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ABSTRACT

A numerical study based on two-fluid model describes the thermal and fluid-dynamic behaviour of the two-phase flow inside ducts -condensers and evaporators-. The discretization of the governing equations has been developed by means of the finite volume technique using a staggered mesh. A semi-implicit pressure based method SIMPLE-like is used to solve the pressure distribution, the velocity field, and the temperature distribution of each phase. Different empirical correlations have been used to evaluate the mass, momentum and energy exchanged through the interface, geometric conditions, gas and liquid distribution into the tube and other terms that appear in the conservation equations. These correlations are known depending on the flow pattern map, which is function of the gas volume fraction and velocities. A comparison between the numerical simulation data obtained with quasi-homogeneous model and two-fluid model in contrast with experimental data are illustrated.

1. INTRODUCTION

In different industrial applications the two-phase flow phenomena is very common, e.g. gas-liquid, solid-liquid or solid-gas are present along the manufacturing process. Furthermore, different fluids can be working at the same time, e.g., oil-water, carbon particles-steam, etc. The high number of applications where the two-phase flow phenomena is present has increased the motivation to study the multiphase flow in multi-component fluids along the last thirty years.

This paper is focused on detailed one-dimensional numerical simulation of phase change phenomena gas-liquid flow into pipes, assuming a only one fluid. The basic field equation consists of two continuity equations, two momentum equations and two energy equations, one equation for each phase. The empirical information used in the nuclear applications has been extrapolated and applied to our case, with the aim of defining some important parameters such as geometric conditions, heat transfer coefficients and shear stresses at the interface and with the wall. All of these values depend on the flow pattern map, which is defined as function of the void fraction and velocities of the flow.

The six governing equations have been organized in a set of algebraic equations that is solved using the three diagonal matrix algorithm TDMA. The coupling between momentum and continuity equations is solve with a semi-implicit pressure based method SIMPLEC (Patankar, 1980). This method is very usual in the single-phase flow finite volume simulation technique. Some difficulties should be take into account in two-phase flow due to two momentum and two continuity equations are coupling together with the effect of the momentum and mass exchanged.

Different numerical aspects have been evaluated with the aim of verifying the quality of the numerical solution. The difference between the numerical simulation results obtained with quasi-homogeneous (Rigola et al., 2004) model and two-fluid model into the tube are illustrated. Finally, a comparison between numeri-
2. MATHEMATICAL FORMULATION

Mathematical formulation is based on the application of the conservative equations on each phase, liquid and gas. Then, six-equations are considered: two continuity, two momentum and two energy equations for each one of the phases (Ishii and Hibiki, 2006). The assumed hypotheses are: one-dimensional flow, constant cross section, and negligible axial heat conduction in fluid, and heat radiation. The governing equations for the two phases are:

\[
\frac{\partial}{\partial t}[\rho_l(1-\alpha)] + \frac{1}{S} \frac{\partial}{\partial z}[(\rho_l(1-\alpha)v_l S)] = -\Gamma 
\]

\[
\frac{\partial}{\partial t}[\rho_g \alpha_l] + \frac{1}{S} \frac{\partial}{\partial z}[\rho_g \alpha_l v_g S] = \Gamma 
\]

\[
\frac{\partial}{\partial t}[(\rho_l(1-\alpha)v_l S)] + \frac{1}{S} \frac{\partial}{\partial z}[(\rho_l(1-\alpha)v_l S)] = -(1-\alpha) \frac{\partial p}{\partial z} - g \rho_l (1-\alpha) \sin \theta - \frac{P_{wl} \tau_{wl}}{S} - \frac{P_{l} \tau_{l}}{S} + \Gamma \tau_{il} + C \alpha \rho S (1-\alpha) \frac{\partial (\bar{v}_l - \bar{v}_i)}{\partial t} 
\]

\[
\frac{\partial}{\partial t}[(\rho_l(1-\alpha)\bar{H}_l)] + \frac{\partial}{\partial t}[(\rho_l(1-\alpha)\bar{v}_l^2/2 - \rho_l(1-\alpha) g \bar{v}_l \sin \theta)] + \frac{q_{il} P_{il}}{S} + \frac{q_{wl} P_{wl}}{S} - \Gamma \left[ \bar{H}_il + \frac{\bar{v}_i^2}{2} - g \bar{v}_l \sin \theta \right] + (1-\alpha) \frac{\partial p}{\partial t} - \xi \frac{P_{l}}{S} \tau_{il} 
\]

\[
\frac{\partial}{\partial t}[(\rho_g \alpha \bar{H}_g)] + \frac{\partial}{\partial t}[(\rho_g \alpha \bar{v}_g^2/2 - \rho_g \alpha g \bar{v}_g \sin \theta)] + \frac{q_{lg} P_{lg}}{S} + \frac{q_{wg} P_{wg}}{S} + \Gamma \left[ \bar{H}_ig + \frac{\bar{v}_i^2}{2} - g \bar{v}_g \sin \theta \right] + \alpha \frac{\partial p}{\partial t} + \xi \frac{P_{g}}{S} \tau_{ig} 
\]

In equations (1) to (6), \( \alpha \) is the void fraction of gas, the void fraction of liquid is expressed as \((1 - \alpha_g)\), and the mass transfer rate per unit volume is labelled as \( \Gamma \). Applying a mass balance, the mass transfer from liquid to gas should be equal to mass lost by liquid. In momentum equations \( \tau_{wg} \) and \( \tau_{wl} \) are the shear stresses acting on the phase at the wall and \( \tau_i \) at the interface, \( \theta \) is the inclination angle of the pipe and \( C \) is the virtual mass coefficient. The virtual mass coefficient depends on regimen present in the flow (\( C=0 \) in stratified regimen flow). The wall perimeter wetted by liquid and gas are denoted as \( P_{wl} \) and \( P_{wg} \) and the cross section area is \( S \). Finally, in the energy equations \( q_{il} \) and \( q_{lg} \) are the heat transfered exchanged at the interface, while \( q_{wl} \) and \( q_{wg} \) are the heat transfered from the wall to the liquid and gas phase, respectively.


2.1 Discretization

The governing equations (1) to (6) can be integrated in terms of the local averaged fluid variables using the finite control volume technique. The concept of a staggered mesh has been used, therefore velocities are defined at the cell boundaries, while pressure, void fraction and enthalpy are located at the middle of the main volume. Figure 1 depicts a control volumes representation.

Before the discretization of the governing equations, it is necessary to define two previous aspects that are important in the equations development. Firstly, the mass flow rate per unit of area is defined as: 

\[ G_{k,c} = \rho_{k,c} \bar{v}_{k,c} \bar{v}_{k,c} \] where \( k \) means the gas or liquid phases. Secondly, the void fraction is defined as the relation between the cross area occupied by the gas and the total cross area, \( \alpha_g = S_g/S \). Based on the parameters defined above, the momentum of the gas and liquid are discretized on the staggered mesh and can be expressed as:

\[
\frac{1}{\Delta t} \left[ \rho^o g,p \bar{v}_g,p \bar{v}_g,p \right] + \frac{1}{\Delta t} \left[ \rho^o g,E \bar{v}_g,E \bar{v}_g,E \right] \] 

\[\begin{align*}
\frac{1}{\Delta t} & \left[ \rho^o g,p \bar{v}_g,p \bar{v}_g,p \right] v_{g,p} + \frac{1}{\Delta t} \left[ \rho^o g,E \bar{v}_g,E \bar{v}_g,E \right] v_{g,E} + \\
& \frac{1}{\Delta z} \left[ -\max \left( \frac{G_{g,c} + G_{g,f}}{2}, 0 \right) v_{g,E} + \max \left( \frac{G_{g,c} + G_{g,f}}{2}, 0 \right) v_{g,p} \right] - \\
& \frac{1}{\Delta z} \left[ \max \left( \frac{G_{g,c} + G_{g,w}}{2}, 0 \right) v_{g,W} - \max \left( \frac{G_{g,c} + G_{g,w}}{2}, 0 \right) v_{g,p} \right] = \\
& -\frac{1}{\Delta z} \left[ (\alpha + \alpha_E) \left[ v_{E} - p_{E} \right] - g \left( \rho_{g,p} \bar{v}_g,p + \rho_{g,E} \bar{v}_g,E \right) \sin \theta \right] - \frac{P_{w} \tau_{w} \bar{v}_w}{S} - \frac{P_{i} \tau_{i}}{S} + \Gamma \left[ v_{i,p} + v_{g,p} \right] \tag{7}
\end{align*}\]

\[
\frac{1}{\Delta t} \left[ \rho^o l,p \left( 1 - \bar{v} \right) + \rho^o l,E \left( 1 - \bar{v} \right) \right] v_{l,p} + \frac{1}{\Delta t} \left[ \rho^o l,p \left( 1 - \bar{v} \right) + \rho^o l,E \left( 1 - \bar{v} \right) \right] v_{l,E} + \\
\frac{1}{\Delta z} \left[ -\max \left( \frac{G_{l,c} + G_{l,f}}{2}, 0 \right) v_{l,E} + \max \left( \frac{G_{l,c} + G_{l,f}}{2}, 0 \right) v_{l,p} \right] - \\
\frac{1}{\Delta z} \left[ \max \left( \frac{G_{l,c} + G_{l,w}}{2}, 0 \right) v_{l,W} - \max \left( \frac{G_{l,c} + G_{l,w}}{2}, 0 \right) v_{l,p} \right] = \\
-\frac{1}{\Delta z} \left[ (1 - \alpha) \left[ v_{E} - p_{E} \right] - g \left( \rho_{l,p} \left( 1 - \alpha \right) + \rho_{l,E} \left( 1 - \alpha \right) \right) \sin \theta \right] - \frac{P_{w} \tau_{w} \bar{v}_w}{S} - \frac{P_{i} \tau_{i}}{S} + \Gamma \left[ v_{i,p} - v_{l,p} \right] \tag{8}
\]

Where \( v_{il} \) and \( v_{ig} \) are the interface velocities, the values of these parameters are assumed to be equals, then, \( v_{il} = v_{ig} = \lambda v_{g} + (1 - \lambda) v_{l} \), where \( \lambda \) is 0 or 1 depending of the mass flux value exchanged at the interface \( \Gamma \) (Idaho National Engineering, 2001).

The pressure correction equation is defined from gas and liquid continuity equations. The sum of both equations is discretized on the main mesh. Next relations help to develop the pressure correction equation:
\[ G_{g,e} = (\rho_{g,e} \bar{\sigma}_{g,e}) v_{g,e} = (\rho_{g,e} \bar{\sigma}_{g,e})^* v_{g,e}^{*} + (\rho_{g,e} \bar{\sigma}_{g,e})' v_{g,e}' + (\rho_{g,e} \bar{\sigma}_{g,e})'' v_{g,e}'' \]

\[ v_{g,e}' = v_{g,p}^{*} = d_{e} v_{e}^{*} p_{e}^{*} (p_{p}^{*} - p_{E}^{*}) \]

\[ (\rho_{g,e} \bar{\sigma}_{g,e})' = \frac{1}{\rho_{g,e} \bar{\sigma}_{g,e}} \frac{\Delta p_{e}^{*}}{\Delta t} = \frac{\rho_{g,e} \bar{\sigma}_{g,e}}{\gamma p_{e}^{*}} \frac{\Delta p_{e}^{*}}{\Delta t} = \frac{\rho_{g,e} \bar{\sigma}_{g,e}}{\gamma} \left[ F_{p}^{*} \frac{p_{p}^{*}}{p_{p}^{*}} + (1 - F_{p}^{*}) \frac{p_{E}^{*}}{p_{E}^{*}} \right] \]

Where the superscript * indicate the guess value of the variable \( \phi \) and the superscript ' means the correction of the variable \( \phi \), assuming \( \phi \) as the variable value, e.g. velocity, density or pressure. Using the above expressions, the pressure correction equation can be written in the following form:

\[
\begin{align*}
\Delta z \frac{\rho_{g,e}^{*} p_{p}^{*}}{\Delta t} + \frac{\Delta z \rho_{g,e}^{*} \bar{\sigma}_{e}^{*} p_{p}^{*}}{\Delta t} - \Delta z \frac{\rho_{g,e}^{*} \bar{\sigma}_{e}^{*} p_{p}^{*}}{\Delta t} + \\
\Delta z \frac{\rho_{l,e}^{*} (1 - \bar{\sigma}_{e}^{*}) p_{p}^{*}}{\Delta t} + \frac{\Delta z \rho_{l,e}^{*} (1 - \bar{\sigma}_{e}^{*}) p_{p}^{*} p_{p}^{*} - \Delta z \rho_{l,e}^{*} (1 - \bar{\sigma}_{e}^{*}) p_{p}^{*}}{\Delta t} + \\
G_{g,e} + \max(G_{g,e}, 0) \frac{1}{\gamma_{pp}^{*}} p_{p}^{*} - \max(-G_{g,e}, 0) \frac{1}{\gamma_{pp}^{*}} p_{p}^{*} + \rho_{g,e} \bar{\sigma}_{g,e} \frac{d_{e}^{*}}{d_{g,e}^{*}} (p_{p}^{*} - p_{E}^{*}) + \\
G_{l,e} + \max(G_{l,e}, 0) \frac{1}{\gamma_{pp}^{*}} p_{p}^{*} - \max(-G_{l,e}, 0) \frac{1}{\gamma_{pp}^{*}} p_{p}^{*} + \rho_{l,e} (1 - \bar{\sigma}_{g,e}^{*}) \frac{d_{e}^{*}}{d_{g,e}^{*}} (p_{p}^{*} - p_{E}^{*}) - \\
G_{g,w} - \max(G_{g,w}, 0) \frac{1}{\gamma_{pw}^{*}} p_{w}^{*} + \max(-G_{g,w}, 0) \frac{1}{\gamma_{pw}^{*}} p_{w}^{*} - \rho_{g,w} \bar{\sigma}_{g,w} \frac{d_{e}^{*}}{d_{g,e}^{*}} (p_{w}^{*} - p_{p}^{*}) - \\
G_{l,w} - \max(G_{l,w}, 0) \frac{1}{\gamma_{pw}^{*}} p_{w}^{*} + \max(-G_{l,w}, 0) \frac{1}{\gamma_{pw}^{*}} p_{w}^{*} - \rho_{l,w} (1 - \bar{\sigma}_{g,w}^{*}) \frac{d_{e}^{*}}{d_{g,e}^{*}} (p_{w}^{*} - p_{p}^{*}) = 0
\end{align*}
\]

The void fraction equation is obtained from the difference between the gas and the liquid continuity equations:

\[
\begin{align*}
\frac{1}{\Delta t} \left( \rho_{g,p} + \rho_{l,p} \right) \bar{\sigma}_{g,p} + \frac{1}{\Delta z} \left[ \max(\rho_{g,p} v_{g,e}, 0) \bar{\sigma}_{g,p} + \rho_{g,E} v_{g,e} - \rho_{l,E} v_{l,e} \right] + \\
\frac{1}{\Delta z} \left[ \max(\rho_{g,w} v_{g,w}, 0) \bar{\sigma}_{g,w} - \rho_{g,p} v_{g,p} \right] = 2 \gamma + \frac{\rho_{g,p} \bar{\sigma}_{g,p}}{\Delta t} + \frac{\rho_{g,p} \bar{\sigma}_{g,p}^{*} p_{p}^{*} - \rho_{l,p} (1 - \bar{\sigma}_{g,p}^{*}) p_{p}^{*}}{\Delta t} + \frac{1}{\Delta z} \left[ \rho_{l,e} v_{l,e} - \rho_{l,w} v_{l,w} \right]
\end{align*}
\]

The gas and liquid energy equations have been obtained discretizing over the main mesh. The difference between the discretized energy equation and the continuity equation multiplied by the value of enthalpy \( h_{P} \) in the center of the CV are expressed in the following form:

\[
\begin{align*}
\frac{\rho_{g,p} \bar{\sigma}_{g,p}^{*} p_{p}^{*} h_{g,p}}{\Delta t} - \frac{\rho_{g,E} h_{g,P}^{*}}{\Delta t} + \\
\frac{1}{S \Delta z} \left[ \max(-G_{g,e}, 0) S_{g} (\bar{\sigma}_{g,e} - \bar{\sigma}_{g,p}) - \max(G_{g,w}, 0) S_{w} (\bar{\sigma}_{g,w} - \bar{\sigma}_{g,p}) \right] = \\
\left[ \frac{q_{g,p} p_{g}}{S} \right]_{p} - \left[ \rho_{g,w} p_{g} \right]_{p} + \frac{1}{\Delta t} \left[ \bar{\sigma}_{g,y} - h_{g,p} \right] + \frac{\rho_{g} (p - p_{g})}{\Delta t}
\end{align*}
\]

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\[ \frac{\rho^i_{l,P}(1 - \overline{\rho}^\alpha)_{g,P} \overline{\Pi}_{l,P}}{\Delta t} - \frac{\rho^i_{l,P}(1 - \overline{\rho}^\alpha)_{g,P} \overline{\Pi}_{l,P}}{\Delta t} + \]
\[ \frac{1}{S\Delta z} \left[ \max(-G_{l,c}, 0) S_c (\overline{\Pi}_{l,c} - \overline{\Pi}_{l,P}) - \max(G_{l,w}, 0) S_w (\overline{\Pi}_{l,w} - \overline{\Pi}_{l,P}) \right] = \]
\[ \left[ \frac{q_{l,P}d}{S} \right]_P + \left[ \frac{q_{w,P}d}{S} \right]_P - \Gamma \left[ \overline{\Pi}_{ll} - H_{l,P} \right] + (1 - \alpha) \rho \left( p - p^* \right) \frac{\Delta t}{\Delta t} \]  \( (12) \)

### 2.2 Numerical resolution

The numerical resolution of the discretized equations described above has been obtained ordering these equations in a generic discretized form \( a_P \phi_P = a_E \phi_E + a_W \phi_W + b_P \). The set of algebraic equations is solved using the three diagonal matrix algorithm TDMA instead of the Newton-Raphson algorithm proposed in a previous paper (Morales et al., 2006). The coupling between momentum and continuity is solved by means of the semi-implicit pressure based method SIMPLEC (Patankar, 1980). Although this method has been commonly used in single-phase flow, when two-phase flow is present in the flux this method need some modifications (Darwish and Moukalled, 2001). The global algorithm consists of solving implicitly velocities \( v_k \), using guess pressure \( p^* \) and density \( \rho_k^* \) fields. After that, solve the pressure correction equation to obtain the pressure correction value \( p^\ast \) and correct with it the velocities \( v_k = v_k^* + v_k^\prime \), pressure \( p = p^* + p^\prime \) and densities \( \rho_k = \rho_k^* + \rho_k^\prime \). Next, the void fraction \( \alpha \), together with the energy equations in function of enthalpy \( h_k \) are solved using the updated values of pressure and velocities. Finally, the density fields must be update with the new values of pressure and enthalpy. Return to the first step and an iterative process should be make until the convergence criteria is reached. The interface momentum and energy terms have been evaluated explicitly, this criteria helps to find a solution when the interface terms are more large.

### 3. EMPIRICAL CORRELATION

Factors such as frictional coefficients, heat transfer coefficients and interfacial area are obtained by means of empirical correlations. Geometry, inclination, velocities and boundary conditions are some parameters that help to find the kind of regimen present into the pipes. A good definition of the regimen is required in order to select the most suitable empirical correlation. A simplified scheme is used to evaluate the fluid flow regimen (Levy, 1999).

The mass flux per unit volume transferred through interface can be defined as the sum of the mass transferred at the interface \( \Gamma_i \) due to the energy exchanged between phases, and the mass transfer from the wall to interface \( \Gamma_w \) due to the external heat transfer, in the following form:

\[ \Gamma_w = \frac{q_{w,P}d}{(h_{g,sat} - h_{l,P})} \text{ and } \Gamma_i = \frac{-\left[ \frac{q_{l,P}d}{S} \right]_P - \Gamma_w(h_{g,sat} - h_{l,P}) + \left[ \frac{q_{w,P}d}{S} \right]_P}{(h_{g,sat} - h_{l,P})} \text{ then } \Gamma = \Gamma_w + \Gamma_i \]  \( (13) \)

The heat transfer at the interface is evaluated by means of the heat transfer coefficient and the difference between saturated condition and liquid or gas phase temperature in the following form:

\[ \left[ \frac{q_{l,P}d}{S} \right]_P = H_{l,w}(T_{sat} - T_l)_P + \Gamma_w(h_{g,sat} - h_l)_P \text{ and } \left[ \frac{q_{w,P}d}{S} \right]_P = H_{l,w}(T_{sat} - T_g)_P \]  \( (14) \)

Where the heat transfer coefficients \( H_{w,k} \) at the wall and at the interface \( H_{k,w} \) have been found from the empirical correlations, together with the shear stresses at the wall \( \tau_{w,k} \) and at the interface \( \tau_{k,w} \). Many of these informations has been taken from the nuclear reactors literature (Idaho National Engineering, 2001) and (Ishii, 1984). In this paper, empirical expressions of stratified regimen have been used due to it is the regimen flow presented in our case. If some other regimen has to be taken into account, the empirical information must be changed and a validation of the new expression is required.
4. RESULTS

Two different cases of the numerical solution of the two-phase flow are shown by means of the numerical model proposed in this paper.

4.1 Transient case

The first case is a well-known water faucet problem. This is a transient case reported as benchmark proof in two-fluid models works, (Hewitt, 1983). It consists of a vertical pipe with 12 meters of length and 1 meter of diameter. Pipe is filled with water to next conditions: inlet void fraction 0.2, gas velocity 0.0 m/s, liquid velocity 10 m/s, and temperature 50 °C, and outlet pressure of 1.0 x 10^5 Pa.

The water faucet assumes that frictional forces and thermal effects on interface are negligible, because they have a minor effect and is possible to obtain a simplification of the problem. It case helps to see the gravity effect over the acceleration of the fluid. Illustrative results of this transient case are depicted in Figure 2, where evolution of the void fraction, and liquid and gas velocities are shown at different times.

Three meshes has been proof 20 CVs, 120 CVs and 600 Cvs, with a 1.0e^-8 as convergence criteria and a time rate of 0.0005 s. were used to solve this case. A comparison of the transient case with analytical solution at 0.5 s to void fraction and liquid velocity are depicted in Figure 3. If the grid number is increased, the numerical solution will be more near to the analytical solution.
4.2 Steady-state case

The second case is a steady-state horizontal pipe evaporator with R134a as refrigerant fluid. The evaporator is a cooper tube of a length of 6 m with a 0.00815 m of internal diameter. The boundary conditions at the inlet are: gas velocity 1.0686 m/s, liquid velocity 0.2586 m/s, pressure 3.8918e5 Pa, fluid temperature 8.12 °C and a void fraction of 0.769. An external distribution of heat flux is applied along of the tube, following the next powers values for meter: 6333.1, 5475.5, 4881.7, 4683.9, 4024.1 and 3364.4 W/m².

Figure 4: Numerical models comparison (Quasi-Homogeneous Model QHM and Two-Fluid Model TFM)

A comparison between numerical results obtained with quasi-homogeneous model QHM and two-fluid model TFM are shown in Figure 4. This case presents stratified flow along of the test tube. This fact have been evaluated by means of the different criteria to define which regimen flow is present in function of velocities and void fraction. Shorts differences have been obtained between both models in temperatures, pressure, void fraction and weight mass fraction values. However, there are a difference in the liquid velocity value between both models. While QHM predict that liquid velocity is always going down until arrive to zero, the TFM predicts that value is increased trying to follow the gas velocity behaviour. Finally, the liquid velocity begins to go down when is near to the point where two-phase flow pass to single-phase gas flow and the value goes to zero. The behaviour of the liquid velocity predicted by the TFM can be understand how the influence of the drag force applied on the liquid phase by the gas phase at the interface, while the QHM only can describe that the value of the liquid velocity always decrease because the mass of liquid is reducing along of the pipe.

Figure 5: A numerical and experimental comparison using the two-fluid model

Now, using the same case described above a numerical and experimental comparison has been used with the aim to knowing the validity of the numerical method proposed to solve the two-phase flow phenomena into the pipes. The experimental data has been obtained by means of an experimental unit (Rigola et al., 2004). Figure 5 depicts the behaviour of the pressure, void fraction, temperature and velocity distributions of gas and liquid. Two different empirical expressions have been used to define the shear stresses, due to two different numerical results have been obtained TFM-A and TFM-B. The aim to showing these differences
is to see how the numerical result depend on empirical correlations. Important differences are noted in the liquid velocity and temperatures between both results. Firstly, the prediction of the point where the fluid flow pass from two-phase to single-phase gas flow by the TFM-A is better than the TFM-B in comparison with experimental data. This point affects the wall and fluid temperature distribution. Secondly, the liquid velocity value presents an increase in both results. Although a continuous increase of the liquid velocity value until the liquid vanish into the tube is predicted by the TFM-B, the TFM-A predicts smaller increase in the liquid velocity than TFM-B and a decreasing when the liquid phase begin to vanishing into the tube. Finally, both pressure distribution along the pipe present a similar behaviour of the experimental data. However, an over-prediction is reported in TFM-A, while an under-prediction is reported by the results of TFM-B.

5. CONCLUSIONS

This paper resumes our work with two-fluid model and its application on refrigeration systems. Our first task has been to implement a semi-implicit pressure method SIMPLEC in two-phase flow to simulate this phenomena into the pipes. A set of empirical correlations have been assumed into the numerical model to evaluate the stratified regimen flow present in our case. Secondly, validation and verification of the code have been presented with a benchmark case and a numerical and experimental comparison. Two different fluids have been used in our numerical study, water and a refrigerant R134a. Thirdly, a comparison between quasi-homogeneous and two-fluid model has been considered and differences between their numerical results have been presented. After that, an important aspect to emphasize is how the two-fluid model gives more detailed information about the fluid flow, i.e. gas and liquid velocities and temperatures, than the quasi-homogeneous model. Finally, a comparative results have shown an important role of the empirical correlations and the good agreement of the numerical results with the experimental data.

REFERENCES


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