Research paper

Effect of lubricating oil on flow boiling characteristics of R-600a/oil inside a horizontal smooth tube

M.R. Momenifar, M.A. Akhavan-Behabadi*, M. Nasr, P. Hanafizadeh

Center of Excellence in Design and Optimization of Energy Systems, School of Mechanical Engineering, College of Engineering, University of Tehran, Tehran, Iran

A R T I C L E   I N F O

Article history:
Received 28 May 2015
Accepted 1 August 2015
Available online 8 August 2015

Keywords:
Flow boiling
Horizontal tube
R-600a
Lubricant oil
Two-phase flow

A B S T R A C T

The aim of the present study is to experimentally investigate the effect of oil on flow boiling of R-600a as a hydrocarbon refrigerant inside a horizontal smooth tube. In order to study the effect of oil on the flow boiling of R-600a, a well instrumented apparatus was designed, fabricated and installed. The experimental conditions of this study include nominal oil concentrations from 0% to 2.5%, mass velocities in range of 130–380 kg/m²s, inlet vapor qualities from 0.05 to 0.77 and heat fluxes from 10 to 28 kW/m²which were conducted in a copper test tube with the inner diameter of 8.7 mm. Several parameters affecting the heat transfer coefficient and pressure drop of refrigerant–oil mixture, such as oil concentration, mass flux and vapor quality were investigated. The comparison between two-phase pressure drop for pure refrigerant and the refrigerant oil mixture reveals that pressure drop increases with the increase of oil concentration in all ranges of vapor quality and mass velocities. However, heat transfer coefficients tend to increase at low vapor qualities and decrease at the middle and high vapor qualities due to presence of oil.

© 2015 Elsevier Ltd. All rights reserved.

1. Introduction

The presence of oil inside refrigerant systems is unavoidable since the oil is required to lubricate the moving parts of the compressor. Oil can affect the flow boiling of the refrigerant by its changing thermal conductivity, surface tension, flow pattern and other properties of refrigerant during flow boiling. Therefore, the effect of oil on the flow boiling heat transfer is important to be investigated. In the open literature, a great number of papers including refrigerant–oil mixture can be found for different test condition and refrigerants.

Cho and Tae [1] reported a 20% increase in pressure drop and a 10% reduction in heat transfer compared with the pure refrigerant during evaporation of refrigerants (R22 and R-407C) with the presence of 5.0% oil. Gao and Honda [2] conducted an experimental study on flow boiling heat transfer of CO₂ (R-744) and oil (PAG) mixtures inside a horizontal tube with an inner diameter of 3 mm. According to their study when the concentration of the lubricant oil in weight was more than 1%, the local heat transfer coefficient was much lower than that without the lubricant oil.

Zheng et al. [3] studied the flow boiling of an ammonia–lubricant oil mixture inside horizontal plain tube and they concluded that the heat transfer increased with evaporation temperature and applied heat flux. Dang et al. [4] conducted a study on flow boiling of CO₂–lubricant oil mixtures in smooth tubes. They used synthetic PAG lubricant oil and their results showed that for a small quantity of this oil in the system, 0.5% in weight, the heat transfer coefficient reduced significantly, less than 50% in comparison with pure CO₂. When the oil concentration was increased to 1.0 or 5.0%, the heat transfer coefficient remained similar to the results obtained for 0.5%.

Hu et al. [5] performed a study on heat transfer and pressure drop in flow boiling of R-410a–oil mixture inside horizontal smooth tube and they reported that the presence of oil enhanced the heat transfer at low and intermediate vapor qualities. Their pressure drop results showed an increase in all vapor qualities. Bandarra Filho et al. [6] presented a comprehensive review of flow boiling characteristics of refrigerant–lubricant oil mixtures. They summarize that there is no clear agreement on the effect of oil. According to their report, an increase in surface tension and foam formation (especially at lower mass fluxes) may lead to a better wetting of the tube wall and thus enhancing heat transfer. On the other hand, frictional pressure drop in general increases with the oil concentration due to an increase in liquid viscosity.
Dang et al. [7] addressed experimental studies on the flow boiling heat transfer of carbon dioxide with PAG-type lubricating oil entrained from 0% to 5% in horizontal smooth tubes with inner diameter of 2–6 mm. Their experiments were conducted at mass fluxes of 360–1440 kg/m²s, heat fluxes of 4.5–36 kW/m² and the saturation temperature of 15 °C. They found that the heat transfer coefficient declined by less than half compared to pure refrigerant at low oil concentrations of 0.5–1%, while it did not decrease with increasing oil concentration. Also they reported that presence of oil caused the mass flux to significantly influence the heat transfer coefficient at a low heat flux till dryout, and the dryout quality decreased at a large mass flux. The measured pressure drops increased monotonously thanks to the formation of an oil layer along the tube’s inner wall and an increase in viscosity due to the entrainment of lubricating oil in CO₂.

Wetzel et al. [8] investigated the flow boiling heat transfer and pressure drop of CO₂—oil mixtures at low temperatures inside a 14 mm smooth tube under near isothermal wall condition. In their test conditions, the nominal concentration of oil (0–3 wt%), saturation pressure (14.3–26.4 bar), mass flux (75–300 kg/m²s), inlet vapor quality (0.1–0.9) and heat flux (0–100 kW/m²) were varied. They observed a decline in heat transfer coefficients and a considerable increase in pressure drop with increasing local oil concentration (i.e. nominal oil concentration and vapor quality) relative to measurements with oil-free conditions. Only for low oil concentrations (1 wt%) and medium vapor qualities (0.5 < x < 0.7) a slight improvement in average heat transfer were seen. Regarding their local measurements, the addition of oil can lead to a significant decrease in liquid heat transfer coefficient (i.e. bottom tube segments). On the other hand an increase in tube wetting induced by foam formation resulted in a growth of local heat transfer coefficients at the top of the tube. By using refrigerant mixture properties, the heat transfer correlations could not describe the experimental data satisfactorily.

Recently, Li et al. [9] have proposed a correlation for flow boiling heat transfer based on experimental data for mixtures of CO₂ and partial miscible PAG oil by Dang et al. [7]. The correlation is based on cubic superposition of convective and nucleate boiling contribution terms as proposed by Steiner and Taborek [10].

Different studies published in the open literature shows different results about the effect of oil on the heat transfer performance under flow boiling condition of refrigerant—oil mixture. Some researchers reported decrease of heat transfer with oil concentration and some investigators observed an increase of heat transfer at low oil concentration and low vapor qualities. One possible explanation for the discrepancy with results may be explained by the fact that, the heat transfer characteristics of refrigerant—oil mixture highly depends on the type of refrigerant, type of the oil, oil concentration and the test conditions. Miscibility, surface tension and viscosity are main properties of oil which can change the properties of refrigerant.

R-600a as a hydrocarbon refrigerant was chosen in this study. Due to environmental issues like ozone depleting and global warming, natural refrigerants such as ammonia (R-717), carbon dioxide (R-744) and hydrocarbons such as isobutane (R-600a) have been studied to replace CFCs, HCFCs and HFCs in refrigeration, air-conditioning and heat pump systems. Comparing the Ozone Depletion Potential of R-600a (0) as a hydrocarbon refrigerant with R-12 (1) as a CFC refrigerant and 100 years Global Warming Potential of R-600a (20) with R-134a (1370) as an HFC refrigerant shows that R-600a has a better environmentally performance [Molina and Rowland [11]–Kurylo [12]].

Hydrocarbons present great advantages such as thermodynamic and transport properties, small molecular weight, and appropriate compatibility with lubricants, compared to the disadvantage of low limit of flammability. Moreover, some recent studies show that R-600a has also better energy performance comparing with other refrigerant. Calm [13] addressed that R-600a have taken the place of R-12 and later R-134a and nowadays dominates in the domestic refrigerators in Europe. Therefore, environmentally advantages and different properties of hydrocarbons refrigerants compared with conventional refrigerants make it necessary to investigate the effect of oil on flow boiling hydrocarbons refrigerants.

As noted above none of the mentioned studies have investigated the oil effect on flow boiling of hydrocarbons refrigerants. The aim of this study is to investigate the convective boiling characteristics of R-600a—oil mixture and the results of this paper could be used in design of evaporators in refrigerators with R-600a—oil mixture as a working fluid.

2. Experimental apparatus

2.1. Description

The schematic of the experimental apparatus, for testing the heat transfer coefficient and the pressure drop of refrigerant—oil mixture, including variable frequency gear pump, coriolis-effect mass flow meter, preheaters, test evaporator, by-pass section,
subcooler and condenser is shown in Fig. 1. The preheaters were installed upstream of the test evaporator to obtain desired vapor quality at the inlet of a test evaporator so as to cover the entire ranges of vapor qualities. The test evaporator was a smooth horizontal copper tube with inside diameter of 8.70 mm and thickness and length of 0.41 mm and 110 cm respectively.

Electrical resistance heater wrapped around the test section uniformly and the test section was insulated by glass wool pad to reduce the heat loss to the surroundings. The outside tube wall temperature was measured by K type thermocouples with the accuracy of ±0.1%. These thermocouples consisted of chrome-aluminum wires with diameter of 0.5 mm and were attached to the top, bottom, and sides of the test section at 6 stations along the test evaporator (Fig. 2).

The average temperature was used as the temperature of the whole test section. The inlet pressure of the test section was monitored by digital pressure indicator with the accuracy of 1 kPa. A precision pressure drop apparatus was used to measure pressure drop across the test section with the accuracy of 0.075% of the full scale. The saturation temperature at the average pressure of the test section was considered as the saturation temperature of the whole test section.

The preheaters were copper tube with outer diameter of 9.52 mm and were heated with an electrical heater uniformly wrapped around them similar to the test evaporator. The input power for each heater was controlled by a 2 kW dimmer. Although, the evaporator and the preheater were completely insulated to prevent any heat leakage, the insulation efficiency was calculated to consider heat loss and the net heat rates were used to calculate vapor qualities and heat transfer coefficients. A variable frequency gear pump with the power of 373 W and the maximum pressure of 1000 kPa was used to circulate the refrigerant into the cycle and compensate for pressure drop. The condenser was a shell and tube cross flow heat exchanger. Water was used as a coolant for condensation of refrigerant. It consisted of a 12 m long coiled tube and 0.7 m long shell. The subcooler was a shell and tube cross flow heat exchanger similar to condenser with coiled tube of 10 m long and shell tube of 0.6 m long. It was used to ensure that refrigerant has been changed to the compressed liquid before entering to the pump and to adjust the saturation temperature at the inlet of test evaporator.

The flow meter was a coriolis mass flow meter (Danfoss-MASS/2100/6000) with the accuracy within 0.1% of the full scale which measure mass flow rate up to 0.07 kg/s with in the temperature ranges from –150 to 180 °C. Thermocouples in the test evaporator were connected to a 24 channel data logger (Lutron-4208 SD) which can be used for K and T type thermocouples.

ISO-Butane (HR-600a) with the purity of 99.5% was used as the working fluid. During each test run, three sets of data were taken at intervals of 10 min each after the system reached steady state condition. An average of three readings was used for further analysis.

The mass velocity was controlled by an inverter which was coupled with the variable frequency pump. Moreover a bypass section is used to control the mixture flow rate (to the main loop) and to inject the oil and into the refrigerant. In by-pass loop, the remaining portion of liquid mixture enters to condenser. In order to inject the oil the following steps are performed.

Fig. 1. Schematic of the experimental apparatus.
a. The initial and terminal valves are closed (valves No. 2 and 3 in Fig. 1).
b. The bypass path is vacuumed by vacuum pump.
c. The oil is injected to the bypass by syringe from the valve that is located in the middle of the bypass line (valve No. 3 in Fig. 1).
d. The middle valve (valve No. 1) is closed and two other valves (valves No. 2 and 3) are opened.

After injection of the oil, the pump works for 3 h to make the mixture homogenized.

The used lubricating oil is a naphthenic refrigeration compressor oil with commercial name “Gulf Eskimo 68”, nominal kinematic viscosity of 68 mm²/s at 40 °C and density of 921 kg/m³ at 15 °C, as reported by the manufacturer. The oil is completely miscible with isobutane refrigerant. In fact, mineral oil in this refrigerant is fully miscible in the liquid state of the refrigerant but with the increasing of the vapor quality and increasing of the local oil concentration into the liquid film, the oil may become immiscible into the refrigerant. Table 1 shows the experimental conditions of all experiments. With determining temperature values of refrigerant at that position. With determining cooled liquid state) was calculated on knowing pressure and temperature values of refrigerant at the inlet of the test evaporator section was measured.

3. Data reduction

All thermo physical and properties of pure R-600a were evaluated using commercial software (Engineering Equation Solver, EES). Enthalpy of the refrigerant at the inlet of the first preheater (at sub-cooled liquid state) was calculated on knowing pressure and temperature values of refrigerant at that position. With determining insulation efficiency and adjusting input power of heaters, enthalpy of working fluid entering the test section was measured. To calculate the insulation efficiency of the preheaters and test evaporator, electrical powers are adjusted to have the superheated state at the outlet of the test evaporator. Qe1, Qe2 and Qe3 are the electrical powers delivered to the preheater1, preheater2 and test evaporator respectively (Fig. 3). The value of insulation efficiency was obtained equal to 0.9. Since the insulation method for preheaters and the test evaporator was the same, this value was considered as the insulation efficiency for all of them.

3.1. Oil concentration

Nominal oil concentration (ωno) is defined by Eq. (1).

\[ ω_{no} = \frac{m_o}{m_o + m_R} \]  

where \( m_o \) is the mass flow rate of oil, kg/s; \( m_R \) is the mass flow rate of refrigerant liquid phase, kg/s and \( x_R \) is the vapor quality of refrigerant—oil mixture.

3.2. Local vapor quality

For refrigerant—oil mixture, the local vapor quality \( x_{R,o} \) is defined by Thome [16] and is presented as Eq. (3).

\[ x_{R,o} = \frac{m_{R,g}}{m_{R,g} + m_{R,L} + m_o} \]  

where \( m_{R,g} \) is the mass flow rate of refrigerant vapor phase, kg/s.

The inlet and outlet vapor qualities of the test section are obtained as follows.

\[ (x_{R,o})_{in} = \frac{(m_o + m_R)h_{R,o,in} - m_o h_{o,in} - m_R h_{R,L,in}}{(m_o + m_R)(h_{R,g,in} - h_{R,L,in})} \]  

\[ (x_{R,o})_{out} = \frac{(m_o + m_R)h_{R,o,out} - m_o h_{o,in} - m_R h_{R,L,out}}{(m_o + m_R)(h_{R,g,out} - h_{R,L,out})} \]  

Vapor quality in all over of the evaporator section is calculated by Eq. (6).

\[ (x_{R,o})_{ave} = \frac{(x_{R,o})_{in} + (x_{R,o})_{out}}{2} \]  

The specific enthalpies of refrigerant—oil mixture at the inlet and outlet of the evaporator section are computed by the heat balance equations as below:

\[ h_{R,o,in} = h_{R,o,1} + \frac{Q_{pre}}{(m_o + m_R)} \]  

Table 1

<table>
<thead>
<tr>
<th>Test section</th>
<th>Copper tube with inside diameter of 8.70 mm</th>
</tr>
</thead>
<tbody>
<tr>
<td>Refrigerant</td>
<td>R-600a (isobutane)</td>
</tr>
<tr>
<td>Lubricant oil</td>
<td>Gulf Eskimo 68</td>
</tr>
<tr>
<td>Mass fraction of oil</td>
<td>1%, 1.5%, 2%, 2.5%</td>
</tr>
<tr>
<td>Mass velocity (kg/m²s)</td>
<td>130–380</td>
</tr>
<tr>
<td>Vapor quality</td>
<td>0–0.7</td>
</tr>
<tr>
<td>Heat flux (kW/m²)</td>
<td>10–27</td>
</tr>
<tr>
<td>Pressure (bar)</td>
<td>5–6</td>
</tr>
</tbody>
</table>

Table 2

<table>
<thead>
<tr>
<th>Parameters</th>
<th>Uncertainties</th>
</tr>
</thead>
<tbody>
<tr>
<td>Diameter</td>
<td>0.05 (mm)</td>
</tr>
<tr>
<td>Length</td>
<td>±0.05 (mm)</td>
</tr>
<tr>
<td>Pressure</td>
<td>±10 (kPa)</td>
</tr>
<tr>
<td>Power</td>
<td>±1 (W)</td>
</tr>
<tr>
<td>Temperature</td>
<td>±0.1 °C</td>
</tr>
<tr>
<td>Heat transfer coefficient</td>
<td>8.0 W/m²K</td>
</tr>
<tr>
<td>Pressure drop</td>
<td>375–450 (kPa)</td>
</tr>
<tr>
<td>Vapor quality</td>
<td>±0.005</td>
</tr>
<tr>
<td>Mass velocity</td>
<td>0.1% kg/m²s</td>
</tr>
</tbody>
</table>

**Fig. 3. Schematic diagram of test section and preheaters.**
\[ (x_{R,o})_{ave} = \frac{(x_{R,o})_{in} + (x_{R,o})_{out}}{2} \]  
\[ h_{R,o,out} = h_{R,o,in} + \frac{Q_{test}}{(m_0 + m_R)} \]

where \( h_{R,o,in} \) and \( h_{R,o,out} \) are the specific enthalpies of refrigerant–oil mixture at the inlet and outlet of the evaporator section respectively, in kJ/kg; \( h_{R,o} \) is the specific enthalpy of refrigerant–oil blend at the pre-evaporator inlet, kJ/kg; \( Q_{test} \) is the measured heat transfer rate into the refrigerant from the test-evaporator, W; \( Q_{test} \) is the total pressure drop during test section measured by differential pressure indicator. This value consisted of frictional pressure drop \( (\Delta P_{fi}) \), momentum pressure drop \( (\Delta P_{mom}) \) and static pressure drop \( (\Delta P_{sta}) \).

\[ \Delta P_{tot} = \Delta P_{fi} + \Delta P_{mom} + \Delta P_{sta} \]  
\[ \Delta P_{mom} = G^2 \left\{ \frac{(1 - \chi)^2}{\rho_f (1 - \varepsilon)} + \frac{x^2}{\rho_g \rho_f} \right\}_{\text{out}} - \left\{ \frac{(1 - \chi)^2}{\rho_f (1 - \varepsilon)} + \frac{x^2}{\rho_g \rho_f} \right\}_{\text{in}} \]  

in which \( \chi \) is the void fraction and is calculated by Steiner [18] correlation expressed by Eq. (16). In this equation, liquid density and surface tension are affected by the presence of oil and their values are calculated by the correlations of refrigerant/mixture density (Bandarra Filho [6]) and surface tension (Jensen and Jackson [19]). Since the test evaporator is horizontal static pressure drop is equal to zero. Finally frictional pressure drop is obtained by deducting momentum pressure drop from measured total pressure drop.

\[ \varepsilon = \frac{X}{P_f} \left[ (1 + 0.12(1 - X)) \left( \frac{X}{\rho_g} + \frac{(1 - X)}{\rho_f} \right) + \frac{1.18(1 - X) [g \sigma (\rho_f - \rho_g)]^{0.25}}{G^2 \rho_f^{0.5}} \right]^{-1} \]  

is the thermal conductivity of the copper tube; \( L \) is the effective heating length. The average of the thermocouple readings at up, down and both sides of each section was considered as the temperature of that station \( T_{W,i} \). 

\[ T_{hub} = \frac{A}{\ln(p_{sat}) - B} \]  
\[ A = a_0 + 182.5(\omega_{loc}) - 724.2(\omega_{loc})^3 + 3868(\omega_{loc})^5 - 5269(\omega_{loc})^7 \]  
\[ B = b_0 - 0.722(\omega_{loc}) + 2.39(\omega_{loc})^3 - 13.78(\omega_{loc})^5 + 17.075269(\omega_{loc})^7 \]  

where values of \( a_0 \) and \( b_0 \) are \(-2395 \) and \( 8.074 \), respectively, with \( p_{sat} \) in MPa and \( T_{hub} \) in Kelvin. Thome [16] proved this approach also works for other refrigerant–oil mixture by utilizing the corresponding values of \( a_0 \) and \( b_0 \) for the pure refrigerant (the type of lubricant oil is not significant). In this study the values of \( a_0 \) and \( b_0 \) for R-600a from saturation pressures between 3 and 6 (bar) have been determined \(-2582 \) and 7.616, respectively.

4. Results and discussion

The presence of lubricating oil may enhance or deteriorate the transfer coefficient depending on the miscibility of oil into the refrigerant, mass velocity, vapor quality, oil concentration and other various conditions. Therefore the effect of oil on flow boiling heat transfer may be positive, insignificant or high negative depending on the operating condition.

4.1. Heat transfer

The effect of mass velocity and vapor quality on flow boiling heat transfer coefficient of refrigerant oil mixture was investigated. “Fig. 4” shows the variation of heat transfer coefficient versus vapor quality for different mass velocities under flow boiling condition of refrigerant oil mixtures with 4 different oil concentrations.

The results show that the heat transfer coefficient increases with the increase of vapor quality for low vapor qualities until it gets to the maximum value and then it decreases at high vapor qualities. The increase of heat transfer coefficient with the increase of vapor quality from low to intermediate vapor qualities can be explained same as with the pure refrigerant. Since the liquid...
convection is the main mechanism in the present study test conditions, the heat transfer coefficient increases with the increase of vapor quality. By increase of the vapor quality, the thickness of the liquid film on the inside wall of the tube decreases which results in augmentation of heat transfer coefficient due to the decrease of the thermal resistance. On the other hand, increase in the vapor quality leads to increase in the local concentration of oil in liquid film which results in an increase of local liquid viscosity. This trend, which can be justified with Thome [20] correlation for viscosity of refrigerant/oil mixture \( \mu_{\text{mix}} = \mu_{\text{ref}} \left\{ 1 - \omega_\text{oil} \right\} \), accounts for the sharp reduction of heat transfer at high vapor qualities.

In addition, by increasing of the mass velocity, the heat transfer coefficient of refrigerant mixture is increased for all oil concentration similar to pure refrigerant. Hihara et al. [21] explained that the vapor-to-droplet heat transfer coefficient increases with an increase in mass flux, which results in the alleviation of the thermodynamic non-equilibrium (a process in which the vapor becomes superheated to evaporate liquid droplets suspended in the bulk vapor phase.) and an increase in the heat transfer coefficient with the vapor quality. According to Bandarra Filho et al. [6], heat transfer increased with the mass velocity. It should be noted that different mass velocities provide different flow patterns in a way that the foaming formation on the liquid–vapor interface and appearance of froth flow can be seen at low mass velocities (stratified/wavy flow) and high mass velocities (annular flow) respectively.

"Fig. 5" shows the heat transfer coefficient ratio (with/without oil) versus vapor quality for different mass velocities. The maximum enhancement is 49% for the highest oil concentration of 2.5% at the vapor quality of 0.31 and mass velocity of 380 kg/m²s. The reason for heat transfer enhancement due to oil presence during flow boiling is not yet well understood. However, some possible explanations are presented for that. As an illustration the increased wettability of the mixture due to the higher surface tension of the mixture may be responsible for heat transfer enhancement at low vapor qualities.

According to this figure oil increases the local heat transfer coefficient at low vapor qualities and decreases the heat transfer coefficient at high vapor qualities with respect to the pure refrigerant. The increased surface tension of the mixture with respect to the
pure refrigerant can increase the heat transfer coefficient by increasing the wetted fraction of tube perimeter and accelerating the change of flow pattern from intermittent to annular which can be responsible for oil’s heat transfer augmentation effect at low to medium vapor qualities. Moreover, the higher thermal conductivity of the mixture can be considered as another reason for the enhancement at this region. The results showed that for the vapor qualities approximately more than 0.6 the heat transfer coefficient decreases. In fact, at high vapor qualities, the effect of growing local oil concentration and viscosity want to decrease heat transfer coefficient, whereas increased wettability due to the higher surface tension want to reverse this trend. In this competition, the reducing factor would overcome the increasing factor and become the dominant factor and therefore the heat transfer coefficient decrease at high vapor qualities. The local oil concentration increases with the increase in vapor quality because the oil remains into the liquid phase of the refrigerant due to its higher boiling temperature. Increased viscosity can result in the decrease in the Reynolds Number of flow and consequently the convection heat transfer process which is the dominant mechanism at this region will be decreased. Zurcher et al. [22] observed in the all ranges of mass velocities (100 and 300 kg/m²s) that at higher qualities, above 70%, the lubricant oil negatively influenced heat transfer.

Since there is not a published paper that consider the effect of oil on convective boiling heat transfer of R-600a, the results of this study cannot be compared quantitatively, but the heat transfer behavior of refrigerant/oil mixture of this study is similar to Hu et al. [5] that studied heat transfer in flow boiling of R-410a—oil mixture inside horizontal smooth tube and reported that the presence of oil enhanced the heat transfer at low and intermediate vapor qualities. Furthermore, these results are similar to the research of Saitoh et al. [23] on a miscible refrigerant/oil mixture (HFO1234yf + PAG oil) who reported that due to the foaming and the increase in wetted surface, the heat transfer coefficient is enhanced in the low vapor qualities, whereas its magnitude decrease in the high vapor quality region.

**Fig. 5.** Heat transfer ratios for flow boiling of R600-a/oil mixture at 6 different mass velocities and 4 different oil concentrations.
4.2. Pressure drop results

It should be mentioned that the pressure drop characteristics of isobutane–oil mixture flow boiling are similar with the mixture of oil and other kinds of refrigerant in horizontal smooth tubes. As authors in literature mentioned, escalation in viscosity and as a result increase in the possibility of foam production, increase of local oil concentration in high qualities are factors which increase the value of two phase pressure drop.

The experimental results are shown in Fig. 6 and reveal that the two-phase pressure drop increases with vapor quality and oil concentration under each mass velocity. This increase of pressure drop with quality can be justified due to the fact that the increase of vapor quality causes decreasing of average flow density, thus, the velocity increases to keep mass flux fixed. By considering the proportional relation between shear stress and mass velocity, this behavior is expectable. Increase of local oil concentration in liquid film, particularly at high vapor qualities, as a result of its higher boiling temperature respect to pure refrigerant intensify the value of pressure drop gradient. Besides, it can be justified on the transition of intermittent flow pattern to annular flow pattern and the fact that annular regime brings about more pressure drop than intermittent one. As discussed in previous section, increasing in surface tension because of oil presence results in this rapid transition.

This trend is in complete accord with results of Dang et al. [4] who concluded that increase in liquid viscosity (because of oil concentration enlargement) leads to the formation of a thicker film on the wall and consequently the pressure drop increase in comparison with pure refrigerant.

"Fig. 7" shows the variation of frictional pressure drop ratio (with oil/without oil) versus vapor quality for different mass velocities and different oil concentrations. The maximum pressure drop augmentation with respect to the pure refrigerant is 69% which happens at the vapor quality of 0.71 and mass velocity of 275 kg/m²s and for the highest oil concentration of 2.5%.

---

**Fig. 6.** Flow boiling frictional pressure gradient of 600-a/oil mixture at 6 different mass velocities and 4 different oil concentrations.
4.3. Comparison with conventional correlations

The experimental results of present study, including about 240 data, were compared with that predicted by conventional heat transfer correlations (Gungor–Winterton [24], Chen [25] and Wattelet [26]) and two-phase pressure drop models (Muller-Steinhagen and Heck [27], Gronnerud [28], Friedel [29] and Cicchitti [30]). In fact, among considering various models, only aforementioned ones were within a deviation of ±40%. Common correlations originally were proposed for pure refrigerant and in this paper the mixture properties of R600-a/oil were applied in these predictions. Tables 3 and 4 show statistical analysis of this comparison for heat transfer coefficient and two-phase pressure drop correlations respectively.

In these tables, the mean relative deviation ($\gamma$), the mean absolute deviation ($\lambda$) and error bandwidth ($e$) are defined by Eqs. (17)–(19) as below:

$$\gamma = \frac{1}{N} \sum_{i=1}^{N} \frac{\text{value calculated}(i) - \text{value experimental}(i)}{\text{value experimental}(i)}$$  \hspace{1cm} (17)

$$\lambda = \frac{1}{N} \sum_{i=1}^{N} \frac{\text{value calculated}(i) - \text{value experimental}(i)}{\text{value calculated}(i)}$$  \hspace{1cm} (18)

$$e = \frac{\text{value calculated}(i) - \text{value experimental}(i)}{\text{value experimental}(i)}$$  \hspace{1cm} (19)

Despite the modification of thermo-physical properties of R600-a/oil mixture, these correlations are not in good agreement with

Table 3
Prediction deviation of heat transfer coefficient correlations.

<table>
<thead>
<tr>
<th>Prediction method</th>
<th>$\gamma$ (%)</th>
<th>$\lambda$ (%)</th>
<th>$e$ (%)</th>
<th>$\delta$ (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Chen [25]</td>
<td>-7.7105449</td>
<td>17.0672392</td>
<td>-20 to 20</td>
<td>71</td>
</tr>
<tr>
<td>Wattelet [26]</td>
<td>1.09894657</td>
<td>22.7102504</td>
<td>-30 to 20</td>
<td>76</td>
</tr>
</tbody>
</table>

$\delta$: the percentage of points predicted within an error bandwidth.

Fig. 7. Frictional pressure drop ratios for flow boiling of R600-a/oil mixtures for 6 different mass velocities and 4 different oil concentrations.
4.4. Developing a heat transfer coefficient correlation for refrigerant/oil mixtures

It was shown in the previous section that the correlations of pure refrigerant with modified properties, cannot estimate the value of heat transfer coefficient well. In the operating condition of this paper, the Chen [25] correlation had the least deviation with experimental data. To develop the heat transfer coefficient correlation for refrigerant/oil mixture, an effective method is to use the oil impact factor, as the function of the oil concentration, to correct the heat transfer coefficient of pure refrigerant. Therefore, the oil impact factor equation combining Chen correlation is recommended for predicting the heat transfer coefficient of refrigerant/oil mixture. The heat transfer coefficient of refrigerant/oil mixture can be calculated by Eq. (20):

\[ h_{\text{ref-oil}} = E_{\text{oil}} \times h_{\text{chen}} \]  

(20)

In this equation, \( E_{\text{oil}} \) is the oil impact factor and it is a function of the liquid Reynolds number (\( R_{\text{ef}} \)) and Martinelli ([31]) parameter (\( X_{\text{ef}} \)) and can be obtained with below equations. It is important to note that \( h_{\text{chen}} \) is measured by using refrigerant/oil mixture properties.

\[ E_{\text{oil}} = \exp \left\{ \omega_{\text{oil}} \left( C_{\text{m}}^{\text{m}} + D_{\text{ref}}^{\text{f}} \right) \right\} \]  

(21)

\[ X_{\text{ef}} = \left( 1 - \frac{x}{X} \right)^{0.9} \left( \frac{f_k}{f_f} \right)^{0.5} \left( \frac{\mu_f}{\mu_g} \right)^{0.1} \]  

(22)

\[ R_{\text{ef}} = \frac{G(1-x)D}{\mu_f} \]  

(23)

Also the magnitudes of \( n, m, C, D \), which are measured with the least square method, are presented in Table 4.

“Fig. 8” shows the comparison of the heat transfer coefficient experimental data with measured values by this generalized correlation. According to this figure, this equation can predict the experimental data within a deviation of \( \pm 20\% \). The statistical comparison of this correlation and experimental data are brought in Table 5. This correlation can be used to other kinds of refrigerant/oil mixture, although it has been validated only by experimental data of the present study, and further verifications are required in order to ensure the accuracy of this correlation (Table 6).

5. Conclusion

Heat transfer coefficient and pressure drop coefficient of R600-a/oil mixture during convective boiling condition inside plain tube was experimentally investigated. The results indicate that heat transfer coefficient in low vapor qualities increases until it gets to the maximum value and then start to decrease at high vapor qualities. Unlike the heat transfer coefficient, the trend of pressure drop coefficient is ascending in all vapor qualities. By growing the mass velocity, the heat transfer coefficient and pressure drop coefficient of refrigerant oil mixture goes up for all oil concentration similar to pure refrigerant. In comparison to pure refrigerant, presence of oil increases the local heat transfer coefficient at low

![Figure 8. Comparison between the experimental heat transfer coefficient data and predicted values by generalized correlation.](image-url)

### Table 4

Prediction deviation of pressure drop correlations.

<table>
<thead>
<tr>
<th>Prediction method</th>
<th>( \gamma ) (%)</th>
<th>( \lambda ) (%)</th>
<th>( \epsilon ) (%)</th>
<th>( \delta ) (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>Muller-Steinhagen and Heck [27]</td>
<td>-23.134977</td>
<td>26.2020874</td>
<td>-25 to 15</td>
<td>54.6</td>
</tr>
<tr>
<td>Gronnerud [28]</td>
<td>13.1380085</td>
<td>32.5700787</td>
<td>-25 to 35</td>
<td>54</td>
</tr>
</tbody>
</table>

\( \delta \): the percentage of points predicted within an error bandwidth.

### Table 5

Constant values in the oil factor (\( EF_{\text{oil}} \)) equation.

<table>
<thead>
<tr>
<th>C</th>
<th>D</th>
<th>m</th>
<th>n</th>
</tr>
</thead>
<tbody>
<tr>
<td>5.19</td>
<td>1.49</td>
<td>1.22</td>
<td>0.12</td>
</tr>
</tbody>
</table>

### Table 6

Statistical comparison of generalized correlation and experimental data.

<table>
<thead>
<tr>
<th>( \gamma ) (%)</th>
<th>( \lambda ) (%)</th>
<th>( \epsilon ) (%)</th>
<th>( \delta ) (%)</th>
</tr>
</thead>
<tbody>
<tr>
<td>-0.56</td>
<td>9.04</td>
<td>95</td>
<td>10.88</td>
</tr>
</tbody>
</table>

\( \delta \): predicted within \( \pm 20\% \) \( \theta = \left\{ \frac{\sum_{i=1}^{N} \left( \frac{\text{value calculated} - \text{value experimental}}{\text{value experimental}} \right)^2}{0.5} \right\}^{0.5} \).
vapor qualities and decreases the heat transfer coefficient at high vapor qualities, but augments frictional pressure drop at all vapor qualities.

Existing correlations for convective boiling heat transfer and pressure drop of pure refrigerant by considering the thermophysical properties of the mixture were unsuccessful to predict well the present experimental data in the case of using modified properties due to the point that the presence of oil change the flow pattern and also these equations had not been derived for HC refrigerants. The generalized correlation for the heat transfer coefficient, can contribute to the design of heat transfer exchangers using hydrocarbon R-600a as the refrigerant for the refrigerator.

Acknowledgements

The authors would like to express their thanks to the Center of Excellence in Design and Optimization of Energy Systems, College of Engineering, University of Tehran for the financial supports through the set-up construction and research implementation and also the center of excellence in design and optimization of energy systems.

References