) gives:

$$\mathrm{d}S_\mathrm{g} = \left[q\pi D_\mathrm{i}\left(\frac{1}{T_\mathrm{f}} - \frac{1}{T_\mathrm{w}}\right) + \frac{8\pi\mu u^2}{\mathrm{d}T_\mathrm{f}}\right]\mathrm{d}x \tag{15}$$

For a tube of length, L, the entropy production:

$$S_\mathrm{g} = q\left(\frac{1}{T_\mathrm{f}} - \frac{1}{T_\mathrm{w}}\right) = Q\left(\frac{1}{T_\mathrm{f}} - \frac{1}{T_\mathrm{w}}\right) + \frac{8\pi\mu u^2}{T_\mathrm{f,out} - T_\mathrm{f,in}} \tag{16}$$

Then the exergy loss of the test section:

$$I = T_0 S_g \tag{17}$$

Exergy analysis and calculation results

According to the previous analysis, the exergy efficiency of the test section can be expressed:

$$\eta_x = 1 - \frac{T_0 S_\mathrm{g}}{Q\left(1 - \dfrac{T_0}{T_\mathrm{w}}\right)} \tag{18}$$

The exergy loss is mainly composed of heat transfer exergy loss and friction exergy loss. In this paper, taking alumina nanofluids as an example to analyze exergy loss. Figure 9 shows the relationship between heat transfer entropy production and Reznolds number due to temperature difference of different working fluids when heating power is 1000 W.

The causes of irreversible heat transfer exergy loss include axial temperature difference and radial temperature difference. For nanofluids, on the one hand, due to its higher thermal conductivity, the decrease amplitude of axial temperature difference should be greater than that of pure water, on the other hand, due to the micro movement of nanoparticles in the flow process, the radial temperature gradient of the working fluid is reduced by constantly moving from near the wall to the center of the fluid and exchanging heat, so the increasing amplitude of radial temperature difference with Reznolds number is lower than that of pure water. Under the same condition, the heat transfer entropy production of alumina nanofluid is lower than that of pure water, and the decrease range increases with the volume fraction of alumina nanoparticle.

Figure 10 shows the relationship between the friction entropy production and Reynolds number of alumina nanofluids with different volume fractions due to flow resistance when the heating power is 1000 W. The irreversible loss caused by the flow resistance during the flow and heat transfer process in the tube increases after the nanoparticles are added. This is because nanoparticles increase the viscosity of the working fluid, so the resistance of the working fluid in the flow process increases. With the increase of Reynolds number, the effect of nanoparticles on the increase of friction entropy production is more significant, and increases with the volume fraction of nanoparticles.

When the Re = 2300, the ratio of friction entropy production of alumina nanofluids

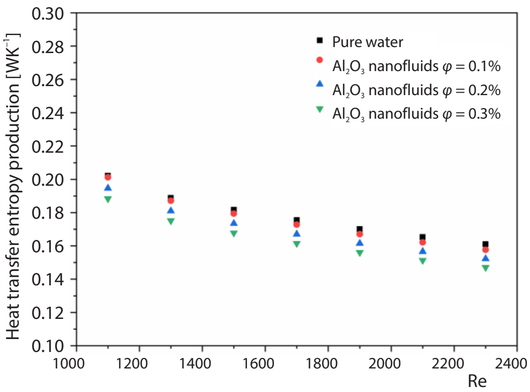


Figure 9. Change of heat transfer entropy production with Reynolds number

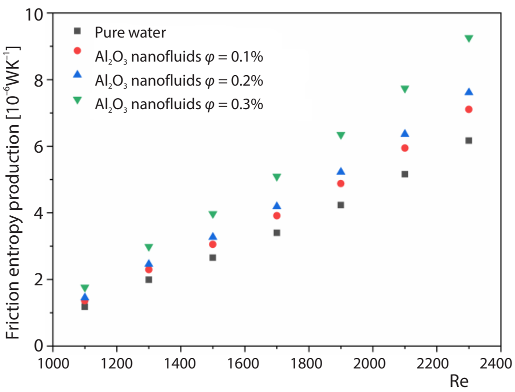


Figure 10. Change of friction entropy production with Reynolds number

with volume fraction of 0.1%, 0.2%, and 0.3% to that of pure water is 1.15, 1.23, and 1.50, respectively. However, the magnitude of friction entropy production is only 10^{-6}, which can be ignored compared with the irreversible exergy loss caused by temperature difference. Therefore, the addition of nanoparticles reduces the total entropy production in the heat exchange process, and the relevant literature also confirms the reduction of the total entropy production [31, 32].

Figure 11 shows the exergy efficiency of different nanofluids at Re = 2300. At the same heat flux, the exergy efficiency of nanofluids is always higher than that of pure water, and the exergy efficiency increases with the volume fraction of nanoparticles. At the same volume fraction, the exergy efficiency of the three nanofluids is in the order of hybrid nanofluids, CuO nanofluids and Al_2O_3 nanofluids. When the volume fraction is 0.3%, the ratio of the exergy efficiency of the three nanofluids to the exergy efficiency of water as working medium under the same condition is 1.1, 1.07, and 1.06, respectively.

By changing the heating power to measure three sets of data, the exergy efficiency of the hybrid nanofluids with different heating power conditions shown in fig. 12 can be obtained.

As can be seen from fig. 12, the heating power is adjusted, and the exergy efficiency still increases with the volume fraction of nanoparticles. At the same Reynolds number, the exergy efficiency of flow and heat transfer can be increased by improving the heat flux density.

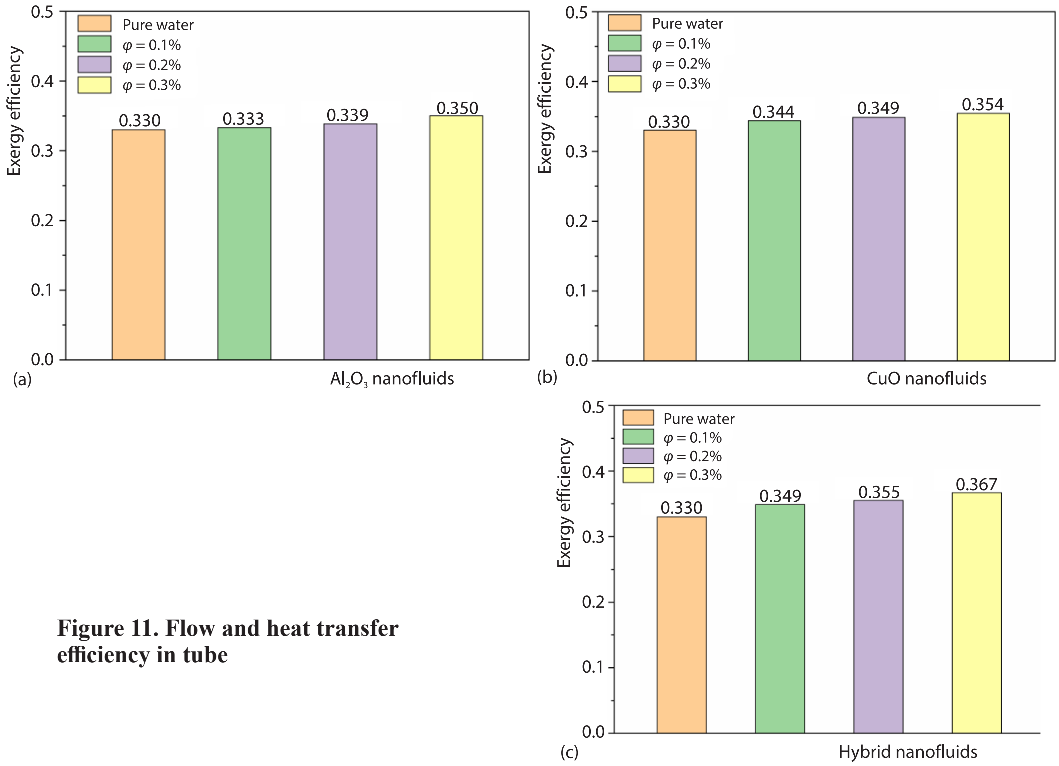


Figure 11. Flow and heat transfer efficiency in tube

Conclusions

In this paper, the convective heat transfer coefficients of different nanofluids were measured by changing the volume fraction and Reynolds number on the convective heat transfer test-bed.

Under different Reynolds number conditions, the convective heat transfer effect of nanofluids is better than that of pure water.

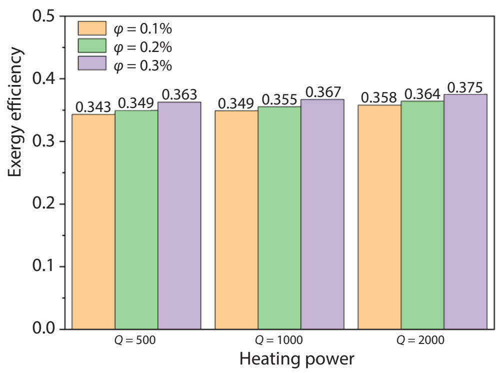


Figure 12. Efficiency under different heating power conditions

Under the condition of laminar flow in a circular tube, the average Nusselt number of the same nanofluid increases with Reynolds number; for different nanofluids with the same Reynolds number, the average Nusselt number of the flow in the tube is Al_2O_3-CuO nanofluids, CuO nanofluids, and Al_2O_3 nanofluids from the largest to the smallest with the same volume fraction. When the volume fraction reaches 0.3%, the maximum Nunf / Nuf are 1.26, 1.21, and 1.09, respectively.

An analytical model for the flow of nanofluids in a tube with constant heat flux was established, and the exergy efficiency of the flow in the tube was calculated of three kinds of nanofluids. Under the same Reynolds number condition, the exergy efficiency of nanofluids is higher than that of pure water; for nanofluids, under the same volume fraction, the increase of Reynolds number enhances the convective heat transfer coefficient, thereby reducing the temperature difference between the inner wall of the tube and the fluid, and the irreversible exergy loss of heat transfer in this process is reduced. Although the addition of nanoparticles will increase the friction exergy loss, it can be ignored compared with the irreversible heat transfer loss caused by temperature difference. When the volume fraction is 0.3% and Re = 2300, the ratio of exergy efficiency of hybrid nanofluids, copper oxide nanofluids and alumina nanofluids to the exergy efficiency of water as working medium under the same conditions is 1.1, 1.07, and 1.06, respectively. Therefore, in general, the addition of nanoparticles can effectively reduce the exergy loss.

Acknowledgment

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Nomenclature

A – area, [m^2]
D – inside diameter of circular pipe, [m]
G – mass-flow density, [kgm^{-2}s^{-1}]
h – average heat transfer coefficient, [Wm^{-2}K^{-1}]
k – thermal conductivity of the tube, [Wm^{-1}K^{-1}]
L – length of the pipe, [m]
Nu – Nusselt number (= hL/k), [–]
P – pressure, [Pa]
Q – heating power, [W]
Re – Reynolds number (= UD/n), [–]
T – temperature, [K]
u – velocity, [ms^{-1}]

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