Effect of lubricating oil on condensation characteristics of R600a inside a horizontal U-shaped tube: Experimental study

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A R T I C L E   I N F O

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R600a
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A B S T R A C T

In this paper, heat transfer characteristics of hydrocarbon refrigerant R600a-oil mixture during condensation inside a horizontal U-shaped tube are investigated. The experiments are performed in three parts: (i) pure refrigerant flow inside a straight tube, (ii) pure refrigerant flow inside a U-shaped tube, and (iii) refrigerant-oil mixture flow with compactness of 1, 1.5, and 2% inside a U-shaped tube. An experimental setup has a copper tube of 8.7 mm inner diameter as a test condenser which is installed into a polyethylene pipe of 6 cm inner diameter to form a tube-in-tube heat exchanger. The experimental tests are carried out in a wide range of operating conditions, including mass flux of 140–280 kg/m²s, vapor quality of 0.04–0.80, and condensation pressure of 510–630 kPa. The results are validated by comparing the experimental condensation heat transfer coefficient of pure refrigerant flow inside straight tube with well-known empirical correlations. Accordingly, they predict most of the experimental data within ±20% error in pure refrigerant part. The results represent that using a U-shaped tube increases the local and the average heat transfer coefficients by a maximum of 34% and 29.7% in comparison to the straight tube, respectively. In addition, in the presence of oil, the local heat transfer coefficient of refrigerant-oil mixtures increases at low vapor qualities, while decreases at middle and high qualities due to the effect of oil on the condensation liquid film.

1. Introduction

Heat transfer enhancement is one of the most critical issues in developing efficient and compact heat exchangers, for which different passive methods have been invented [1–3]. One of these methods is using U-shaped tubes with consecutive 180° return bends, which are widely employed in condensers and evaporators of refrigeration systems. Modification of a straight tube to a U-shaped one enhances the heat transfer and compactness of the heat exchangers.

Through many research efforts, flow condensation in straight tubes is studied, and some correlations have been proposed by Cavallini et al. [4], Traviš et al. [5], Cavallini and Zecchin [6], Shah [7,8], and Thome [9]. However, heat transfer characteristics in U-bends are entirely different from that in straight tubes. Few papers have been investigated the condensation flow inside U-bends. Lee et al. [10] studied evaporation and condensation heat transfer characteristics of R600a, R290, R1270, and R22 inside a horizontal double-pipe U-shaped heat exchanger. Tube diameter of 12.70 mm with 1.315 mm wall thickness was used for the experiments. The experimental results showed that inside a U-shaped tube, the local two-phase heat transfer coefficient of hydrocarbons are higher than those of R22.

The main characteristic of R600a, as a natural refrigerant, is that besides having zero ODP (ozone depletion potential), its GWP (global warming potential) is close to zero. These environmental advantages make R600a a proper candidate for replacing common refrigerants in air conditioning and refrigeration systems. Cahn [11] reviewed the progress of refrigerants - from early uses to present and addressed that natural refrigerants have taken the place of HCFCs and HFCs refrigerants so that nowadays they dominate domestic refrigerators in Europe. Chang et al. [12] investigated the Coefficient of Performance (COP) and heat-transfer characteristics of hydrocarbon refrigerants, including propane, iso-butane, butane, and propylene in a heat pump system. They showed that the system performance using hydrocarbon refrigerants is comparable with R22. In another research work, Lee and Son [13] studied experimentally the condensation heat transfer coefficient of R290, R600a, R22, and R134a inside a horizontal double-pipe heat exchanger with inner diameters of 5.80, 6.54, 7.73, and 10.07 mm. They concluded that the average condensation heat transfer coefficient of R600a is higher than the others. Similar results have been also reported by other recent studies [14–16].

The main duty of oil in refrigerant systems is lubricating and sealing compressor moving parts. A portion of lubricant oil mixes with
### Nomenclature

- \( c_p \) Specific heat capacity (kJ/kgK)
- \( d \) Tube diameter (m)
- \( G \) Mass flux (kg/m²s)
- \( h \) Heat transfer coefficient (W/m²K)
- \( i \) Enthalpy (kJ/kg)
- \( k \) Thermal conductivity (W/mK)
- \( L \) Tube length (m)
- \( m \) Mass flow rate (kg/s)
- \( p \) Pressure (kPa)
- \( Pr \) Prandtl number (-)
- \( Q \) Heat transfer rate (W)
- \( q \) Heat flux (W/m²)
- \( Re \) Reynolds number (-)
- \( T \) Temperature (K)
- \( W \) Electrical power (W)
- \( x \) Vapor quality (%)  
- \( X_{H} \) Martinelli parameter (-)

### Subscripts

- \( avg \) Average
- \( b \) Bubble
- \( c \) Condenser
- \( cw \) Cooling Water
- \( eq \) Equal
- \( f \) Fluid
- \( fg \) Fluid-gas
- \( g \) Gas
- \( h \) Heater
- \( i \) Inner
- \( in \) Inlet
- \( r \) Refrigerant
- \( l \) Liquid phase
- \( loc \) Local
- \( m \) Mixture
- \( o \) Oil, Outer
- \( out \) Outlet
- \( sat \) Saturated
- \( v \) Vapor phase
- \( w \) Wall

### Greek symbols

- \( \eta \) Insulation efficiency (-)
- \( \mu \) Dynamic viscosity (Pa/s)
- \( \rho \) Density (kg/m³)
- \( \omega \) Oil concentration (%)  

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refrigerant and circulates in the refrigerant system. Therefore, the presence of oil is unavoidable and its effect on the heat transfer characteristics should be investigated. A lot of papers considered the impact of oil on refrigerant-oil mixtures inside straight tubes. Eckels et al. [17–19], Usmani and Ravishurajan [20], Shao and Granryd [21], Infante Ferreira et al. [22], Momenifar et al. [23], Huang [24], and Ref. [25,26] reported different heat transfer behaviors because of adding lubricant oil to refrigerant flow inside straight tubes, which depends on operating conditions such as refrigerant type, vapor quality, mass flux, oil type and its concentration. Some previous research works were about the effect of oil on the condensation heat transfer of refrigerant-oil mixture inside U-shaped tubes. Cho and Tae [27,28] investigated condensation and evaporation heat transfer of R407C and R22 mixed with polyester and mineral oils, respectively, in straight and U-bend sections of a micro-fin tube. They demonstrated that the local heat transfer coefficient shows a maximum at 90° of the U-bend. The heat transfer coefficient in the straight tube after the U-bend is more significant than that of the entrance part. They reported a 10% reduction in the heat transfer coefficient due to adding 5% oil compared with the pure refrigerant flow. Wen and Ho [29,30] reported the heat transfer coefficient of condensation and evaporation flows for R290/R600a mixed with Emkarate RL 32H POE lubricating oil in a serpentine U-tube with 2.46 mm diameter. Oil concentration varied from 0 to 5%. It was found that the heat transfer coefficient is enhanced as mass flux, vapor quality, and number of tube bends increases, while it decreases with oil concentration.

Regarding the importance of lubricating oil on the heat transfer characteristics of natural refrigerants inside enhanced tubes, the main purpose of this study is to measure experimentally the effect of mineral lubricant oil (Gulf Eskimo 68) on the heat transfer characteristics of R600a condensation flow inside U-shaped tubes which has not been investigated before to best of authors’ knowledge. The test conditions include a wide range of parameters including mass flux of 140–280 kg/m²s, vapor quality between 0.04 and 0.80, condensation pressure from 510 to 630 kPa, and oil compactness of 0, 1, 1.5, and 2%. The results of this paper can be used in design of condensers for modern refrigeration systems using R600a HC refrigerant as a working fluid.

2. Experimental method

2.1. Experimental apparatus

A schematic of the experimental apparatus for testing the heat transfer coefficient of refrigerant-oil mixture condensation flow is shown in Fig. 1. The set up consists of a gear pump, a Coriolis mass flowmeter, two pre-heaters, an evaporator, a test condenser, and a post condenser. A 373 W gear pump is used to circulate the refrigerant. Refrigerant mass flux can be controlled using a variable frequency drive which is coupled with the pump. In addition, a bypass line is foreseen to provide more precise control over flowrate and make it possible to add oil easily to the system. The mass flowmeter is a Coriolis type (Danfoss-MASS/2160/6000) calibrated precisely to measure mass flow rate up to 250 kg/h with the accuracy of 0.1% full scale. Two pre-heaters are used to increase the refrigerant enthalpy and make it saturated liquid, then the desired vapor quality at the test condenser inlet can be achieved using an evaporator. The pre-heaters and the evaporator are 1-m copper tubes heated with a wire uniformly wrapped around them. Each element can produce up to 3 kW electrical power. The input power of each heater can be controlled by a dimmer and can be measured using a Wattmeter with the precision of 1%. After the test condenser, a post condenser is used to condense the refrigerant completely and to prevent any vapor enters the gear pump. It is a 12 m copper tube which is coil in a 10 cm in diameter polyethylene shell with 60 cm length. The refrigerant is condensed inside the tube by cold water flows inside the shell.

As shown in Fig. 2, the test condensers are double-pipe heat exchangers in which refrigerant flows inside the inner tube. Condenser (a) has a straight tube with inside diameter of 8.7 mm, outside diameter of 9.52 mm, and length of 120 cm which is embedded in a polyethylene shell of 6 cm in diameter. Condenser (b) consists of two straight sections of 135 cm in length and a U-bend with inside diameter of 8.7 mm, outside diameter of 9.52 mm, and the curve ratio of 21 which are installed in a 6 cm polyethylene shell. The test condensers are completely insulated to prevent any thermal leakage. K-type thermocouples are used to measure outside wall temperature at five points in the condenser (a) and at fifteen points in the condenser (b) all around the
Fig. 1. Schematic view of the experimental apparatus.

Fig. 2. Geometry of test condensers and position of thermocouples.
Table 1
Experiments conditions.

<table>
<thead>
<tr>
<th>Refrigerant</th>
<th>R600a (Isobutane)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Lubricant oil</td>
<td>Gulf Eskimo 68</td>
</tr>
<tr>
<td>Mass fraction of oil (%)</td>
<td>1, 1.5, 2</td>
</tr>
<tr>
<td>Mass flow (kg/m²s)</td>
<td>140, 187, 233, 280</td>
</tr>
<tr>
<td>Vapor quality</td>
<td>0.04-0.80</td>
</tr>
<tr>
<td>Condensation pressure (kPa)</td>
<td>510-650</td>
</tr>
<tr>
<td>Cooling water temperature change (°C)</td>
<td>1.6-5.8</td>
</tr>
<tr>
<td>Cooling water mass flow rate (kg/s)</td>
<td>0.003-0.36</td>
</tr>
</tbody>
</table>

Table 2
Properties of the lubricating oil.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Value</th>
</tr>
</thead>
<tbody>
<tr>
<td>Nominal kinematic viscosity @ 40 °C (mm²/s)</td>
<td>68.9</td>
</tr>
<tr>
<td>Flash point (°C)</td>
<td>206</td>
</tr>
<tr>
<td>Pour point (°C)</td>
<td>~36</td>
</tr>
<tr>
<td>Density @ 15 °C (kg/m³)</td>
<td>921</td>
</tr>
<tr>
<td>Conradson carbon residue (CCR)</td>
<td>&lt; 0.01</td>
</tr>
<tr>
<td>Copper corrosion, 1hr @ 100°C</td>
<td>1a</td>
</tr>
<tr>
<td>Total acid number (mg KOH/g)</td>
<td>0.01</td>
</tr>
<tr>
<td>Saponification No. (mg KOH/g)</td>
<td>0.05</td>
</tr>
</tbody>
</table>

Table 3
Uncertainty of different parameters.

<table>
<thead>
<tr>
<th>Parameter</th>
<th>Uncertainty</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter (mm)</td>
<td>± 0.05</td>
</tr>
<tr>
<td>Length (mm)</td>
<td>± 0.05</td>
</tr>
<tr>
<td>Temperature (°C)</td>
<td>± 0.1</td>
</tr>
<tr>
<td>Pressure (kPa)</td>
<td>± 10</td>
</tr>
<tr>
<td>Power (W)</td>
<td>± 1</td>
</tr>
<tr>
<td>Cooling water mass (kg)</td>
<td>± 0.001</td>
</tr>
<tr>
<td>Vapor quality</td>
<td>± 0.005</td>
</tr>
<tr>
<td>Flow rate (m³/s)</td>
<td>± 0.1%</td>
</tr>
<tr>
<td>Mass flux (kg/m²s)</td>
<td>0.1%</td>
</tr>
<tr>
<td>Heat transfer coefficient (W/m²K)</td>
<td>7%</td>
</tr>
</tbody>
</table>

Table 4
The evaluated correlations for pure R600a condensation flow in the straight tube.

<table>
<thead>
<tr>
<th>Correlation</th>
<th>Equation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Cavallini and Zecchin [6]</td>
<td>$h = 0.036 \theta^{0.75} \frac{d_p}{r_p} \left( \frac{v}{v_d} \right)$</td>
</tr>
<tr>
<td>Shah [7]</td>
<td>$h = 0.023 \theta^{0.8} \frac{d_p}{r_p} \left( \frac{x}{1-x} \right)^{0.2} \left( \frac{v}{v_d} \right)$</td>
</tr>
</tbody>
</table>

As shown in Fig. 2. The thermocouples are connected to a data logger system which can measure the temperature with the accuracy of 0.1 K. In addition, at inlet and outlet of the test section, the flow temperature and pressure are measured using two thermocouples and two pressure gauges. It should be noted that before recording any data, the system worked for approximately 30 min until no evidence of unsteady flow was monitored. In addition, the repeatability of the apparatus was tested by repeating the experiments three times at different refrigerant mass fluxes and vapor qualities.

The conditions of experiments are shown in Table 1. Also, properties of the mineral lubricating oil are listed in Table 2.

The uncertainty analysis of Schulz and Cole [31] were performed for all essential parameters in the present study and their results are summarized in Table 3.

2.2. Data reduction

Average inside wall temperature of the tubes can be estimated as:

$$T_{w} = T_{w} + \Delta T_{w} = \frac{q_{d}}{2k_{w}} \ln \left( \frac{d_{o}}{d_{i}} \right)$$ (1)

where the test condenser heat flux is calculated from Eq. (2):

$$q = \frac{m_{cw} c_{pw} \Delta T_{cw}}{\pi d_{i} L}$$ (2)

Defining the isolation coefficient, $\eta$, as heat leakage at the preheaters and the evaporator can be estimated. Therefore, for each of the preheaters and evaporator, the heat given to the refrigerant is obtained:

$$Q_{h} = \eta \times W$$ (3)

where insulation efficiency, $\eta$, is estimated using a similar method of Diani et al. [32, 33] which showed that it varies between 0.88 and 0.92 based on the flow parameters.
For the pure refrigerant flow, vapor quality at the test condenser inlet can be obtained by using the first thermodynamics law over the evaporator:

$$x_i = \frac{\Delta h_p}{q_p(T_{in} - T_{in})}$$

Similarly, using the energy conservation equation for the test condenser, vapor quality at its outlet is as below:

$$x_o = \frac{\Delta h_p}{q_p(T_{out} - T_{out})}$$

Finally, average vapor quality is:

$$x_{ave} = \frac{x_i + x_o}{2}$$

The condensation heat transfer coefficient can be evaluated from the following equation:

$$h = \frac{q}{T_{out} - T_{in}}$$

The presence of oil changes the refrigerant properties. For the refrigerant-oil mixture, thermodynamic characteristics are:

$$c_{p,m} = (1 - \omega_{oil})c_{p,r} + \omega_{oil}c_{p,o}$$

$$i_{f,m} = \frac{m_f}{m_a + m_d}i_{f,r} + \frac{m_d}{m_a + m_d}i_{f,r}$$

$$i_{g,m} = \frac{m_g}{m_a + m_d}(i_o - i_{f,r}) + i_{g,r}$$

where $\omega_{oil}$ is the local oil concentration in the mixture liquid phase and can be calculated from Eq. (11):

$$\omega_{oil} = \frac{\omega_{oil}}{1 - x_{r,o}}$$

where $x_{r,o}$ is vapor quality of the refrigerant-oil mixture which can be determined as:

**Table 5**

<table>
<thead>
<tr>
<th>Ref.</th>
<th>Correlation</th>
</tr>
</thead>
<tbody>
<tr>
<td>Travis et al. [5]</td>
<td>$h = R_{eff}^{0.5}Pr_{eff}^{0.5}L^{0.5} \left( \frac{h_{ref}}{h_{ref}} \right) \times \left( F_i \right)$</td>
</tr>
<tr>
<td>Shah [8]</td>
<td>$h = \left( \frac{H_i}{1 + \frac{H_i}{H_{ref}}} \right)^{0.004} + 0.557Pr^{0.5}</td>
</tr>
</tbody>
</table>

$$\frac{1}{X_0} + \frac{2.85}{X_0^{0.5}}$$

$F_i = 5Pr + 5ln(1 + 5Pr) + 2.5ln\left(0.00313Re^{0.12}\right)$
\[ x_{r,i} = \frac{m_{r,i}}{m_{r,i} + m_{v,i} + m_{o}} \]  

(12)

In the case of the refrigerant-oil mixture flow, the bubble point temperature should be used instead of \( T_{out} \) in Eq. (7). As the boiling point of the lubricating oil is much higher than the refrigerant, the bubble point of the mixture is higher than the refrigerant saturation temperature at the same pressure. The difference is more essential at high vapor quality due to the high local oil concentration in the mixture liquid phase. Using Takaishi and Oguchi [34] correlation which was originally suggested for R22/oil mixture:

\[ T_{hub} = \frac{A}{\ln(P_{sat})} - B \]  

(13)

where A and B are determined through Eqs. (14) and (15), respectively:

\[ A = a_0 + 182.5a_{unc} - 724.2a_{unc}^2 + 3868a_{unc}^3 - 5269a_{unc}^4 \]  

(14)

\[ B = b_0 - 0.722a_{unc} + 2.39a_{unc}^2 - 13.78a_{unc}^3 + 17.075a_{unc}^4 - 5269a_{unc}^5 \]  

(15)

where \( a_0 \) and \( b_0 \) are \(-2395\) and \(8.074\), respectively. Thome [35] modified the abovementioned equations for other refrigerants. For R600a/oil mixture, they proposed \(-2582\) and \(+7.616\) for \( a_0 \) and \( b_0 \), respectively.

Using the energy conservation for the evaporator, vapor quality at the test condenser inlet is obtained as below:

\[ x_1 = \frac{1}{l_{f,m} - l_{f,in}} \left( \frac{Q_k}{m_r + m_o} - c_{p,m} (T_{hub} - T_{in}) \right) \]  

(16)

In addition, vapor quality at the test condenser outlet can be calculated by using the first law of thermodynamics over the test condenser:

\[ x_2 = \frac{1}{l_{f,m,out} + l_{f,m,in} + x_1 l_{g,m,in}} \]  

(17)

So, average vapor quality is:

\[ x_{ave} = \frac{x_1 + x_2}{2} \]  

(18)

Finally, the condensation heat transfer coefficient can be calculated from the following equation:

\[ \dot{h} = \frac{q}{T_{hub} - T_{w}} \]  

(19)
3. Results and discussion

3.1. Validation

To evaluate the experimental procedure, the heat transfer coefficient of pure R600a condensation flow in the straight tube is compared with two well-known correlations, including Cavallini and Zecchin [6] and Shah [7] ones, as summarized in Table 4. It should be noted that both correlations are valid for the operating conditions of this study. As shown in Fig. 3, all experimental data lie within −20% to +13% and −9% to +20% of Cavallini and Zecchin [6] and Shah [7] predictions, respectively, which reveals the validity of the procedure.

3.2. Flow regime

It is well known that the flow regime is an important matter in the condensation heat transfer behavior. In this study, El Hajal et al. [36] flow pattern map is used to identify the flow regime in the straight tube, as well as, at the inlet of the U-shaped tube based on the experimental data. Fig. 4 shows the predicted El Hajal et al. [36] for various mass fluxes of pure R600a flow regime. According to the results, most of the experimental results are in the annular region so that only in low vapor quality the flow regime is intermittent (a stratified-wavy flow pattern with large amplitude waves that wash the top of the tube). Some previous research works showed that adding oil in the range of 1–2% mass concentration, which used in this study, only accelerates the flow regime transition to the annular ones [27] and, as a result, has no major effect on the flow pattern in the current study.

3.3. Heat transfer coefficient

The results are presented in three parts, including the pure refrigerant flow inside the straight tube, the pure refrigerant flow inside the U-shaped tube, and the refrigerant-oil mixture flow with the oil mass concentration of 1, 1.5, and 2%. In each part, the effect of main parameters, e.g. refrigerant mass flux and vapor quality, on the condensation heat transfer behavior is investigated.

The effect of vapor quality and mass flux on the heat transfer coefficient of pure R600a condensation inside the straight tube is shown in Fig. 5. Based on the results, the heat transfer coefficient increases with vapor quality as well as mass flux. In addition, the effect of mass flux on the heat transfer coefficient is more severe at high vapor quality. Other researchers such as Agra and Teke [37] and Shah [8] reported similar behavior for condensation of pure refrigerants inside straight tubes.

As the dominant flow regime is annular, the condensate forms a liquid film on the inner wall of the tube. The film is not uniform in thickness, and it is thicker at the bottom of the tube due to the gravitational force. The conductive thermal resistance of this liquid film plays an important role in the condensation heat transfer phenomenon. At high vapor quality, the condensate liquid film is thin and its conductive thermal resistance is low, accordingly. In addition, in these conditions, the relative velocity between liquid and vapor phases is high which leads to high interfacial advection. Because of both reasons, the heat transfer coefficient enhances with vapor quality. Regarding the effect of mass flux on the heat transfer coefficient, increment of mass flux increases the liquid-vapor interfacial shear stress which makes the liquid film thickness more uniform and reduces its thermal resistance. This contribution is more effective at high vapor quality region. Also, at high mass flux, there is high advection between liquid and vapor phases which increases the heat transfer coefficient, in turn.

Fig. 6 shows the effect of vapor quality and mass flux on the condensation heat transfer coefficient of R600a inside the U-shaped tube. The general behavior is similar to the flow inside the straight tube, as discussed previously. Lee et al. [10] reported a similar trend for the condensation of some other refrigerants inside U-shaped tubes. Based on the results, using the U-shaped tube at the mass flux of 140, 187, 233, and 280 kg/m²s leads to the average heat transfer coefficient enhancement up to 16.2, 20.5, 26.1, and 29.7%, respectively, in comparison to the straight tube. The presence of curvature in the tube forms a double-vortex secondary flow which causes the condensate liquid film thickness to be more uniform and reduce its conductive thermal resistance, accordingly. In addition, due a secondary flow, the interfacial advection heat transfer between liquid and vapor phases is enhanced. The curvilinear acceleration due to the U-bend may also entrain liquid droplets from the condensate film to the vapor phase. This feature reduces the liquid film thickness, in turn, and increases the heat transfer coefficient.

The experimental heat transfer coefficient of pure R600a condensation flow in the U-shaped tube is compared with Traviss et al. [5] and Shah [8] correlations (see Table 5) in Fig. 7. According to the results, although these correlations were originally suggested for straight tubes, in the lack of a reputable correlation, they can provide good predictions for U-shaped tubes as well, as also used by Lee et al. [10].

In Fig. 8, the effect of oil concentration and vapor quality on the condensation heat transfer coefficient of the refrigerant-oil mixture flow inside the U-shaped tube is shown at different mass velocities. Enhancement or deterioration of the heat transfer coefficient depends on the miscibility of oil into the refrigerant as well as mass flux, vapor quality, and oil concentration. Viscosity and density of the refrigerant-oil mixture are more than the pure refrigerant. Also, adding the oil to the refrigerant leads to an increase in surface tension which accelerates flow regime to the annular region. Therefore, the effect of oil on the flow condensation heat transfer may be positive, insignificant or negative based on the operating conditions. Due to experimental limitations, the oil concentration is limited to 2% in this study. Some previous research works showed that adding more than 2% of oil to other refrigerants causes a sharp reduction in the heat transfer coefficient due to thermal resistance increment [27,29]. However, as the operating conditions in their works were different from ours, this conclusion should be used with caution.

As shown in Fig. 8, it can be concluded that in low vapor quality, the refrigerant-oil mixture has higher heat transfer coefficient than the pure refrigerant which is due to increment of wetted tube surface in the presence of oil and the high thermal conductivity of the oil. By increasing vapor quality, the process of heat transfer enhancement slows down so that after vapor quality of about 0.4–0.5, the mixture heat transfer coefficient falls below the pure refrigerant one. The reason for such a trend is that by decreasing the amount of liquid phase, the concentration of oil increases locally in the liquid film even by about ten times. Increment of the oil concentration makes the liquid film flow regime from turbulent to laminar. As a consequence, the effect of advection heat transfer mechanism at the liquid-vapor interface reduces which results in a decrease in the heat transfer coefficient. Cho and Tae [27] and Wen and Ho [29] reported approximately similar behavior for HFC refrigerant-oil mixture flow inside U-tubes.

4. Conclusion

By investigating the entire experiments database, the following findings were achieved:

- The condensation heat transfer coefficient increases with mass flux and vapor quality for the pure refrigerant flow inside both the straight and the U-shaped tubes.
- For the pure refrigerant flow inside the U-shaped tube, the maximum local and average heat transfer coefficients enhancement in comparison to the straight tube were 34% and 29.7%, respectively, at the mass flux of 280 kg/m²s.
- Heat transfer coefficient of pure R600a condensation flow may be predicted using Cavallini and Zecchin [6], and Shah [7] correlations for the straight tube and Traviss et al. [5] and Shah [8] correlations
for the U-shaped tubes with reasonably good accuracy.

- The refrigerant-oil mixture flow had higher heat transfer coefficient than the pure refrigerant flow in low vapor quality; while at medium and high vapor quality, the pure refrigerant flow showed higher heat transfer coefficient.

- Maximum and minimum changes of the refrigerant-oil mixture local heat transfer coefficient, related to 2% oil concentration at 280 kg/m²s mass flux, were +33% and −29%, respectively.

References


