A NUMERICAL INVESTIGATION ON CONVECTIVE FILM BOILING OF THE SINGLE COMPONENT REFRIGERANT AS ALTERNATIVE OF CFCs IN A VERTICAL PIPE*

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Abstract-- In the present work, a numerical analysis has been done to calculate the heat and mass transfer during boiling based on the two-phase annular single component refrigerants in a vertical pipe. The conservation equations of mass, momentum, energy and turbulence models of k-ε for each phase have been solved through a numerical finite volume method. A specific wave model has been used for considering the effects of interfacial wavy instabilities in the transport phenomena. The numerical results of heat transfer for R–113 show a good agreement with the available experimental results reported by the others investigators. To compare the R-12 with its suggested alternative, R-134a, the numerical results have been achieved at the same boundary conditions for both of them. Results show that the behaviors of these refrigerants are similar. In addition, the heat transfer coefficient in the case of R-134a is greater than that of R-12, nearly 30 percent.

Keywords-- Convective boiling, refrigerants, gas–liquid annular flow, vertical pipe, turbulent flow

1. INTRODUCTION

The CFC refrigerants can, to a large extent, cause considerable damage to the ozone layer. Furthermore, they have a greater effect on the increase in global warming than carbon dioxide. During the search for alternative substances to replace them, many safe refrigerants have been introduced. These fall into two groups: Hydrofluoro Carbons (HFC) and Hydrochlorofluoro Carbon (HCFC). The latter is considered as a temporary alternative and should be eliminated until 2020 [1, 2].

The situation of refrigerants relating to the above groups is shown in Fig. 1, based on their ozone depleting potential (ODP) and global warming potential (GWP). From this figure, it is obvious that CFCs have a much more harmful effect on nature.

Extensive research has been initiated to replace acceptable alternatives with more appropriate refrigerants emphasized in the Montreal protocol 1987 [3]. Because of different problems encountered during the replacement, it is very important to study their heat and mass transfer. Using alternatives in refrigeration systems requires some parts of the equipment to be changed and involves new designs. Sometimes, for diminution, the irriversibilities of the cycles and the mixtures of the refrigerants with some of the alternatives according to the lorentz cycle are used [4, 5]. Implementation of these changes in accordance with the new cycles of thermal systems facilities, requires the hydrodynamics and heat transfer behaviors of the refrigerants during their phase change. By virtue of the extensive applications of R-12 as a common refrigerant in industry, its replacement has played a major role. The investigation of this refrigerant and its most remarkable alternative, R-134a, is emphasized here. In this study, the heat and mass transfer of a single–component refrigerant in the downward direction of a vertical pipe was considered. In this case, the convective film boiling of a two–phase vapor–liquid with an annular flow pattern was to be studied. The numerical simulation of Shibata et al. [6] is approximately similar to the present study . Nevertheless, the difference in this study is the turbulent flow case of the liquid film and the vapor phase which is handled.

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through appropriate turbulence modeling. For the reliability of the calculation, the numerical results of R-113 have been compared with their available experimental results obtained by the other authors [7, 8]. Moreover, the calculation was extended for R-12 and its alternative R-134a [9]. The results themselves have been compared in order to understand their thermodynamics and transport properties during the convective film boiling in a vertical pipe.

Fig.1. The values of ODP and GWP for refrigerants [1]

2. THEORETICAL INVESTIGATION

According to the rate of heat flux on the wall, the flow pattern of a two phase liquid-vapor during convective boiling could be quite different. In this study, only the annular case has been taken into account. In this kind of two phase flow, a thin liquid film is attached to the wall and its vapor is settled in the core of the pipe. According to the mass quality, the annular two-phase flow could appear in the range of 20 to 90 percent into the pipe [10]. In the literature for this kind of two-phase flow and for the heat transfer coefficient, the best known experimental correlation of Kandlikar is mostly used [11, 12]

\[
\frac{h_{lp}}{h_1} = C_1 C_o^{-2} + C_3 B_o^{-4} F_t l'
\]

With \( h_1 \), heat transfer coefficient of single phase flow from Dettius–Boelter as:

\[
h_1 = 0.023 \text{Re}^{0.8} \text{Pr}_I^{0.4} \left( \frac{\lambda_c}{D} \right)
\]

where \( B_o = \frac{q_w^0}{m^o \Delta h_v} \) boiling number, \( C_o = \left( \frac{1-x}{x} \right)^{0.8} \left( \frac{\rho_v}{\rho_l} \right)^{0.5} \), convective number, \( F_t \) is the correction factor of the refrigerants and \( C_1 \) to \( C_4 \) the constants which are obtained from experimental data [11]

Table 1.

<table>
<thead>
<tr>
<th>C_1</th>
<th>C_2</th>
<th>C_3</th>
<th>C_4</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.136</td>
<td>0.9</td>
<td>667.2</td>
<td>0.7</td>
</tr>
</tbody>
</table>

In the interface of liquid film with its vapor, the instabilities due to the effects of shear stress produce the ripple and disturbance waves. These waves have a considerable effect on the transport phenomena between phases. In the present study, based on Suzuki’s model [13], an apparent viscosity of the wave effects is introduced into the conservation equations. From the model, for wavy apparent viscosity we have

\[
\bar{\mu} = C \rho L^2 F
\]

where \( \bar{\mu} \) represents wavy viscosity, \( L \) wave amplitude, \( F \) wave frequency, \( \rho \) fluid density and \( C \), a regular constant. The data of \( F \) and \( L \) are obtained from the experimental measurements [14].
Two kinds of interfacial waves exist, so Eq. (3) for both phases has the following form:

$$\bar{\mu} = C_r \rho L_r^2 F_r + C_d \rho L_d^2 F_d$$

where \( r \) denotes ripple waves and \( d \) represents disturbance waves. In Table 2 the experimental results of Hagiwara et al. [14] for the value of parameters are shown. The value of \( C_r \) and \( C_d \) in Eq. (4) are regulator constants. They are chosen from the agreement between the numerical and experimental results.

Table 2. The experimental values of wave parameters [14]

<table>
<thead>
<tr>
<th>( G_\nu ) (gs(^{-1}))</th>
<th>( R_\nu )</th>
<th>( C_\eta ) (gs(^{-1}))</th>
<th>( R_\eta )</th>
<th>( L ) (mm)</th>
<th>( F ) (Hz)</th>
<th>Observed wave</th>
</tr>
</thead>
<tbody>
<tr>
<td>11.5</td>
<td>3.0 \times 10^4</td>
<td>3.33</td>
<td>1.4 \times 10^2</td>
<td>0.075</td>
<td>7.5 \times 10^1</td>
<td>Ripple</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6.67</td>
<td>2.8 \times 10^2</td>
<td>0.082</td>
<td>6.7 \times 10^1</td>
<td>Ripple</td>
</tr>
<tr>
<td></td>
<td></td>
<td>10.0</td>
<td>4.2 \times 10^2</td>
<td>0.60</td>
<td>4.4</td>
<td>D.W</td>
</tr>
<tr>
<td></td>
<td></td>
<td>11.7</td>
<td>5.0 \times 10^2</td>
<td>0.89</td>
<td>4.3 \times 10^1</td>
<td>D.W</td>
</tr>
<tr>
<td>15.5</td>
<td>4.0 \times 10^4</td>
<td>3.33</td>
<td>1.4 \times 10^2</td>
<td>0.73</td>
<td>7.9 \times 10^1</td>
<td>Ripple</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6.67</td>
<td>2.8 \times 10^2</td>
<td>0.066</td>
<td>6.9 \times 10^1</td>
<td>Ripple</td>
</tr>
<tr>
<td></td>
<td></td>
<td>10.0</td>
<td>4.2 \times 10^2</td>
<td>0.55</td>
<td>5.6</td>
<td>D.W</td>
</tr>
<tr>
<td></td>
<td></td>
<td>11.7</td>
<td>5.0 \times 10^2</td>
<td>0.81</td>
<td>5.4</td>
<td>D.W</td>
</tr>
<tr>
<td>21.1</td>
<td>5.5 \times 10^4</td>
<td>3.33</td>
<td>1.4 \times 10^2</td>
<td>0.070</td>
<td>8.2 \times 10^1</td>
<td>Ripple</td>
</tr>
<tr>
<td></td>
<td></td>
<td>6.67</td>
<td>2.8 \times 10^2</td>
<td>0.078</td>
<td>0.64 \times 10^1</td>
<td>Ripple</td>
</tr>
<tr>
<td></td>
<td></td>
<td>10.0</td>
<td>4.2 \times 10^2</td>
<td>0.43</td>
<td>7.0</td>
<td>D.W</td>
</tr>
<tr>
<td></td>
<td></td>
<td>11.7</td>
<td>5.0 \times 10^2</td>
<td>0.68</td>
<td>7.1</td>
<td>D.W</td>
</tr>
</tbody>
</table>

a) Governing equations

For the basic equations of the phases, the following assumptions are required:

1. The flow of vapor is turbulent, but the liquid film is considered laminar or turbulent.
2. In the interface, the fluid is always in the saturation temperature,
3. The flow in all lengths of the pipe is in an annular pattern,
4. Phase change only occurs on the interface,
5. Uniform heat flux is assumed on the wall,
6. Entrainment from the liquid film is ignored.

Figure 2 shows a schematic diagram of the two–phase flow with its boundary conditions.
I. Equations for liquid film \((r_1 < r < r_0)\)

\[
\frac{\partial \bar{u}_l}{\partial x} + \frac{1}{r} \frac{\partial (r \bar{v}_l)}{\partial r} = 0
\]  

(5)

\[
\rho_l \frac{\partial \bar{u}_l}{\partial x} + \rho_l \bar{v}_l \frac{\partial \bar{u}_l}{\partial r} = -\frac{\partial P_l}{\partial x} + \frac{1}{r} \frac{\partial}{\partial r} \left[ r(\mu_l + \bar{\mu}_l + \mu_t) \frac{\partial \bar{u}_l}{\partial r} \right] + \rho_l g
\]  

(6)

\[
\rho_l C_{p_l} \frac{\partial T_l}{\partial x} + \rho_l C_{p_l} \bar{v}_l \frac{\partial T_l}{\partial r} = \frac{1}{r} \frac{\partial}{\partial r} \left[ \left( \mu_l C_{p_l} + \bar{\mu}_l C_{p_l} + \mu_t C_{p_l} \right) \frac{\partial T_l}{\partial r} \right]
\]  

(7)

II. Equations for vapor phase \((0 < r < r_1)\)

\[
\frac{\partial}{\partial x} \left( \rho_g u_g \right) + \frac{1}{r} \frac{\partial}{\partial r} \left( r \rho_g v_g \right) = 0
\]  

(8)

\[
\rho_g \frac{\partial u_g}{\partial x} + \rho_g \frac{\partial v_g}{\partial r} = -\frac{\partial P_g}{\partial x} + \frac{1}{r} \frac{\partial}{\partial r} \left[ r(\mu_g + \bar{\mu}_g + \mu_g) \frac{\partial u_g}{\partial r} \right] + \rho_g g
\]  

(9)

\[
\rho_g C_{p_g} \frac{\partial T_g}{\partial x} + \rho_g C_{p_g} \frac{\partial v_g T_g}{\partial r} = \frac{1}{r} \frac{\partial}{\partial r} \left[ \left( \mu_g C_{p_g} + \bar{\mu}_g C_{p_g} \right) + \mu_g C_{p_g} \right] \frac{\partial T_g}{\partial r} \right]
\]  

(10)

where \(\text{Pr}_t\) is a turbulent Prandtl number (\(\text{Pr}_t = 0.9\) for fluids with \(\text{Pr} > 0.7\) [17]). For the vapor, flow is always turbulent, but for the liquid film the flow may be laminar or turbulent. For the turbulent flow, the standard \(k-\varepsilon\) model considering momentum exchange between each phase proposed by Jones and Launder was used [15, 16]. The turbulence kinetic energy, \(k\), and its dissipation rate, \(\varepsilon\), by introducing

\[
Dk = \frac{1}{r} \frac{\partial}{\partial r} \left( \sigma_k \frac{\partial k}{\partial r} \right) + \nu_k \left( \frac{\partial u}{\partial y} \right)^2 + \beta_k \frac{\nu_k}{\sigma_k} \frac{\partial k}{\partial x} - \varepsilon
\]  

(11)

\[
D\varepsilon = \frac{1}{r} \frac{\partial}{\partial r} \left( \sigma_\varepsilon \frac{\partial \varepsilon}{\partial r} \right) + C_1 \nu_k \left( \frac{\partial u}{\partial y} \right)^2 + \frac{\beta_k \nu_k}{\sigma_k} \frac{\partial \varepsilon}{\partial x} - C_2 \frac{\varepsilon^2}{k}
\]  

(12)

Table 4. The values of the constants [15, 16]

<table>
<thead>
<tr>
<th>(C_1)</th>
<th>(C_2)</th>
<th>(C_\mu)</th>
<th>(\sigma)</th>
<th>(\sigma_k)</th>
<th>(\sigma_\varepsilon)</th>
<th>(\alpha_k)</th>
</tr>
</thead>
<tbody>
<tr>
<td>1.44</td>
<td>1.92</td>
<td>0.09</td>
<td>0.7</td>
<td>1</td>
<td>0.9</td>
<td>1.3</td>
</tr>
</tbody>
</table>

The standard values of the constants and the values of Prandtl number, \(\sigma\), and the turbulence Prandtl number, \(\sigma\), are reported from the references [15, 16].

In the liquid film, for an accurate calculation of the turbulent viscosity, the algebraic model of Reynolds stresses developed by Rodi [17, 18] was used in the region closed to the interface.

\[
-\bar{u'} \bar{v'} = \frac{2}{3} \frac{1 - C_\gamma}{C_1} \frac{C_1 - 1 + \frac{P}{\bar{\varepsilon} k^2} \frac{\partial u}{\partial y}}{(C_1 - 1 + \frac{P}{\bar{\varepsilon}})^2 \frac{\partial \varepsilon}{\partial y}} \frac{\partial u}{\partial y}
\]  

(13)

The values of turbulent stresses are related to the turbulent viscosity from the Boussinesq theory as:

\[
\tau_i = -\rho \bar{u'} \bar{v'} = \mu_t \left( \frac{\partial \bar{u}}{\partial y} \right)
\]  

(14)

P is the rate of production of the turbulent kinetic energy.
\[
\rho = \frac{v_i}{\eta^2}
\] (15)

From the comparison of the above equations with (13), it is seen that the coefficient of \( C_\mu \), this time is not constant and varies as \( C_\mu = C_\mu(p/\varepsilon) \). The constant values of (12) are recommended as: \( C_1 = 1.5 \) and \( C_\gamma = 0.6 \) [15, 16].

**III. Boundary conditions**

If in the calculation a control volume contains both liquid and vapor, the properties and quantities of the dependant variables could not be considered monotonous because in this case a great error through calculation occurs. To avoid the error, each of the phases has been separately exposed. In other words, the discretizing equations and the dependant variables were independently considered for the phases. In addition to the inlet, outlet and wall boundary conditions, it is required to introduce the conditions on the interface. But from the behavior of the transport phenomena in this problem, we are motivated to ignore the outlet boundary conditions.

**a. Inlet conditions**

- Uniform velocities in the phases which were calculated from the flow rates and mass qualities
- Uniform temperatures in the phases by means of their saturated values related to the pressure of the inlet section
- The values of \( \kappa \) and \( \varepsilon \) are not easy to impose accurately. In this work the values used by Baker et al. [19] for incompressible air flow with high Reynolds numbers have been adopted for the vapor as follows:

\[
\kappa(x_o, y) = C_\omega^{2(2-a)} L^2 \left( \frac{\partial u}{\partial y} \right)
\] (16)

\[
\varepsilon(x_o, y) = C_\gamma^{0.5} \omega^{b-d} C_\varepsilon[k(x_o, y)]^{3/2}
\] (17)

\( L \) is a mixing length scale and its values for the inner and outer layers are: \( L = ky \), \( L = 0.09\delta \). The thickness of the inner layer is \( y_{\text{max}} = \frac{0.09\delta}{k} \). \( \omega \) is Van-Driest damping factor and \( k \) is Von-Karman constant. The other constants of (16) and (17) are as:

| Table. 5. Constant values of (16) and (17) [19] |
|---|---|---|---|
| a | b | \( C \) | \( C_\varepsilon \) |
| 1.0 | 2.0 | 3.0 | 0.068 |

For the liquid film from the same reference [19]

\[
K(x_0, y) = K^*(x_0, y) + \frac{\nu}{\sqrt{C_D}} \left( \frac{\partial u}{\partial y} \right)
\] (18)

\[
\varepsilon(x_0, y) = \varepsilon^*(x_0, y) + \nu \left( \frac{\partial u}{\partial y} \right)^2
\] (19)

\[
\kappa^*(x_o, y) = \eta L_{in}^2 \quad \text{and} \quad \varepsilon^*(x_o, y) = \frac{2[k^*(x_o, y)]^{3/2}}{L_{in} D}
\] (20)

where, \( \eta \) represents a turbulence intensity and \( L_{in} \) is a length scale factor. According to [19], \( \eta = 0.03 \) and \( L_{in} = 0.005 \).

**b. Wall conditions (r=\( r_0 \))**

\( r = r_0 : \ u_i = 0, \ v_i = 0, \ -\lambda_1 \frac{\partial T_i}{\partial r} = q^* = Cte \). On the wall the values of \( k = 0 \). But for the \( \varepsilon \) it is not exactly the same. Therefore instead of the wall, the values of \( k \) and \( \varepsilon \) may be obtained for the nearest point of the wall (P point in Fig. 3) as follows:
2/1

\[ K_p = \frac{v^*}{\mu}^{1/2} \quad \text{and} \quad \varepsilon_p = \frac{v^*}{k_p^{3/2}} \]  

(21)

where \( v^* = \sqrt{\frac{\tau_0}{\rho}} \) is the wall friction velocity.

![Fig. 3. The nearest point of the calculation to the boundary (P point)](image)

c. Interface conditions (\( r=r_i \))

\[ r = r_i : u_{li} = u_{gi} = u_i, \quad \frac{dp_l}{dx} = \frac{dp_g}{dx}, \quad p_l = p_g, \ \tau_{li} = \tau_{gi} = \tau_i, \quad T_l = T_g = T_i, \quad -\lambda_l \partial T_l / \partial r + I_{bg} m^* = -\lambda_g \partial T_g / \partial r \]  

(22)

For the \( k \) and \( \varepsilon \) in the vicinity of the interface in the liquid film, by considering the rate balance of dissipation of turbulence kinetic energies with their convective and diffusive terms, we have

\[ K_{p'} = \left[ \frac{m^0}{\rho(C^*_{\mu} / \sigma_k - C_D)} \right]^{1/2} \quad \text{and} \quad \varepsilon_{p'} = C_D \frac{k_{p'}^{3/2}}{L} \]  

(23)

with mean thickness \( C^*_{\mu} = \frac{C_{\mu}}{C_D} \), \( L = 0.005 \delta \), and \( p' \) is the nearest point to the interface in the liquid film with mean thickness \( \delta \). For the vapor phase, \( k \) and \( \varepsilon \) are introduced from Eq. (21), but in this case the friction velocity is obtained from the interfacial shear stresses as \( v^* = \sqrt{\frac{\tau_i}{\rho_g}} \).

d. Axis conditions (\( r=0 \))

For \( r = 0 \):

\[ v_g = 0, \quad \frac{\partial u_g}{\partial r} = 0, \quad \frac{\partial k}{\partial r} = 0, \quad \frac{\partial \varepsilon}{\partial r} = 0, \quad \frac{\partial T_g}{\partial r} = 0 \]  

(24)

3. CALCULATION PROCEDURE

As an analytical solution to the governing equations is not possible, the numerical procedure based on the control volume method was used [20]. The numerical code of calculation named GENMIX developed by Spalding [21] for a general computer program for the single phase flow of two dimensional parabolic phenomena has been taken into consideration after some improvements to the present study. The improved version, called GENMIX-2P, has already been used in the previous work of the authors [22].

By transforming the cylindrical coordinates to the \((x-\omega)\) one, the general form of the main equation becomes

\[ \frac{\partial \phi}{\partial x} + (a + b\omega) \frac{\partial \phi}{\partial \omega} = \frac{\partial}{\partial \omega} [C_{\phi} \frac{\partial \phi}{\partial \omega}] + d \]  

(25)

with

\[ \omega = \frac{\psi - \psi_i}{\psi_{\varepsilon} - \psi_i} \]
wherein $\psi$ denotes stream function, $\phi$ represents any dependant variables ($u, v, h^2, k, \varepsilon, \ldots$), constants a to d depend upon the variations of mass flow rates from the boundaries (E and I boundaries) and the properties of the dependant variables $\phi$ [21]. First, in this method, Eq. (25) is integrated in the control volume around any grid point for the achievement of the discretizing equations, and then these algebraic equations are solved.

The numerical procedure of the calculation is shown in the flowchart of Fig. 4. As it is seen, at first the values of the evaporative rate of liquid $m^*$, film thickness $\delta$, and some flow properties at interface as $u_i$ are unknown. Hence, it is necessary to guess $m^*$, $\delta$ and the flow properties at the interface. Then, the main equation was solved. From the boundary conditions on the interface, the values of pressures, velocities, temperatures and $k, \varepsilon$ of the turbulence for each step of the iteration are carried out until the matching conditions are met.

![Flowchart of the calculation procedure](image)

The calculation has been taken from the selection of grid numbers in liquid film and gas phase. In this study, 20 grids in liquid film and 40 grids in vapor phase have also been chosen (e.g N20NG40). For the sensitivity analysis of the calculation, the cases of N15NG30 and N25NG50 have also been introduced. As it is shown in Fig. 5, the independency for liquid film profile at x/d=70 is obviously approved.

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4. RESULTS AND DISCUSSION

At first, due to the experimental data of interfacial waves existing in the literature for two phase annular air-water flow in pipes, the numerical calculation of the convective film boiling of air-water has been conducted. Two different interfacial boundary conditions both with and without instability waves have been introduced in the calculations. The inlet data for this numerical analysis was as follows:

\[ d=26.4 \text{ mm} , \quad q_w =9.63 \text{ kW/m}^2 , \quad T_i =27.4 \text{ C} , \quad G_g =11.5 \text{ Kg/m}^2s, \quad G_l =6.67 \text{ Kg/m}^2s \]

A few of the results of this calculation are presented here. In Figs. (6a, 6b) and (7a, 7b) the velocity and temperature profile of liquid film and gas at \( X/d=35 \) from the inlet of pipe are shown.

In Fig.8 the distribution of wall and gas bulk temperature along the pipe are presented. In Fig. 9 the variation of the heat transfer coefficient with the pipe length has been shown. The behavior of this variation is completely in accordance with the data existing in the previous works [8, 11, 12].
To estimate the validity and reliability of the numerical results for single component refrigerants, the experimental values of Jensen & Bensler [7] have been used. In their work, the saturated forced convection boiling of R–113 in a vertical pipe has been investigated. The following conditions were conducted:

d= 8.1 mm, P= 560.03 kPa, G= 634.91 Kg/m²s, X= 0.304, \( q_w = 34.71 kW/m² \).

In Fig. 10, the numerical results of the present work were compared with Jensen’s data and the Kandlikar correlation for the heat transfer coefficient, \( h_p \). The result is fairly competent, except in the small entrance region of the pipe. This discrepancy may be dependent on the case in which in the thick liquid film, nucleate and convective film boiling simultaneously exist in the inlet of the pipe. But in the numerical analysis, the heat is transferred by conduction and convection through a thin liquid film and it is proposed that the evaporation has taken place at the interface.
In Fig. 11, the forced convection boiling of R-11 from this work is compared with the experimental data of Fujita et al. [23], the empirical correlation of Kandilikar [12] and the numerical results of Bennet and Chen [8]. Our study is also compared with the numerical analysis of Shibata et al. [6], in which the turbulence effects have been simply modeled in vapor phase with the mixing length theory. In this figure, of \( h_{ip}/h_{lo} \) versus the Lockhart-Martinelli parameter \( 1/X_u \) are plotted. Where \( h_{ip} \) denotes heat transfer coefficient of two phase flow and \( h_{lo} \) indicates the heat transfer coefficient for the liquid phase in the pipe obtained by the Dettius-Bulter equation. The data of Bennet and Chen, and Shibata are for the case of upward flow, whereas Kandilikar’s correlation extends for both up and down directions of flow.

Besides the importance of the numerical procedure, the main object of this study was the investigation of R-134a heat and mass transfer as an alternative of R-12 during evaporation. The experimental data of Lavin and Young [24] for R-12 have been used for comparison, therefore the inlet conditions for both of R-12 and R-134a were taken as follow:

\[
d = 10 \text{ mm}, \quad P = 705.3 \text{ Kpa}, \quad G = 450 \text{ Kg/m}^2\text{s}, \quad X = 0.30, \quad \dot{q}_{w} = 30 \text{Kw/m}^2
\]

In Figs. 12 and 13 the velocity distribution of R-134a during the convective boiling in the three cross sections of the pipe were shown for liquid film and vapor core. Because of the liquid boiling at the interface, the velocity on the liquid film decreases with the pipe length, while in the vapor phase the opposite is proven. The effects of instabilities on the momentum transfer for both phases can be observed.

As is shown in Fig. 13, the velocity distribution in vapor near the interface has some inflections, and its effect on the entrance region of the pipe is high, while it’s effect along the pipe decreases by the length. In the region just near the interface there is wave induced motion. The frequency and length scale of ripple and disturbance waves strongly depends on liquid film Reynolds number. The distribution of Reynolds number for vapor and liquid film versus pipe length are presented in Figs. 14 and 15. As is illustrated, in the entrance region of the pipe, the liquid film Reynolds number is high (greater than 15000), therefore as is seen in Eq. (5), the influence of waves on effective viscosity is also high. From the experimental data of Sekoguchi et al. [25, 26], and by means of the least square polynomial approximation, the following correlations obtained for the frequency and length scale of ripple and disturbance waves have been used in our calculations.
In the liquid film, the wave-induced motion is considered to be suppressed near the wall by the viscous effect. So, it is supposed that the wave effect is decreased in the region of $0 \leq y^+ < 5$, and disappeared between $0 \leq y^+ < 5$. Here $y^+$ denotes the nondimensional distance from the wall in wall units. Thus the distribution of the effective viscosity in the liquid film, $\mu_{eff}$, was given as follows:

$$0 \leq y^+ < 5 \quad \mu_{eff} = \mu_l$$
$$5 \leq y^+ < 30 \quad \mu_{eff} = \mu_l + \bar{\mu}_l (y^+ - 5)/25$$
$$30 \leq y^+ < \delta^+ \quad \mu_{eff} = \mu_l + \bar{\mu}_l + \mu_l$$

(29)

where

$$y^+ = \frac{u^+ y}{v_l}, \quad \delta^+ = \frac{u^+ \delta}{v_l}, \quad u^+ = \sqrt{\frac{\tau_w}{\rho_l}}$$

In the vapor phase, it is considered that the interfacial waves intermittently break the viscous sublayer, otherwise persisting in the vapor phase layer near the interface. Such a wave effect prevailed in a layer close to the interface, but the turbulent transport should become more significant than other sources of viscosity in the region far from the interface. Therefore, the effective viscosity in the vapor phase was described as follows:

$$\bar{\mu}_g \geq \mu_{gr} \quad \mu_{eff} = \mu_g + \bar{\mu}_g$$
$$\bar{\mu}_g \leq \mu_{gr} \quad \mu_{eff} = \mu_g + \mu_{gr}$$

Here the distribution of the turbulent viscosity in the vapor phase, $\mu_{gr}$, is calculated by the standard $k-\varepsilon$ model [14,15].

In Figs. 16 and 17, the distribution of the interfacial velocity and shear stress along the pipe are shown. As is seen, the velocity during the permanent boiling of the liquid from the interface is reduced and causes an increase in the shear stress along the pipe. The profile of the turbulent kinetic energy and the turbulent viscous dissipation in the vapor phase are shown in Figs. 17 and 19. The behaviour of $k$ and $\varepsilon$ in this study is similar with the result obtained by Mansour et al. [27] for the case of a single phase flow near the wall. In the case of turbulent liquid film flow, the distribution of $k$ is also in Fig. 20, whereas the great values of $k$ near the interface and the wall indicate the similar treatment as [27].

In Fig. 21, variation of pressure along the pipe is shown. As is seen, the pressure decreases in the direction of the flow. Nevertheless, the pressure drop is not intensive, therefore we can consider the physical and thermodynamical properties of the single component refrigerants as constants for these ranges of pressure variation during evaporation.
In Figs. 22 and 23, the mass flow rate of evaporation, and the liquid film thickness along the pipe are represented for R-12 and R-134a. It is seen from these figures that the thickness of the liquid film is obviously decreased, but the evaporative mass flow rate becomes approximately constant along the pipe.
In Fig. 24, the quality distributions along the pipe presented. As is seen, the length of the evaporator for R134a is greater than that of R12. Hence, the liquid film thickness in the case of R12 is always small because of its small values of vapor latent heat compared with values of R134a.

The result of constant evaporation mass flow rate may be interpreted by establishing the equilibrium between the liquid and its vapor in the pipe. From Fig. 24, we can also compare the results of R12 and R134a. It is clearly proved that for the same values of $G$ and $q_w^{\theta}$, the evaporation in the case of R12 is quicker than that of R134a. This result is simply related to the vapor latent heat of the fluids which is greater in the R134a than R12.

In Fig. 25, the variation of the heat transfer coefficient with vapor quality for R12 and R134a is presented. From this figure, it is seen that the quality varies between 0.3 and 0.8 along the pipe. The heat transfer coefficient for R134a is always greater than the heat transfer coefficient for R12 and the difference between them along the pipe remains 25 to 35 percent. The experimental study of Eckels and Pate [28] for the case of evaporation and condensation of the same fluids into the pipes shows a slightly greater difference between them than this study.

Finally, it is found from that the behavior of R134a is similar to R12 from the view point of the transport phenomena during convective film boiling. Hence for the greater part of the thermal equipment regarding the ODP inactivation of R12, it can be easily alternated by R134a, in spite of its small rate of evaporation compared with R12.
5. CONCLUSIONS

In this study, by introducing the Suzuki wave model, the standard k-ε and the Reynolds stress model of turbulence in both phases, the convective film boiling of the refrigerants in a vertical pipe were investigated. The numerical results compared with the available experimental results come into agreement. From these results, it has been proven that R134a has thermal properties very close to R12. The heat transfer coefficient of R134a is greater than that of R12, so it should be replaced as a proper alternative in thermal equipment.

NOMENCLATURE

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
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<tbody>
<tr>
<td>B₀</td>
<td>boiling number</td>
</tr>
<tr>
<td>Cₚ</td>
<td>specific heat (J/kg °C)</td>
</tr>
<tr>
<td>Cₒ</td>
<td>convective number</td>
</tr>
<tr>
<td>Cₜ</td>
<td>coefficient of turbulent viscosity</td>
</tr>
<tr>
<td>D</td>
<td>tube inner diameter</td>
</tr>
<tr>
<td>F</td>
<td>wave frequency (Hrtz)</td>
</tr>
<tr>
<td>F₀</td>
<td>refrigerants corrective factor</td>
</tr>
<tr>
<td>G</td>
<td>mass flux (kg/s/m²)</td>
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<td>g</td>
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<td>h</td>
<td>heat transfer coefficient (w/m²²)</td>
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<td>stagnation enthalpy (J/kg)</td>
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<td>Iₑg</td>
<td>Latent heat of evaporation (w)</td>
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<td>length scale of wave</td>
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<tr>
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<td>liquid film mass flow rate (kg/s)</td>
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<tr>
<td>P</td>
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<td>Pᵣ</td>
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<tr>
<td>q</td>
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<td>Vₒ</td>
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<tr>
<td>x</td>
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<td>X</td>
<td>vapor quality</td>
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Greek letters

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<tr>
<td>δ</td>
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<tr>
<td>ε</td>
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<td>Φ</td>
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<tr>
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<td>ηₑ</td>
<td>Prandtl number for ε</td>
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<tr>
<td>τ</td>
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Subscripts

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<tr>
<td>g</td>
<td>gas, vapor</td>
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Superscript

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REFERENCES