Experimental investigation of condensation heat transfer of R600a/POE/CuO nano-refrigerant in flattened tubes

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A R T I C L E   I N F O

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A B S T R A C T

In this study, the effect of addition of copper oxide nanoparticles on condensing heat transfer coefficient of R600a refrigerant flowing in a flat tube condenser has been investigated experimentally. The test setup consists of a pump, condenser test, second condenser, evaporator, heaters, and flow meter. The validation of the study was done by comparing the obtained condensation heat transfer coefficients with different empirical correlations in the literature. Different fluids including pure R600a, R600a-oil with Polyester oil (POE) mass percentage of 1%, and three R600a-oil-nanoparticle mixtures with mass percentages of 0.5%, 1%, and 1.5% were studied experimentally. It was shown that adding nanoparticles will result in 4.1%, 8.11%, and 13.7% average increase in condensing heat transfer coefficient with respect to the R600a-oil mixture. The greatest amount of increase was reported for the weight fraction of 1.5%, where it was observed that the condensing heat transfer coefficient for the mixture passing through the flattened tube is averagely 109.3% higher than its corresponding value for the pure refrigerant flowing in the round tube with the same mass flux. It was also found that an increase in mass flux resulted in an increase the heat transfer coefficient at all vapor qualities.

1. Introduction

A nano-fluid is defined as a heat transfer fluid produced by suspending nano-scale ($10^{-9}$ m) materials in a host fluid [1–4]. The nanoparticles typically have a higher thermal conductivity, resulting in higher overall thermal conductivity of the fluid. If refrigerant is used as the host fluid, the obtained solution is called nano-refrigerant. Owing to the higher thermal conductivity, nano-refrigerants have higher heat transfer coefficient at a given Nusselt number. Considering thermal conductivity as an ability of the material for conducting or transmitting heat, higher thermal conductivity of nanofluids leads to the idea of employing them in refrigeration applications.

Recently, natural refrigerants such as R-717, R-744, and R-600a have attracted attention as an alternative to CFCs, HCFCs, and HFCs because of their safe operating condition, durability, and environmental concerns like ozone depleting and global warming. R-600a has been shown to be better for the environment [5,6] possessing superior energy performance [7,8] compared to other natural refrigerants. Hence R-600a was used as the refrigerant in the current study.

Several studies in the literature have used CuO nanoparticles owing to improvement in thermo-physical properties and their capability to increase the energy efficiency and effectiveness of refrigeration and air conditioning units [9]. CuO nanoparticles find wide application as they can be easily stabilized into the polyester oil and are also commercially affordable. Kedzierski [10] quantified the effect of CuO nanoparticles on boiling heat transfer coefficient of R134a/Polyolster flowing on a rough flat surface and demonstrated that the nanoparticle concentration plays a crucial role in heat transfer enhancement of the nano-refrigerants. Liu et al. [11] studied the thermal conductivity of nano-refrigerants containing CuO nanoparticles. Their study demonstrated that CuO nano-refrigerants possess considerably higher thermal conductivities in comparison with base liquids without solid particles. Coumaressin and Palaniradja built an experimental apparatus to investigate the evaporative heat transfer coefficient of R134a/CuO mixture. It was found that using CuO nanoparticles with mass concentrations up to 1% increase the heat transfer to a great extent [12]. Akhavan-Behabadi et al. [13] carried out an experimental research to investigate the effects of CuO nanoparticles on condensation heat transfer coefficient of R600a refrigerant in a horizontal round tube. Different nanoparticle mass fractions of 0.5%, 1.0%, and 1.5% were used with the vapor qualities ranging from 10% to 80%. The study reports a maximum heat transfer enhancement of 83% compared to the pure refrigerant at a mass percentage of 1.5%.

Compactness of heat exchangers has recently become another
important issue, and different passive methods have been proposed to increase the heat transfer coefficient of condensers; one of the methods proposed by the literature is flattening the tube. An experimental study on condensation heat transfer coefficient and pressure drop of R134a and R410a inside the flattened tubes with different internal heights of 5.74, 4.15, 2.57, and 0.97 mm obtained by flattening the round tube with initial diameter of 8.9 mm has been done by Wilson et al. Flattening the tube resulted in both increase in pressure drop and enhancement in heat transfer coefficient [14]. Quiben et al. also investigated the effects of flattening the tube on boiling heat transfer

![Diagram of the experimental apparatus](image-url)
coefficient of R22 and R410a. Two different round tubes with the initial diameters of 8 and 13 mm were flattened to an oblong shape tube with internal heights of 2 and 3 mm, respectively. The heat flux was found to have a negligible effect on pressure drop, with channel height significantly influencing both heat transfer coefficient and pressure drop [15,16]. Nasr et al. performed an identical study with R-134 flowing in flattened tubes with internal heights of 6.6, 5.5, 3.8, and 2.8 mm made by flattening the round copper tube with initial diameter of 8.7 mm. It was found that both pressure drop and heat transfer coefficient increase with flattening the tube, introducing the tube with internal height of 5.5 mm as the most energy efficient case [17]. Kim et al. also studied the condensation heat transfer coefficient and pressure drop of R410a in three flattened stainless tubes. They showed that flow regime plays an important role when studying the effects of flattening the tube, indicating that decreasing the internal height of the tube results in heat transfer enhancement at annular flow regime while the opposite is true for the stratified flow [18]. Darzi et al. investigated the condensation heat transfer coefficient and pressure drop of R600a in three flattened copper tubes with inner heights of 6.7, 5.2, and 3.1 mm made by flattening the round copper tube with initial diameter of 8.7 mm. They concluded that the flattened tube with internal height of 5.2 mm has the best overall performance [19].

This study uses an experimental setup identical to the one by Akhavan-Behabadi et al. [13,17,20–23], and the condensation heat transfer coefficient of CuO/oil/R600a mixture flowing inside a horizontal flattened tube with internal height of 5.2 mm, is experimentally investigated for the first time. For this purpose, the heat transfer coefficient was computed for a wide range of mass fluxes and vapor qualities, and then it was compared with heat transfer coefficient of the pure refrigerant, and the baseline mixture (R600a/oil). Mass flux was in the range of 110–372 kg/m²s, with the vapor quality between 0.06 and 0.78. Five fluids including pure R600a, R600a-oil with Polyester oil mass percentage of 1%, and three R600a-oil-nanoparticle mixtures with mass percentages of 0.5%, 1%, and 1.5% were studied experimentally.

2. The experimental setup

An experimental setup has been designed to inspect the condensation heat transfer coefficient and pressure drop within both round and flattened tubes, as depicted in Fig. 1.  

2.1. Description of the experimental units

The test condenser (#5) which is used for condensing the R-600a refrigerant, is a counter-flow double-pipe heat exchanger with the refrigerant flowing in the inner tube and the cooling water with the temperature between 15 °C and 18 °C on the outside; the flattened tube with an internal height of 5.2 mm, a length of 105 cm, and the initial diameter of 8.7 mm was used in the test condenser section. Studying the impact of refrigerant vapor quality on the condensation heat transfer coefficient was one of the purposes of the current work. As the experimentally flattened tube length is only 105 cm, it would be impossible to cover a wide range for refrigerant vapor quality by using only two condensers. Hence, two heaters and an evaporator were utilized for achieving desired range of vapor quality. A copper tube with the length of 1 m was employed for each heater as well as the evaporator, with heating elements uniformly twisted around the outside. Each element can produce up to 3000 W of electrical power, and the watt meter precision is 1%. The internal height of 5.2 mm has been found to be the optimum value from the stand point of heat transfer and pressure drop for a flattened tube with standard initial diameter of 8.7 mm [17]. The temperature of the flattened tube outer surface is measured by a K-type thermocouple at six different equally spaced points at the top and bottom side of the tube. The thermocouple device can measure temperatures as high as 1000 K with the accuracy of 0.1 K. The measured temperatures can be read by the data logger which is connected to the computer. The pressure is also measured at the inlet and outlet of the test condenser, using a pressure gauge with the accuracy of 10 kPa. The outer surfaces of all heat exchangers were insulated using glass wool. While the glass wool significantly reduces heat transfer, the heat loss cannot be completely eliminated and has been included in our calculations. A post condenser with 12 m of spiral tube was used in order to fully condense the refrigerant, preventing multiphase flow at the pump inlet. A one-way valve is utilized before the pumping section in order to prevent back flow into the post condenser.

Pressure drops within the test section were measured by the Endress Hauser PDM-75, calibrated by the company. The device can operate in the range of 0 to 150 kPa with the precision of 0.075SPN percent, where SPN is the pressure at the condenser. The fluid mass flow rate is computed by the Danfoss MASS/2100/600 flowmeter; this device can measure the mass flow rates less than 250 kg/h with the precision of 1%.

2.2. Experimental setup operation

As illustrated in Fig. 1, the refrigerant is pumped through the system using the gear-pump (#1). The refrigerant then flows through the flow meter (#2), followed by the heaters (#3). The heater is used to achieve the varying vapor qualities as the vapor quality at the inlet of the test section is affected by the amount of heat added by the heating elements as well as the heat losses in the Sections 3 and 4. The multiphase flow then passes the test condenser (#4), where data for computing the condensing heat transfer coefficient is measured. Finally, the refrigerant is totally condensed in the post condenser (#7) to be pumped again through the gear-pump (#1).

In the current study, 192 experiments with different mass flux and vapor quality values were conducted for six different cases at four different flow rates and eight different vapor qualities for each. For all experiments, the following parameters were measured:

1. Refrigerant mass flow rate
2. Cooling water mass flow rate at the test condenser
3. Refrigerant temperature at inlet and outlet of test condenser and post condenser
4. Outer wall of the inner tube temperature at 12 points within the test condenser section
5. Cooling water inlet and outlet temperatures at the test condenser
6. Pressure at inlet and outlet of the test condenser and the post condenser
7. Amount of heat added at the heaters and the evaporator section
8. Pressure drop through the test condenser

Using all of the mentioned parameters, the condensing heat transfer coefficient within the flattened tube located at the test condenser was computed. First, the round tube and pure refrigerant were utilized; in the next step, the flattened tube and pure refrigerant were studied. Afterwards, the parameters were measured for refrigerant-oil flow in the flattened tube; finally, the heat transfer coefficient for the R600a-oil- CuO nanofluids with mass concentrations of 0.5, 1.0, and 1.5% were investigated experimentally. Note that adding nanoparticles directly to the refrigerant does not make sense as the R600a-CuO mixture is not stable [24]. Hence, the nano particles were first dissolved in the polyester oil with mass concentrations of 0.5, 1.0, and 1.5% using homogenizer, and then the oil-CuO mixture was added to the refrigerant with weight fraction of 1%, as done by [24]. The range of different operating conditions is mentioned in Table 1.

3. Heat transfer coefficient calculations

Saturation temperature, saturation enthalpy, and all thermo-physical properties of R600a were computed using EES software. The vapor quality at the test condenser inlet can be computed by writing the
energy equation for a control volume including the heaters and the evaporator as follows:

\[
\dot{Q}_{\text{h,1}} + \dot{Q}_{\text{h,2}} + \dot{Q}_{\text{f}} = \dot{m}_f [C_{p_f} (T_{\text{sat,f}} - T_{\text{in}}) + \epsilon_r (\psi_{\text{out}} - \psi_{\text{in}})]
\]

Where, \(\dot{Q}_{\text{h,1}}\) and \(\dot{Q}_{\text{h,2}}\) represent the effective heat input from the two heaters and \(\dot{Q}_f\) is the heat input from the evaporator. The first term in the right hand side of the Eq. 1 represents the change in the enthalpy of the liquid refrigerant from the inlet of the first heater to the saturation point; the second term is the enthalpy of the two-phase refrigerant at the outlet of the evaporator or at the inlet of the test condenser, where the last term corresponds to the enthalpy of the liquid refrigerant in saturation temperature. It may be noted that

\[
\psi_{\text{out}} - \psi_{\text{in}} = x_{\text{v, out}} (\psi_{\text{sat}} - \psi_{\text{v, in}})
\]

To account for the heat loss, a factor, \(\eta\), the fraction of the electrical power that is transferred to the refrigerant at the heater sections is included. Hence,

\[
\dot{Q}_f = \eta \ W_i
\]

Where, \(W_i\) is the total electric power shown by the watt meter. The value of \(\eta\) was experimentally determined to be 0.7. Combining the Eqs. 1 to 3, the vapor quality at the test condenser inlet is given by:

\[
x_{\text{v, in}} = x_{\text{v, out}} = \frac{1}{\epsilon_r} \left[ \frac{\eta W_i}{\dot{m}_f} - C_{p_f} (T_{\text{sat,f}} - T_{\text{in}}) \right]
\]

Condensation heat transfer for each experiment is computed by using the amount of heat absorbed by the cooling water and the temperature difference between the inner wall of the tube and refrigerant according to the following procedure:

1. The local temperature of the tube outer wall at six axial points is computed by the following equation as depicted in Fig. 2

\[
T_{\text{w,0,m}} = \frac{T_{\text{w,0}} + T_{\text{w,0,p}}}{2}
\]

Where, \(T_7\) and \(T_9\) are the measured temperature at the upper and at the lower surfaces of the tube, respectively.

2. The mean temperature of the outer wall of the tube is calculated by averaging the values of \(T_{\text{w,0,m}}\)through the whole section

\[
T_{\text{w,0,m}} = \frac{\sum_{i=1}^{6} T_{\text{w,0,i}}}{6}
\]

3. The rate of heat transfer within the test condenser is computed by measuring the inlet and outlet cooling water temperatures, as well as its mass flow rate

\[
\dot{Q}_w = \dot{m}_w C_{p_w} (T_{\text{w,0}} - T_{\text{w,1}})
\]

4. The radial heat flux is then calculated using the following equation

\[
\dot{q}_w = \frac{\dot{Q}_w}{\pi D_L}
\]

Where \(D\) is the diameter for the round tube, or the hydraulic diameter for the flattened tube. The Cross sectional area of the flattened tube is needed for computing the hydraulic diameter which is calculated as:

\[
A = \frac{\pi H^2}{4} + wH
\]

Where, \(w\) is the flattened tube width, which was 5.2 mm; the value of 7.29 mm was found for the hydraulic diameter.

5. The temperature drop through the flattened tube wall \(\Delta T_w\) is computed by the following equation:

\[
\Delta T_w = \frac{\dot{q}_w D_s L (\frac{T_9}{T_0})}{2k_w}
\]

In which \(k_w\) is the thermal conduction of the copper wall, and \(D_s\) and \(D_t\) are outer and inner diameters of the tube.

6. The mean temperature of the inner wall \(T_{\text{in,m}}\) is assumed equal to the mean outer wall temperature, \(T_{\text{w,0,m}}\).

7. The saturation temperature of the vapor \(T_s\) corresponds to the mean static pressure at the condenser which was assumed to be the average of inlet and outlet static pressure values.

8. The average condensation heat transfer coefficient is finally calculated using the following equation.

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**Table 1**
The range of different operating conditions for R600a, R600a-oil, and R600a-oil-CuO.

<table>
<thead>
<tr>
<th>Fluids</th>
<th>Values</th>
<th>Units</th>
</tr>
</thead>
<tbody>
<tr>
<td>Mass flux</td>
<td>110-372</td>
<td>kg/m²s</td>
</tr>
<tr>
<td>Condensation temperature</td>
<td>36.2-45.6</td>
<td>°C</td>
</tr>
<tr>
<td>Vapor quality at the inlet of test condenser</td>
<td>0.10-0.85</td>
<td>–</td>
</tr>
<tr>
<td>Vapor quality at the outlet of test condenser</td>
<td>0.01-0.66</td>
<td>–</td>
</tr>
<tr>
<td>Average vapor quality at the test condenser</td>
<td>0.066-0.78</td>
<td>–</td>
</tr>
<tr>
<td>Cooling water mass flow rate</td>
<td>0.065-0.21</td>
<td>–</td>
</tr>
<tr>
<td>Cooling water thermal flux</td>
<td>22.31-23.19</td>
<td>kW/m²</td>
</tr>
<tr>
<td>Cooling water temperature change</td>
<td>0.7-2.8</td>
<td>°C</td>
</tr>
</tbody>
</table>

**Fig. 2.** Schematic view of thermocouples locations used for finding temperature in test condenser and cross section view of the flattened tube.
\[ h = \frac{q_v}{T_s - T_{in, in}} \]  
\[ (11) \]

**4. Results and discussion**

The value of the condensation heat transfer coefficient for all 192 cases were calculated using the procedure explained in the previous section. Effects of different parameters such as refrigerant mass flux, vapor quality, and flattening the tube on the condensation heat transfer coefficient for six different cases mentioned in Table 2 were investigated. Note that for the cases of nanoparticle-oil-refrigerant, the CuO nanoparticles are solved within the oil first, then the solution is added to the R600a refrigerator.

### 4.1. The flow regime

The flow regime for test cases were also investigated in the current study using two different methods. First, according to the Breber flow pattern map shown in Fig. 3, almost all of the experimented cases lie in the annular and mist annular region except for the very low values of vapor quality which results in transition regime from wavy and stratified to the annular and mist annular.

In the Breber Map, \( X_n \) is the Martinelli parameter calculated as:

\[ X_n = \left[ 1 - \frac{x}{x} \right]^{0.9} \left[ \frac{\mu_s}{\mu} \right]^{0.11} \left[ \frac{\rho_s}{\rho} \right]^{0.5} \]  
\[ (12) \]

\( J_{fl}^{*} \) presents the vapor dimensionless velocity which is computed by the following equation:

\[ J_{fl}^{*} = \frac{G_{fl}}{[Dg_{fl}(\rho_f - \rho)]} \]  
\[ (13) \]

Secondly, the flow pattern was studied using the method presented by Nitheanandan et al. [25] The proposed method uses the Froude and the Weber dimensionless numbers for predicting the flow regime. The Froude number was computed using the equation presented by Dobson [26]:

\[ Fr = 0.025Re_{fl}^{0.75} \left( 1 + 1.09X_{fl}^{0.039} \right) \]  
\[ Ga^{0.5} \text{ for } Re \leq 1250 \]  
\[ (14.1) \]

\[ Fr = 1.26Re_{fl}^{0.94} \left( 1 + 1.09X_{fl}^{0.039} \right) \]  
\[ Ga^{0.5} \text{ for } Re > 1250 \]  
\[ (14.2) \]

Where,

\[ Ga = \frac{\rho_l(\rho_l - \rho_f)gD^3}{\mu^2} \]  
\[ (15) \]

\[ FRe_l = \frac{GD(1 - x)}{\mu} \]  
\[ (16) \]

The Weber number was determined using the equation provided by Soliman et al. [27]:

\[ We = 2.45Re_{fl}^{0.64} \left( \frac{\mu_s}{\mu} \right)^{0.3} \left( 1 + 1.09X_{fl}^{0.039} \right)^{-0.4} \text{ for } Re \leq 1250 \]  
\[ (17.1) \]

\[ We = 0.85Re_{fl}^{0.79} \left( \frac{\mu_s}{\mu} \right)^{0.3} \left( 1 + 1.09X_{fl}^{0.039} \right)^{-0.4} \left( \frac{\mu_l}{\mu} \right)^{0.084} X_{fl}^{0.155} \text{ for } Re > 1250 \]  
\[ (17.2) \]

It was found that for all cases \( We < 40 \) and \( Fr > 7 \), indicating that the flow regime was annular for the experiments, as discussed earlier. The same result was achieved using the Breber pattern flow map. Besides, the We number did not exceed the value of 40 which means that the flow regime is not even mist annular for any case and pure annular flow pattern exists at the experiment conditions. However, effects of the gravity force should also be investigated [28]. As the flow pattern is annular, which occurs at high velocities and high vapor qualities, the shear forces are dominated within the fluid zone. Hence, the gravity force does not affect the flow significantly and it can be assumed that a layer of saturated liquid phase is produced on the inner side of the tube with vapor phase flowing in the middle. Although this phenomenon increases the pressure drop, it is highly desired from the heat transfer point of view [20]. With decreasing the vapor quality and velocity to much lower values, which did not happen in the current study, the gravity forces play a major role and liquid phase flows on the lower side of the tube, while the vapor phase passes through the upper side; this will reduce the heat transfer coefficient because the condensed liquid acts as a thermal resistance against the heat flux [29].

### 4.2. Study validation

Each case presented in Table 2 was experimentally studied at 4 different mass fluxes and the value of condensing heat transfer coefficient was measured using flows with 8 various vapor qualities. In order to validate the experimental setup, obtained heat transfer coefficients for pure R-600a were compared with those predicted by previous studies. There are so many analytical models for computing the condensation heat transfer coefficient within the round tubes in the literature. In the current study, the experimental results have been compared with five well-known correlations to see if the experiment errors lie within an acceptable range. Correlations used for the validation purpose are all mentioned in the Table 3, and the comparison of the empirical correlations against the current data is shown in Figs. 4–8.

According to the presented figures, Shah correlation shows the minimum deviation from the experimental results with the maximum deviation of 13%; the same trend has been reported by other studies in the literature, introducing the Shah correlation as the most reliable tool for computing the condensation heat transfer coefficient of mixtures [34–39]. However, other correlations predicted greater heat transfer coefficients in comparison with the current data. Many factors can cause deviation of the experimental results from the empirical correlations. For example, impurities affect the rate of heat transfer to a high degree. Besides, the transport properties of the refrigerants presented by the various references vary from 20 to 40%. Furthermore, one must consider the error due to inadequate instrumentation, faulty instruments and human errors [40]. In view of the many sources of error which was discussed, a good agreement between measurements and the empirical correlations exists, and the maximum mean deviation was found to be about 25%.

#### 4.3. Heat transfer enhancement

Experiments were conducted for pure refrigerant, refrigerant-oil mixture with weight fraction of 1% and refrigerant based nanofluids

<table>
<thead>
<tr>
<th>Case</th>
<th>Fluid</th>
<th>Tube type</th>
<th>Nanoparticle concentration in oil</th>
<th>Oil/nanoparticle concentration in refrigerant (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>R600a</td>
<td>Round tube</td>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td>2</td>
<td>R600a</td>
<td>Flattened tube</td>
<td>0.0</td>
<td>0.0</td>
</tr>
<tr>
<td>3</td>
<td>R600a-oil</td>
<td>Flattened tube</td>
<td>0.0</td>
<td>1.0</td>
</tr>
<tr>
<td>4</td>
<td>R600a-oil-CuO</td>
<td>Flattened tube</td>
<td>0.5</td>
<td>1.0</td>
</tr>
<tr>
<td>5</td>
<td>R600a-oil-CuO</td>
<td>Flattened tube</td>
<td>1.0</td>
<td>1.0</td>
</tr>
<tr>
<td>6</td>
<td>R600a-oil-CuO</td>
<td>Flattened tube</td>
<td>1.5</td>
<td>1.0</td>
</tr>
</tbody>
</table>
with three different nanoparticle mass fractions of 0.5, 1, and 1.5%. Finally, 192 experiments were done and the obtained values of condensing heat transfer coefficients were computed as presented in Fig. 9.

For the case of pure refrigerant in the flattened tube, it was observed that heat transfer coefficient decreases with decreasing the vapor quality at all values of mass flux. The same trend is true for the pure refrigerant flowing in the round tube. This makes sense because the liquid layer on the inner side of the tube is thinner at higher vapor qualities which results in lower thermal resistance against the heat flux.

Before studying the effects of adding CuO nanoparticles to the refrigerant, the behavior of the R600a-oil solution with mass

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### Table 3
The empirical correlations used for the study validation.

<table>
<thead>
<tr>
<th>Number</th>
<th>Flow pattern correlation</th>
<th>Model/correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>1</td>
<td>Traviss et al. [30]</td>
<td>$Nu = \frac{k_d^{0.8} Pr}{F(X_t)}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$F_2 = 0.707 Pr Re_0^{0.5}$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$F_2 = 5Pr + 5.5ln(1 + Pr(0.096Re_0^{0.585} - 1))$</td>
</tr>
<tr>
<td></td>
<td></td>
<td>$F_2 = 5Pr + 5.5ln(1 + Pr) + 2.5ln(0.00313Re_0^{0.811})$</td>
</tr>
<tr>
<td>2</td>
<td>Cavalini et al. [31]</td>
<td>$Nu = 0.05PrRe_0^{0.8}$</td>
</tr>
<tr>
<td>3</td>
<td>Dobson [26]</td>
<td>$Nu = 0.023PrRe_0^{0.8}$</td>
</tr>
<tr>
<td>4</td>
<td>Shah et al. [32]</td>
<td>$h = h_t(1 - X_0)^{0.8} + \frac{3.5X_0^{0.78}(1 - X_0)^{0.04}}{Pr_0^{0.28} Pr_0^{0.8} Pr}^{0.8}$</td>
</tr>
<tr>
<td>5</td>
<td>Jung et al. [33]</td>
<td>$Nu = 22.4h_t(1 + \frac{2}{X_0})^{0.81}(\frac{g}{Re_0 D})^{0.29}$</td>
</tr>
</tbody>
</table>

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Fig. 3. Breber flow pattern map.

Fig. 4. Comparison of the Traviss correlation against the present data [30].

Fig. 5. Comparison of the Cavallini correlation against the present data [31].

Fig. 6. Comparison of the Dobson correlation against the present data [26].
Reynolds number is reduced and this will result in lower heat transfer in the liquid phase. Consequently, increasing the viscosity, the value of the heat transfer coefficient in the liquid phase by increasing the vapor quality, as oil remains in the refrigerant-oil-CuO in Fig. 9 with the corresponding values of refrigerant-oil-CuO [13]. On the other hand, there is an increase in oil concentration within the pure refrigerant which helps the mechanism of heat transfer to improve condensation heat transfer coefficient compared to the pure refrigerant, the CuO nanoparticles were side a horizontal plain tube. For the feasibility of nanoparticle dispersion into the refrigerant, the CuO nanoparticles were first solved into polyester oil, and then the mixture was added to the refrigerant. Many predictive methods are available for two phase flow patterns and heat transfer coefficient in the horizontal tubes which were compared with the current experimental data and it was found that Shah correlation has the minimum deviation from the experimental results. Annular flow pattern was found to be the dominant flow pattern in this study. The results showed that the heat transfer coefficient increases with increasing mass velocity and vapor quality. However, for the vapor qualities higher than approximately 0.6, the heat transfer deteriorated due to the rise in local oil concentration. Adding nanoparticles can improve condensation heat transfer coefficient up to 13.70% compared to oil/refrigerant mixture occurred at the mass velocity of 212 kg/m²s for the highest mass fraction of nanoparticles of this study, i.e. 1.5%. Some reasons were presented for the heat transfer enhancement during nanofluid condensation such as disturbance of nanoparticles, reduction in boundary layer thickness, formation of molecular adsorption layer, the Brownian motion effects, increase in the surface area as well as in the heat capacity of the fluid, and rise in the effective thermal conductivity of the mixture. For the 0.5% mass fraction of nanoparticles at 110, 212, 295 and 372 kg/m³s mass flux, the average heat transfer coefficient is di 13.7% higher than the refrigerant-oil mixture. Note that achieving even higher values of increase in heat transfer coefficient is possible at some vapor qualities. The average values of heat transfer enhancement for all cases are mentioned in Table 4. This rise in rate of heat transfer can be accounted for by different phenomenon happening by adding the nanoparticles and flattening the tube. First, disturbance of nanoparticles enhances the heat transfer due to the reduction of boundary layer thickness as the boundary layer acts as a thermal resistant against the heat flux [41]. Second, the liquid molecules can be absorbed by the nanoparticles which leads to the formation of the molecular adsorption layer on their surface [42]. The rate of heat transfer in the body of the fluid is then increased by the formation of molecular layer and nanoparticles movements.

Likewise, one of the other dominant parameters affecting the thermal conductivity of the mixture and the heat transfer coefficient within the fluid media is Brownian motion due to addition of nanoparticles. Brownian motion can increase the thermal conductivity of the nanofluid by an insignificant value, primarily by increasing the “random walk” motion instead of motion only through diffusion. This improving effect only occurs at lower concentrations and after reaching a limit in weight fraction of nanoparticles, no more increase in heat transfer coefficient is achievable [43]. Besides, after more increasing the nanoparticles mass concentration, while the cluster formation leads to an increase in conductivity, the maximum rise remains well below the maximum value allowed by Maxwell’s upper limit [44]. Furthermore, the movement velocity of larger particles is much lower than that of smaller ones, which reduces the probability of collision [45]. Therefore, the condensing heat transfer coefficient within the fluid with maximum weight fraction of 1.5 for CuO nanoparticles was investigated at this study.

Other reasons can also be accounted for the improvement of heat transfer by suspending nanoparticles in heating or cooling fluids. The suspended nanoparticles increase the surface area, the heat capacity of the fluid, and the effective thermal conductivity of the mixture; they also intensify the flow passage surface and mixing fluctuations, as well as the turbulence of the fluid [46].

5. Conclusion

Experiments were conducted to investigate the effects of CuO nanoparticles on the condensation heat transfer coefficient of R-600a inside a horizontal plain tube. For the feasibility of nanoparticle dispersion into the refrigerant, the CuO nanoparticles were first solved into polyester oil, and then the mixture was added to the refrigerant. Many predictive methods are available for two phase flow patterns and heat transfer coefficient in the horizontal tubes which were compared with the current experimental data and it was found that Shah correlation has the minimum deviation from the experimental results. Annular flow pattern was found to be the dominant flow pattern in this study. The results showed that the heat transfer coefficient increases with increasing mass velocity and vapor quality. However, for the vapor qualities higher than approximately 0.6, the heat transfer deteriorated due to the rise in local oil concentration. Adding nanoparticles can improve condensation heat transfer coefficient up to 13.70% compared to oil/refrigerant mixture occurred at the mass velocity of 212 kg/m²s for the highest mass fraction of nanoparticles of this study, i.e. 1.5%. Some reasons were presented for the heat transfer enhancement during nanofluid condensation such as disturbance of nanoparticles, reduction in boundary layer thickness, formation of molecular adsorption layer, the Brownian motion effects, increase in the surface area as well as in the heat capacity of the fluid, and rise in the effective thermal conductivity of the mixture. For the 0.5% mass fraction of nanoparticles at 110, 212, 295 and 372 kg/m³s mass flux, the average heat transfer coefficient is 13.7% higher than the refrigerant-oil mixture.
enhancement up to 0.96%, 4.10%, 1.61% and 3.53% was achieved. Besides, for 1% mass fraction of nanoparticles the average heat transfer enhancement up to 7.1%, 8.11%, 6.37% and 6.90% was obtained. Finally, the most average heat transfer enhancement occurred at the 1.5% mass fraction of nanoparticles, which are up to 11.67%, 13.70%, 10.77% and 11.57% for the mass velocities of 110, 212, 295 and 372 kg/m²s, respectively. Therefore, one can use oil/CuO/r600a and flattened tube instead of round tube to increase heat transfer which consequently increases the overall efficiency of any condensing system.

Table 4
Increase in heat transfer coefficient with respect to the refrigerant-oil case for different mass fluxes.

<table>
<thead>
<tr>
<th>Case number</th>
<th>Percent of increase in heat transfer coefficient</th>
</tr>
</thead>
<tbody>
<tr>
<td>G = 110 kg/m²s</td>
<td>G = 212 kg/m²s</td>
</tr>
<tr>
<td>4</td>
<td>0.96</td>
</tr>
<tr>
<td>5</td>
<td>7.1</td>
</tr>
<tr>
<td>6</td>
<td>11.67</td>
</tr>
</tbody>
</table>

Fig. 9. Estimated values of condensing heat transfer coefficient for different values of mass flux.

References
