Experimental Investigation on Convective Heat Transfer Characteristics of TiO$_2$ Ammonia Water Nanofluids

J. Weixue $^1$, D. Kai $^1$, X. Jianyong $^1$

$^1$ School of Energy and Environment, Southeast University, Nanjing, Jiangsu 210096, China

SUMMARY

The Wilson diagram method is applied to investigate the convective heat transfer characteristic of titanium dioxide (TiO$_2$) ammonia water nanofluids in this paper. At first, water is used to check whether the method is suitable because there already have the classical convective heat transfer correlation on the turbulence of water in the tube. What can be validated is that the convective heat transfer coefficient obtained by the Wilson diagram method is much closer to the results calculated by the classical correlation and the error is about only 5%, which can be deemed acceptable. Heat transfer experiments for different particle mass concentration are carried out, and the results show that the heat transfer performances of the TiO$_2$ ammonia water nanofluids are significantly enhanced when it was compared with the base fluid, which the maximum enhancement can be up to 15.1% when the particle mass concentration is 0.3%.

INTRODUCTION

Over the past few decades, nanofluid, as a new heat transfer medium, has been receiving widespread attention and developing quite rapid to provide another way to enhance the heat transfer coefficient in the heat transfer equipment (Choi 1995). The reason for this is obvious, compared to the conventional heat transfer medium such as water, ethylene glycol, etc., due to the addition of nanoparticles, making the original solution with better physical properties (Yang and Du 2017).

In today's society, refrigeration and air conditioning have become indispensable part of life and industry, and one of the main areas of energy consumer and greenhouse gases maker at the same time. In this severe situation, ammonia absorption refrigeration (AARS) has become a research hotspot again. The reason is owing to it has own superiority by utilizing low-grade heat sources without the threats of ozone depletion and global warming and outputting the considerable cooling capacity. Thus, it is not surprisingly that more and more attention has been drawn to it within research communities in recent years. But the reasons why it was neglected by researchers before this period are its low refrigeration performance and bulky size equipment in practical engineering system. Therefore, the researchers in this field focus on enhance the efficiency of AARS in particular, mostly by improving heat and mass transfer performance of the equipment in system, making full use of absorption heat, and improving working fluid performance. Among these directions, improving working fluid performance, especially the application possibility of NH$_3$/H$_2$O-nanofluids, in which nanoparticles are suspended evenly in ammonia-water, has become a new branch of AARS field.

Until now, there have been so many researchers tried to investigate the influence of nanoparticles on the heat transfer applications, including natural convective and force convective heat transfer.

Kuznetsov and Nield (2010) studied the natural convective boundary-layer flow of a nanofluid past a vertical plate, which found that the reduced Nusselt number ($N_u$) is a decreasing function of each of buoyancy-ratio parameter, Brownian motion parameter ($N_b$) and thermophoresis parameter ($N_t$). Khan and Pop (2010) investigated the laminar fluid flow which results from the stretching of flat surface in a nanofluid numerically. It was found that the reduced Nu is a decreasing function of each dimensionless number, while the reduced Sherwood number is an increasing function of higher Prandtl number ($P_r$) and a decreasing function of lower Pr number for each Lewis number ($L_e$), $N_b$ and $N_t$. Laminar flow forced convection heat transfer of Al$_2$O$_3$/water nanofluid inside a circular tube with constant wall temperature was investigated experimentally (Heris et al. 2007). Experimental results emphasize the enhancement of heat transfer coefficient due to the nanoparticles presence in the fluid. Heat transfer coefficient is increased within the increasing concentration of nanoparticles in nanofluid. Experimental investigation on the behaviour and heat transfer enhancement of a particular nanofluid, Al$_2$O$_3$ nanoparticle–water mixture was presented by Nguyen et al. (2007), results have clearly shown that the inclusion of nanoparticles into distilled water has produced a considerable enhancement of the cooling block convective heat transfer coefficient. Qiang and Yimin (2002) investigated the convective heat transfer and flow characteristics of the nanofluid in a tube, 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. The thermal resistance of heat pipe with nanofluid or with distilled water was measured, and there is a significant reduction in thermal resistance of heat pipe with nanofluid as compared with distilled water at the same charge volume (Tsai et al. 2004). An experimental investigation on the convective heat transfer characteristics in the developing region of tube flow with constant heat flux is carried out with alumina–water nanofluids by Anoop et al. (2009), it was observed that in the developing region, the heat transfer coefficients show higher enhancement than in the developed region. Ma et al. (2006) developed an ultrahigh-performance cooling device, called the nanofluid oscillating heat pipe (OHP). Experimental results show that when the OHP is charged with nanofluid, heat transport capability significantly increases. Otanicar et al. (2013) reported the experimental results on solar collectors based on nanofluids made from a variety of nanoparticles (carbon nano-tubes, graphite, and silver), the experimental and numerical results demonstrate an initial rapid increase in efficiency with volume fraction, followed by a levelling off in

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efficiency as volume fraction continues to increase. As mentioned in the previous which through the application of the nanoparticles to enhance the heat transfer performance of the article can also refer to the literature: Kumar et al. 2009; Kang 2006; Abu-nada and Oztop 2009; Duangthongsuk and Wongwises 2009; Wenhua Yu et al. 2009; W Yu et al. 2010.

TiO₂ nano-particles have been considered as one of the closest kinds to the practical application owing to its comprehensive properties such as sensational dispersity, chemical stability and non-toxicity (Yang et al. 2017). However, there is no literature in heat exchange within the application of TiO₂ nanofluids, not to mention ammonia water as a base liquid. Hence, it will be of great use to investigate the convective heat transfer characteristics of TiO₂ ammonia water nanofluids and explore the convective heat transfer correlation for future AARS researches.

Based on the above statement, we construct a device to investigate the dynamic characteristics of ammonia-water based TiO₂ nanofluids and heat transfer characteristic. But this paper only introduces the part of the convective heat experiment, and the heat transfer experiment is carried out after the dispersion stability experiment. The physical properties of the TiO₂ used in experiment are shown in table 1 and the schematic diagram of the device is shown in figure 1. Through the research of this paper, the turbulent heat transfer correlation of TiO₂ ammonia water is provided for the future researches.

Table 1. Parameters of the TiO₂ nanoparticles added to ammonia-water.

<table>
<thead>
<tr>
<th>Crystal shape</th>
<th>Rutile type</th>
</tr>
</thead>
<tbody>
<tr>
<td>Particle size, nm</td>
<td>20</td>
</tr>
<tr>
<td>Purity, %</td>
<td>99</td>
</tr>
<tr>
<td>Colour of powder</td>
<td>White</td>
</tr>
<tr>
<td>Specific surface area, m²/g</td>
<td>40</td>
</tr>
</tbody>
</table>

Figure 1. The schematic diagram of the device (The shadow part is the convective heat transfer section)

METHODS

Wilson diagram method

The reasons why Wilson diagram method is applied for the analysing the convective heat transfer, rather than directly arranged a series of temperature measurement points on the heat exchanger tube to directly acquired the wall temperature changes, and then directly obtained convective heat transfer coefficient of are: the practical operation is too difficult, if the temperature measured too much will directly affect the original flow field, too thin will lead to the measurement of fluid temperature changes is not accurate. Therefore, this experiment uses the Wilson graphical method, which is not need too many temperature measurement points, but through the diagram method to obtain convective heat transfer coefficient and the specific theory can be explained as follows:

The total heat transfer resistance in convective heat transfer can be expressed as:

\[
\frac{1}{k_o A_o} = \frac{1}{h_o A_o} + R' + \frac{1}{h_i A_i}
\]

where \(k_o\) is the total heat transfer coefficient based on the outside tube surface area, \(h_o\) and \(h_i\) are the convective heat transfer coefficient of the fluids outside and inside of the tube, respectively, \(A_o\) and \(A_i\) are the area outside and inside the tube, respectively, \(r_o\) and \(r_i\) are the thermal resistance of the tube and the scale, respectively.

Equation (1) can be changed to equation (2):

\[
\frac{1}{k_o} = \frac{1}{h_o} + R' + \frac{A_o}{h_i A_i}
\]

where \(R'\) is the sum of the thermal resistance of the tube and the scale.

The total heat transfer coefficient can be obtained by the heat balance of the whole heat exchange section:

\[
\frac{1}{k_o} = \frac{LMTD}{\Delta t c_p q_m} \cdot A_o
\]

where LMTD is the logarithmic mean temperature difference of convective heat transfer, \(c_p\) is the specific heat of the fluid inside the tube, \(q_m\) is the mass flow of the fluid in the tube, \(\Delta t\) is the temperature difference of the fluid inside the tube.

It has also been found that in the strong turbulent region of the tube, the convective heat transfer coefficient in the tube is proportional to the 0.8 power of the fluid velocity:

\[
h = c_w \cdot w^{0.8}
\]

Substituting equation (4) into equation (2) yields:

\[
\frac{1}{k_o} = \frac{1}{h_o} + R' + \frac{1}{c_w^{0.8} A_i}
\]

If the convective heat transfer coefficient of the outside fluid and the thermal resistance of the pipe and the scale are kept constant in the experiment, the first two terms of equation (5) right hand can be made constant.
The convective heat transfer coefficient can be obtained by the slope of the fitted straight line after the calculation is completed after 72 hours of dispersion stability experiment, and restart the next set of experiments after thorough cleaning, therefore, scale changes will not be too obvious.

A little different between the experimental study and the above describes is that the above-mentioned convective heat transfer coefficient in the tube is proportional to the 0.8 power of the fluid velocity in equation (4), but the relationship between the convective heat transfer coefficient and the velocity of the nanofluid is not clear and needs to be explored in the experiment:

\[ h_i = c_i w_i^0.8 \]  \hspace{1cm}  (9)

There are two unknowns in the equation (9), which needs to be explored. First, the n value can be obtained by trial of the approximation of the scatter and the straight line, and then \( c_i \) can be obtained by the slope of the fitted straight line after the value of n is obtained. Finally, the convective heat transfer correlation of the nanofluid can be obtained.

**Verification of the applicability of the Wilson graphical method with water**

The inner diameter of the inner tube of the casing is 20 mm and the wall thickness is 1.5 mm. In order to reduce the effect of the convective heat transfer inlet effect, we set a straight pipe section of 0.5 m before the heat transfer tube. The range of Reynolds numbers in this experiment is 10000 (the minimum Reynolds number required for the classical correlation of motion) to 136000 (the maximum Reynolds number experimental equipment can reach), the corresponding flow rates range is 0.4 m\(^3\)/h to 0.54 m\(^3\)/h, the water flow of outside pipe remained at 1.140 m\(^3\)/h. The data of each test point in the test are shown in Table 2. Take the

\[ 1/w_i^{0.8} \]  as the abscissa, and \( 1/k_o \) as the ordinate, and the experimental data is fitted to obtain the convective heat transfer line of the water in the tube. Finally experimental data and fitting curve are shown in Figure 2. All units in the calculation are in international units.

**Table 2. Experimental results obtained from different volumetric flows**

<table>
<thead>
<tr>
<th>Volume flow (m(^3)/h)</th>
<th>Flow rate (m/s)</th>
<th>( 1/w_i^{0.8} )</th>
<th>( 1/k_o )</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.54</td>
<td>0.4777</td>
<td>1.8058</td>
<td>8.5189</td>
</tr>
<tr>
<td>0.52</td>
<td>0.460</td>
<td>1.8612</td>
<td>8.78406</td>
</tr>
<tr>
<td>0.49</td>
<td>0.4334</td>
<td>1.9518</td>
<td>8.92779</td>
</tr>
<tr>
<td>0.46</td>
<td>0.4069</td>
<td>2.053</td>
<td>9.34122</td>
</tr>
<tr>
<td>0.43</td>
<td>0.3804</td>
<td>2.1668</td>
<td>9.55993</td>
</tr>
<tr>
<td>0.40</td>
<td>0.3538</td>
<td>2.2958</td>
<td>10.02121</td>
</tr>
</tbody>
</table>

**Figure 2. Experimental results and fitting curve (water)**

Since the calculated \( 1/k_o \) is very small, basically in the order of 10\(^{-4}\), the value of the \( 1/k_o \) is expanded 10\(^{4}\) times as the ordinate value in figure 4, so the slope of the fitting curve in the figure needs to be divided by 10\(^{4}\) to be the true slope. The slope of the fitted line is 2.46248 in the figure 4, so the actual slope is 0.000295038. Then \( C_i \) equals 3897.803 according to equation (5-8), therefore, the convective heat transfer coefficient of the forced convection in the tube is calculated as follows:

\[ h_i = 3897.803 \cdot w_i^{0.8} \]  \hspace{1cm}  (10)

The classical Dittus-Boelter correlation is used to verify the fitting correlation obtained in this paper, which is commonly used in convective heat transfer:

\[ Nu_f = 0.023 \cdot Re_f^{0.8} \cdot Pr_f^* \]  \hspace{1cm}  (11)

when the fluid is heated, \( n = 0.4 \); when the fluid is cooled, \( n = 0.3 \). In the experimental conditions, the fluid is cooled in the tube, so the value of \( n \) is 0.3. The reason why using this correlation to verify the fitting correlation is also because the Reynolds number applicable to this correlation starts at 10000, \( Pr \) is about 0.7 to 120. The average temperature of
the fluid inside the pipe in the experiment is about 35 °C, also, the other physical parameters of the fluid at this temperature are: thermal conductivity is 0.627W/(m·k), kinematic viscosity is 7.320×10⁻⁷ m²/s and Prandtl number is 4.87. The equation (11) is transformed into the form of equation (10):

\[ h_2 = 4106.1915w_i^{0.8} \]  \tag{12}

Within the measured fluid velocity range, the convective heat transfer coefficient calculated by the fitting correlation is compared with the convective heat transfer coefficient calculated by the classical calculation correlation. The results are shown in Table 3.

\[ \text{Table 3 Comparison of experimental fit correlation and classical correlation} \]

<table>
<thead>
<tr>
<th>Water velocity (m/s)</th>
<th>Convective heat transfer coefficient W/(m²·k)</th>
<th>Calculated from the fitting correlation</th>
<th>Calculated from the classical correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.4777</td>
<td>2158.4604</td>
<td>2273.8583</td>
<td>0.05075</td>
</tr>
<tr>
<td>0.4600</td>
<td>2094.2386</td>
<td>2206.2030</td>
<td>0.05075</td>
</tr>
<tr>
<td>0.4334</td>
<td>1996.7836</td>
<td>2103.5378</td>
<td>0.05075</td>
</tr>
<tr>
<td>0.4069</td>
<td>1898.4975</td>
<td>1999.9970</td>
<td>0.05075</td>
</tr>
<tr>
<td>0.3804</td>
<td>1798.9218</td>
<td>1895.0977</td>
<td>0.05075</td>
</tr>
<tr>
<td>0.3538</td>
<td>1697.5640</td>
<td>1788.3209</td>
<td>0.05075</td>
</tr>
</tbody>
</table>

From the table can be observed that the error is the same, because the error is calculated at different flow rates:

\[ \Delta = \frac{h_1 - h_2}{h_2} = \frac{4106.1915 - 3897.803}{4106.1915} = 0.05075 \]  \tag{13}

Therefore, it can be observed that the calculation of the error has nothing to do with the flow rate of water inside the tube.

In this experiment, the convective heat transfer coefficient error of the water inside the pipe obtained by Wilson’s graphic method is about 5% compared with the classic correlation, therefore, it is reasonable to believe that the experimental equipment can measure the Convective heat transfer coefficient of the fluid, and Wilson graphical method for the experimental system is applicable.

**RESULTS**

**Analysis of Convective Heat Transfer Enhancement of TiO₂ - Ammonia Nanofluid**

As mentioned earlier in the paper, there are two unknowns in the application of Wilson diagram method, namely: the factor C, and the index n are unknown in equation 5-9. The first one need to determine is the value of index n to draw a plot with 1/ko as the ordinate and 1/w³ as the absissa, and then use the scatter fit to get the slope to determine C. When the value of the index n is determined, the effect of convective heat transfer on the turbulent flow in the convective heat transfer of copper-water nanofluid is referred (Qiang and Yimin, 2002):

\[ Nu_f = 0.0059(1.0 + 7.6286\cdot10^{-0.006}Pe_{nf}^{0.001})Re_{nf}^{0.9238}Pr_{nf}^{0.4} \]  \tag{14}

From this correlation, it can be found that \( Nu_f \) is proportional to \( Re_{nf}^{0.9238} \) of the Cu-water nanofluids, thereby, 0.9238 can be regard as the referable value of index n.

In this experiment, the measured nanofluids are within 0.1% wt%, 0.2% wt%, 0.3% wt% and 0.5% wt% TiO₂ nanoparticles, respectively, and the flow rate range of nanofluid is 0.254 m³/h to 0.54 m³/h. By measuring the inlet and outlet temperature and the flow rate of the nanofluids inside the pipe and the cooling water outside the pipe, the Wilson method is also used, with 1/ko as the ordinate and 1/w³ for the abscissa. To find out the most relevant index n to the experimental data (that is, draw the curve when the error is minimal), we tried a lot of values near 0.9238 and ultimately determine the optimal value of n in this experiment is 0.91. The experimental data and the fitting curves for different concentrations are shown in figure 3. It can be observed from the figure that, when the 1/w³ is same, the 1/ko of fluid with nanoparticles added is smaller than that of the base fluid, that is, the total heat transfer coefficient is increased compared with the base fluid. In the case of the flow and the inlet temperature of water outside the tube are remain the same, the increase in the total heat transfer coefficient can be regarded as the increase of the convective heat transfer coefficient of nanofluids inside the tube. In the figure, the experimental data scatter is fitted under different conditions, and a total of five straight lines are obtained. From the obtained slope of the fitted slope, we can see that the slope of the fitting curve decreases with the increase of the nanoparticle concentration (except for the 0.5% concentration) and is less than the slope of the base fluid.

From the equation (8), the smaller the slope of the fitting straight line shows that the larger the coefficient C in the experimental correlation, it means the greater the convective heat transfer coefficient under the same conditions, indicating that the TiO₂ can promote the convection heat transfer efficiency of the fluid in the tube. But it does not mean that the higher the concentration of nanoparticles to produce the better promotion, at a particle concentration of 0.3% achieves optimal effect. Under the experimental conditions, the maximum convective heat transfer enhancement is 15.1% at 0.3% concentration.

\[ h_1 = c_i w_i^{0.91} = 4517.6465 w_i^{0.91} \]  \tag{15}
Figure 3. The experimental data and the fitting curves for different concentrations

DISCUSSION

It can also be observed from the figure 3 that the data at 0.5% concentration is almost the same as the data of 0.1% concentration, that is, the enhancement of convective heat transfer at the concentration of 0.5% does not develop according to the increase of the concentration increase. This is because the convective heat transfer experiments are carried out on the same test bed but after the stability experiment is completed, which is operated last for 72 hours. After the stability experiment, the dispersion of nanofluid within 0.5% concentration TiO₂ is not good, means the concentration of stable dispersed nanoparticles in the convective heat transfer experiment is not high. Therefore, the strengthening effect is not matched with the concentration at the time of dispensing, but matched with the concentration of particles involved in the actual process.

CONCLUSIONS

In this paper, the Wilson diagram method is used to investigate the convective heat transfer coefficient of TiO₂-ammonia water nanofluids in the tube. Also, this method is proven to be applicable because the error can be controlled within 5% through the known classical correlation of water as an experimental method to verify. Afterwards, the convective heat transfer experiment is carried out on 25% ammonia water within 0.1% wt, 0.2% wt, 0.3% wt and 0.5% wt TiO₂ nanoparticles respectively. The experimental results are processed using the Wilson graphical representation, and all the results are shown on the same figure. The results show that the convective heat transfer coefficient of the nanofluid is better than that of the base fluid, and increases with the increase of the particle concentration of the convective heat transfer. When the particle concentration is 0.3% achieved the best enhancement effect of 15.1%. What needs to be mentioned is that the strengthening effect is not matched with the concentration at the time of dispensing, but with the concentration of particles involved in the actual process.

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