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A SENSITIVITY ANALYSIS OF THE MASS BALANCE EQUATION TERMS IN SUBCOOLED FLOW BOILING

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ABSTRACT

In a heated vertical channel, the subcooled flow boiling occurs when the fluid temperature reaches the saturation point, actually a small overheating, near the channel wall while the bulk fluid temperature is below this point. In this case, vapor bubbles are generated along the channel resulting in a significant increase in the heat flux between the wall and the fluid. This study is particularly important to the thermal-hydraulics analysis of Pressurized Water Reactors (PWRs). The computational fluid dynamics software FLUENT uses the Eulerian multiphase model to analyze the subcooled flow boiling. In a previous paper, the comparison of the FLUENT results with experimental data for the void fraction presented a good agreement, both at the beginning of boiling as in nucleate boiling at the end of the channel. In the region between these two points the comparison with experimental data was not so good. Thus, a sensitivity analysis of the mass balance equation terms, steam production and condensation, was performed. Factors applied to the terms mentioned above can improve the agreement of the FLUENT results to the experimental data. Void fraction calculations show satisfactory results in relation to the experimental data in pressures values of 15, 30 and 45 bars.

1. INTRODUCTION

The fluid flow adjacent to a heated wall under subcooled boiling regime is characterized by high heat flux rates. The subcooled boiling regime occurs when the liquid bulk temperature is lower than the saturation temperature and the wall temperature is higher. The heat flux in this situation is limited by the critical heat flux (CHF). The CHF causes a sudden increase in the wall temperature that can melt the heating wall.

The study of subcooled boiling is very important for the thermo-hydraulic analysis of Pressurized Water Reactors (PWRs), since CHF is the main design parameter for these reactors.

The main computational programs of reactor thermo-hydraulic analysis use one-dimensional models to solve this problem. These models can obtain good results for specific cases, such as a geometry type, a certain pressure or mass flow rate and liquid properties. Multidimensional models are independent on the geometry and usually they are employed by Computational Fluid Dynamic (CFD) calculations.

The CFD software FLUENT [1] uses the Eulerian multiphase model to analyze the subcooled flow boiling. In a previous paper [2], the comparison of the FLUENT results with experimental data for the void fraction presented a good agreement, both in the beginning of boiling as in nucleate boiling at the end of the channel. In the region between these two points the comparison with experimental data was not so good. Thus, in this study, a sensitivity analysis of the mass balance equation terms, vapor production and condensation, was performed.

2. MODEL DESCRIPTION

Subcooled boiling is observed at heated surfaces, when the heat flux applied to the wall is too high to be transferred to the core flow of liquid by the single-phase convective–conductive mechanisms. The term “subcooled” means that the saturation temperature is exceeded only in a local vicinity of the wall, whereas the bulk temperature is still below saturation. The point where the local wall temperature reaches the saturation temperature is considered as the onset of subcooled boiling. Steam bubbles are generated at the heated surface at nucleation sites. Further downstream the attached bubbles grow and then leave the wall at certain critical size. This critical size may depend on the surface tension and on the flow regime of the surrounding fluid. Heat transfer from the wall is then described as being carried by turbulent convection of liquid, by transient conduction due to the departing bubbles, and by evaporation. Distribution of the entire wall heat flux between these mechanisms (wall heat partitioning) can be calculated by modeling each mechanism in terms of the nucleation site density, the size of departing bubbles, their detachment frequency, and waiting time until the next bubble appears on the same site (mechanistic modeling approach). When steam bubbles move through the subcooled liquid, they condense, releasing the latent heat.

The model utilized in this work was developed initially by Kurul and Podowski [3] and has been applied in CFD codes. This model has been implemented in FLUENT via user-defined functions (UDFs) in conjunction with the Eulerian multiphase model.

The conservation equations are written for each phase, liquid and vapor, in the Eulerian multiphase model. The following is a summary of the main model equations for a certain phase *q*.

The description of multiphase flow as interpenetrating continua incorporates the concept of phase volume fractions, denoted here by α_q. Volume fractions represent the space occupied by each phase, and the laws of conservation of mass, momentum and energy are satisfied by each phase individually. The derivation of the conservation equations can be done by ensemble averaging the local instantaneous balance for each of the phases or by using the mixture theory approach.

The volume of *q* phase, V_q, is defined by

$$V_q = \int_V \alpha_q dV \qquad (1)$$

where

$$\sum_{q=1}^{n} \alpha_q = 1 \qquad (2)$$

and *n* represents the number of phases.

The effective density of q phase is

$$\hat{\rho}_q = \alpha_q \rho_q \qquad (3)$$

where ρ_q is the physical density of q phase.

The mass conservation equation for q phase is given by

$$\frac{\partial}{\partial t}(\alpha_q \rho_q) + \nabla.(\alpha_q \rho_q \vec{v}_q) = \sum_{p=1}^{n} \dot{m}_{pq} \qquad (4)$$

where $\vec{v}_q$ is the velocity vector and $\dot{m}_{pq}$ is the volumetric mass exchange rate between phases p and q.

The vapor formation rate per unit of volume can be written as

$$\sum_{p=1}^{n} \dot{m}_{pq} = \dot{m}_{lv} = \frac{(T_l - T_{sat})\, H_{R-M}\ A_i}{L + Cp_l\,(T_{sat} - T_l)} + \frac{\dfrac{Q_e\ A_t}{V_c}}{L + Cp_l\,(T_{sat} - T_l)} \qquad (5)$$

where A_i is the interfacial area density, A_t is the interfacial area density of wall, $\dot{m}_{lv}$ is the vapor formation rate per unit of volume, T_l is the liquid temperature, T_{sat} is the saturation temperature, V_c is the cell volume, L is the latent heat per unit of mass, Q_e is the evaporative heat flux, Cp_l is the liquid specific heat and H_{R-M} is the interfacial heat transfer coefficient (Ranz-Marshall).

The momentum conservation equation for q phase is given by

$$\frac{\partial}{\partial t}(\alpha_q \rho_q \vec{v}_q) + \nabla.(\alpha_q \rho_q \vec{v}_q \vec{v}_q) = -\alpha_q \nabla P + \nabla.\bar{\tau}_q + \alpha_q \rho_q \vec{g} +$$

$$+ \sum_{p=1}^{n} (\vec{R}_{pq} + \dot{m}_{pq} \vec{v}_{pq}) + (\vec{F}_q + \vec{F}_{lift,q} + \vec{F}_{vm,q}) \qquad (6)$$

where $\vec{F}_q$ is an external body force, $\vec{F}_{lift,q}$ is a lift force, $\vec{F}_{vm,q}$ is a virtual mass force, $\vec{R}_{pq}$ is an interaction force between phases, P is the pressure shared by all phases, $\bar{\tau}_q$ is the $q^{\underline{th}}$ phase stress-strain tensor and $\vec{v}_{pq}$ is the inter-phase velocity. The interfacial drag force per unit of volume is calculated as

$$\vec{R}_{pq} = \vec{R}_{lv} = 0.75\, C_d \rho_l \alpha_v |\vec{v}_r| \vec{v}_r / d_v \qquad (7)$$

where C_d is the drag coefficient and d_v is the bubble diameter.

The energy conservation equation for q phase is given by

$$\frac{\partial}{\partial t}\left(\alpha_q\rho_q h_q\right)+\nabla.\left(\alpha_q\rho_q\vec{v}_q h_q\right)=$$

$$=-\alpha_q\frac{\partial p}{\partial t}-\nabla\vec{q}_q+S_q+\sum_{p=1}^{n}\left(Q_{pq}+\dot{m}_{pq}h_{pq}\right) \tag{8}$$

where h_q is the enthalpy, $\vec{q}_q$ is the heat flux vector, S_q is the source term, Q_{pq} is the energy exchange term between the different phases, and h_{pq} is the difference in the formation enthalpies of phases p and q.

According to the Reference 2, part of the total heat flux from wall to liquid phase is partitioned into three components

$$q''_w = q''_l + q''_Q + q''_E \tag{9}$$

which are liquid convective heat flux, quenching heat flux, and evaporative heat flux, respectively. Under subcooled boiling conditions, the wall surface is subdivided into portion φ ($0 \le \varphi \le 1$), covered by nucleating bubbles, and portion ($1 - \varphi$), covered by fluid. Therefore, convective heat flux is expressed as

$$q''_l = h_{lw}\left(T_w - T_l^{cell}\right)(1-\varphi) \tag{10}$$

where h_{lw}, the single-phase heat transfer coefficient, is derived from either log law if flow is turbulent or Fourier law if flow is laminar. Liquid phase properties must be used while calculating h_{lw} for either turbulent or laminar flow.

Quenching heat flux (q''_Q) represents additional energy transfer related to liquid filling the wall vicinity after the bubble detachment

$$q''_Q = 2\pi^{-0.5}\varphi\left(f\kappa_l\rho_l C_l\right)^{0.5}\left(T_w - T_l^{cell}\right) \tag{11}$$

where f is the bubble departure frequency, κ_l is the thermal conductivity, C_l is the specific heat, and ρ_l is the density.

The evaporative heat flux is given by

$$q''_E = \frac{\pi}{6}d_{vw}^3 fn\rho_v L \tag{12}$$

where d_{vw} is the bubble departure diameter and n is the nucleation site density.

The mixture turbulence model is the default multiphase turbulence model. It represents the first extension of the single-phase $k-\xi$ model. In this case, using mixture properties and mixture velocities is sufficient to capture important features of the turbulent flow.

It was necessary to implement the various correlations and modify the equations in FLUENT through UDFs. The routines were developed by Troshco [4] and adapted to this case.

As previously mentioned the model for analysis of two-phase flow is the Eulerian multiphase flow. The boundary conditions are as follows: i) at the inlet, Velocity Inlet; ii) at the outlet, Pressure Outlet; iii) at the external wall, Heat Flux; iv) at the interface wall-liquid, the UDFs was used to specify heat flux and heat transfer coefficient; and v) at the centerline axis, Symmetry. The SIMPLE algorithm and the First Order Implicit method were used to solve the problem.

3. EXPERIMENTAL APPARATUS AND COMPUTATIONAL MESH

For the analyzes, experimental data for a heated vertical tube were used [5]. For the heated channel, a tube with an inner diameter of 15.4 mm and length of 2000 mm, fabricated of stainless steel, was used. The fluid used in the experience was water, flowing in upward direction at the test section. As the wall temperature rises above the fluid saturation temperature, vapor bubbles are formed and they migrate away from the wall. Since the bulk flow is subcooled, the bubbles condense near the center of the pipe.

The geometry used for problem formulation was two-dimensional axi-symmetric. The code manual recommends using a quadrilateral computational mesh for Eulerian multiphase model so, after several attempts to find the best computational meshes, the Fig. 1 shows the adopted grid. This mesh consists of 22000 cells.

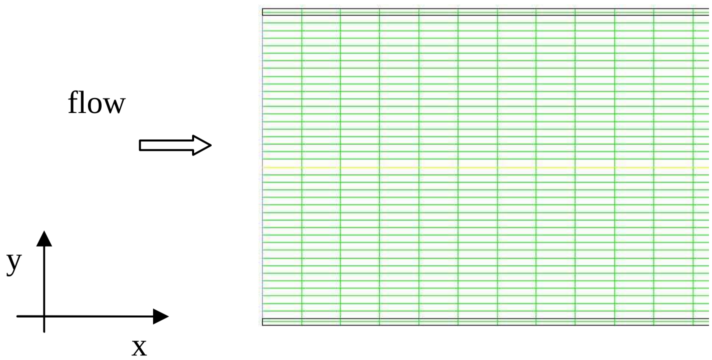

Figure 1. The mesh near the inlet used in simulation.

4. RESULTS

In a previous work [2], the values of void fraction calculated with FLUENT showed good agreement with experimental data both in the significant onset boiling as at the channel end under nucleate boiling. Between these extremes the results were not so good. In addition to the conservation equations, the model is completed with several constitutive equations that depend on parameters valid in certain ranges of use. Initially, the use of other correlations to calculate some parameters, such as diameter of the bubbles, was attempted, but the FLUENT results did not improve satisfactorily. Therefore a sensitivity study is required to better understand this problem. The terms of mass balance equation, condensation and vapor production (Eq. 5), were selected for a first step in this study.

The first term on the right hand side of Eq. 5 is the condensation of vapor bubbles in the volume

$$condensation\ term = f_c \frac{(T_l - T_{sat}) H_{R\_M} A_i}{L + Cp_l (T_{sat} - T_l)} \tag{13}$$

where the f_c factor was introduced as a correction factor that multiplies the term of the mass balance equation concerning the condensation.

The second term on the right hand side of Eq. 5 is the vapor production in the volume

$$vapor\ \ production\ term = f_{vp} \frac{\dfrac{Q_e A_t}{V_c}}{L + Cp_l (T_{sat} - T_l)} \tag{14}$$

where the f_{vp} factor was introduced as a correction factor that multiplies the term of the mass balance equation concerning the vapor production.

The input data used for the sample problems are as follows: the velocity mass flow is 900 kg/m^{2}s, the heat flux is 0.38 10^6 W/m^2 and the pressure values are 15 bar, 30 bar and 45 bar. The results will be presented in next sections for each pressure.

4.1. Pressure of 15 bar

Figure 2 shows the graph of the void fraction versus thermodynamic quality along the channel obtained with FLUENT without the application of correction factors. It is observed in the region between the beginning of boiling and the outlet channel that the calculated values are underestimated regarding to the experimental data.

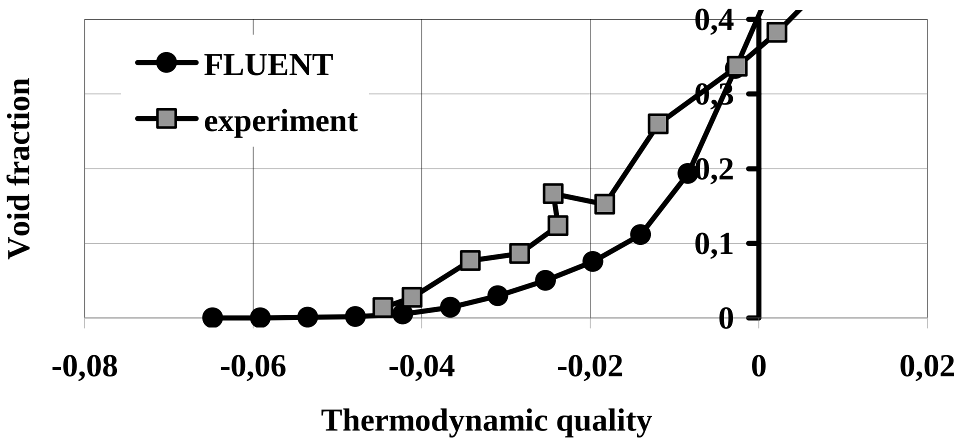

Figure 2. Void fraction versus thermodynamic quality without correction factors.

Void fraction values should be adjusted upwards because the results are below the expected calculation. Thus, although several tests had been performed with values above and below 1 for the correction factors, only values above 1 for the factor f_{vp} and below 1 for the factor f_c will be presented, because only these factors increase the void fraction calculation. Figure 3 present void fraction versus thermodynamic quality along the channel with f_c=1.0 and f_{vp}=1.2. With a 20% increase in steam production, improvement in results was almost negligible.

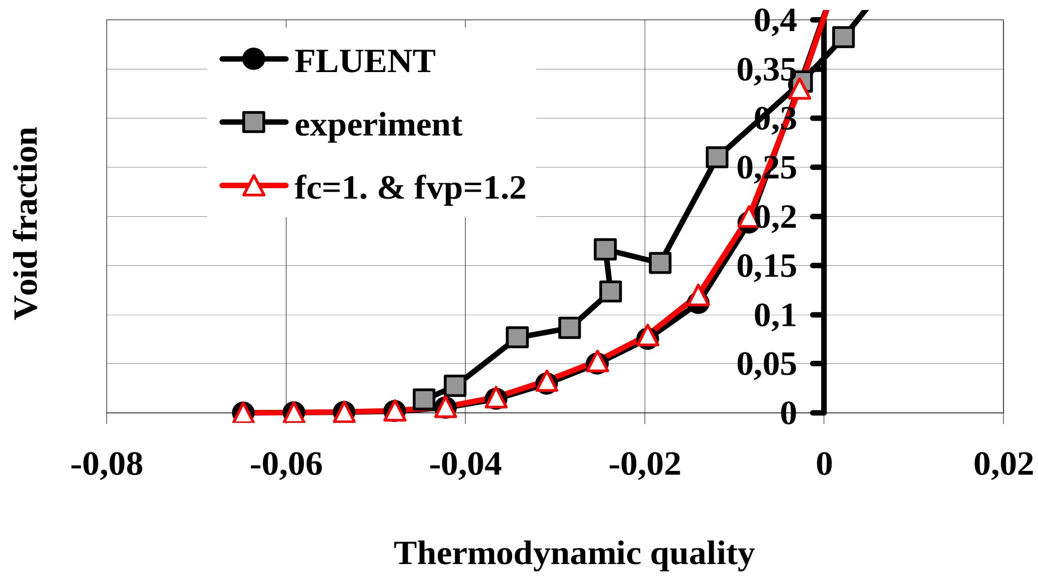

Figure 3. Void fraction versus thermodynamic quality (f_c=1.0 and f_{vp}=1.2).

Figures 4 to 6 show the graphs of the void fraction versus thermodynamic quality for the different correction factors applied. In Fig. 4, f_c=0.8 and f_{vp}=1.0; in Fig. 5, f_c=0.6 and f_{vp}=1.0; and in Fig. 6, f_c=0.8 and f_{vp}=1.2.

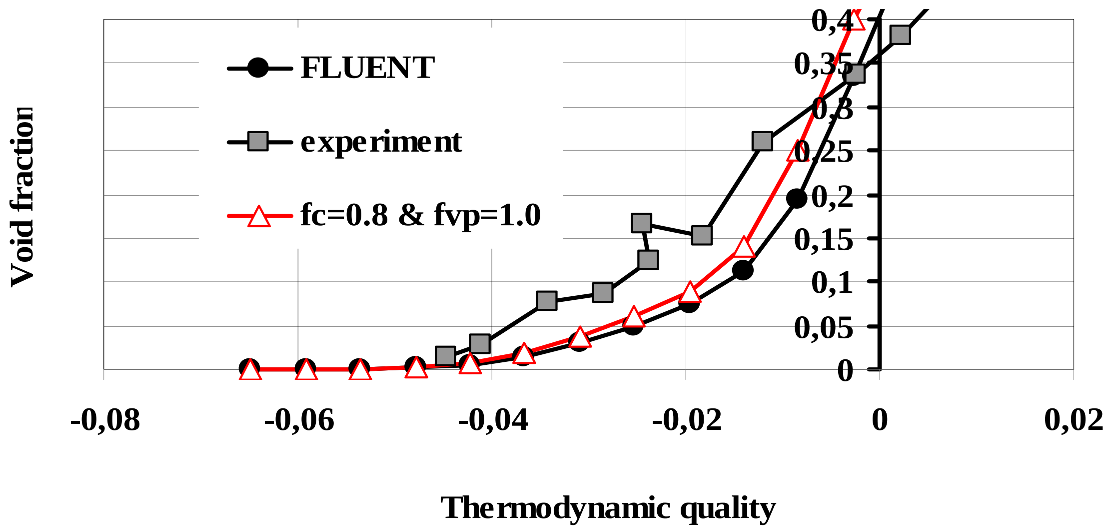

Figure 4. Void fraction versus thermodynamic quality for 15 bar (f_c=0.8 and f_{vp}=1.0).

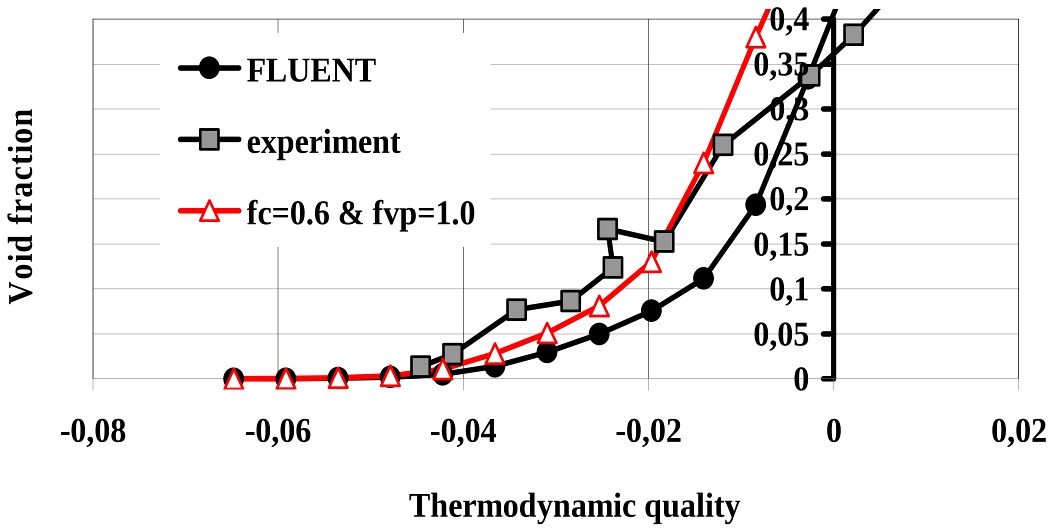

Figure 5. Void fraction versus thermodynamic quality for 15 bar (f_c=0.6 and f_{vp}=1.0).

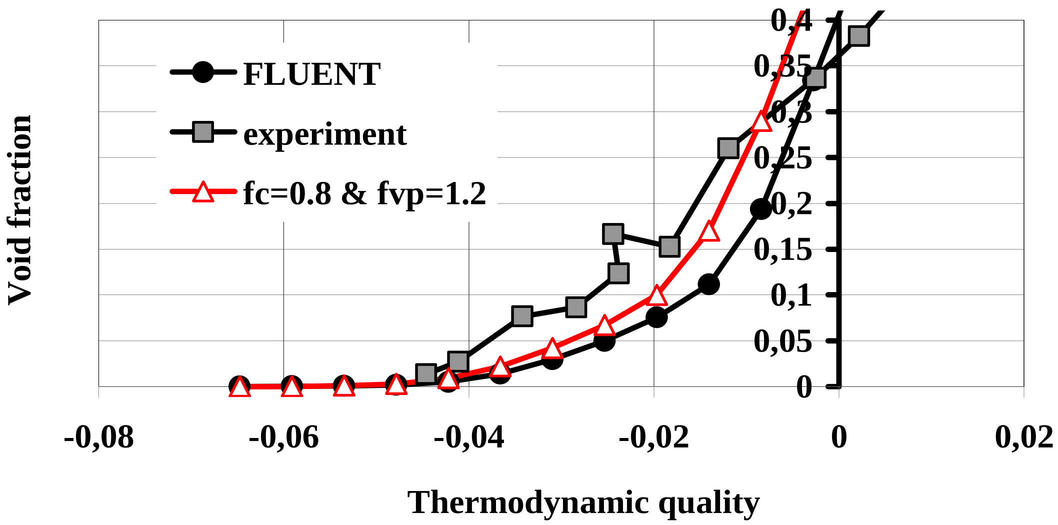

Figure 6. Void fraction versus thermodynamic quality for 15 bar (f_c=0.8 and f_{vp}=1.2).

In Fig. 4, with f_c=0.8, the results showed a significant improvement in comparison with the experimental data. In Fig. 5, with f_c=0.6 in the region of sub-cooled boiling, the results were closer to the experimental data, but in the nucleate boiling region, the results were worse. In Fig. 6, with f_c=0.8 and f_{vp}=1.2, the previous conclusion was observed.

4.2. Pressure of 30 bar

In pressure of 30 bar, Figs. 7 to 9 show the graphs of the void fraction versus thermodynamic quality for the different correction factors applied. In Fig. 7, f_c=1.0 and f_{vp}=1.2; in Fig. 8, f_c=0.8 and f_{vp}=1.0; and in Fig. 9, f_c=0.8 and f_{vp}=1.2.

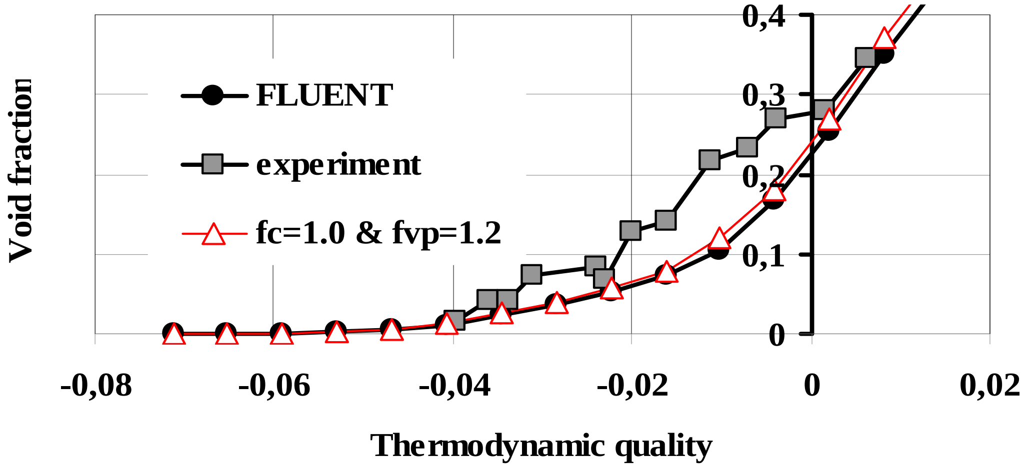

Figure 7. Void fraction versus thermodynamic quality for 30 bar (f_c=1.0 and f_{vp}=1.2).

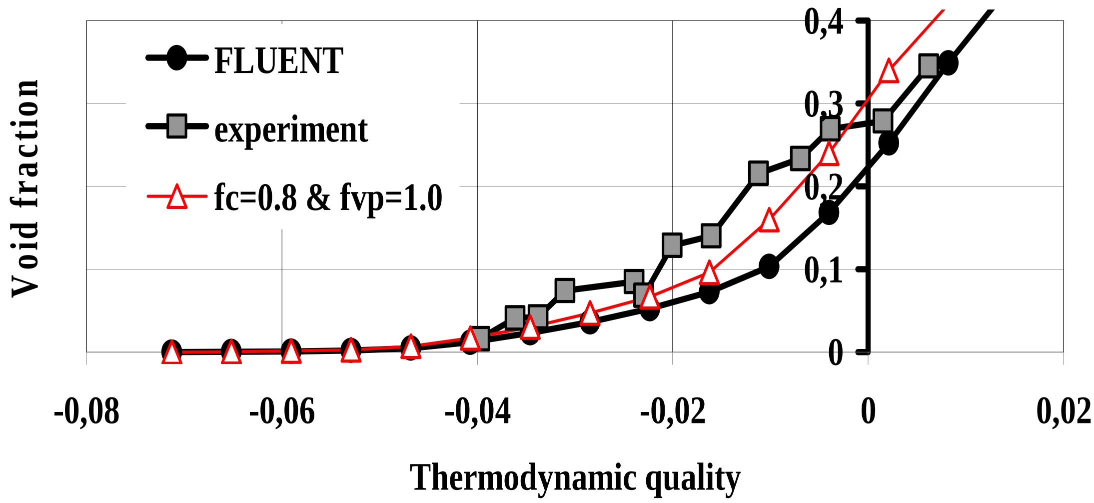

Figure 8. Void fraction versus thermodynamic quality for 30 bar (f_c=0.8 and f_{vp}=1.0).

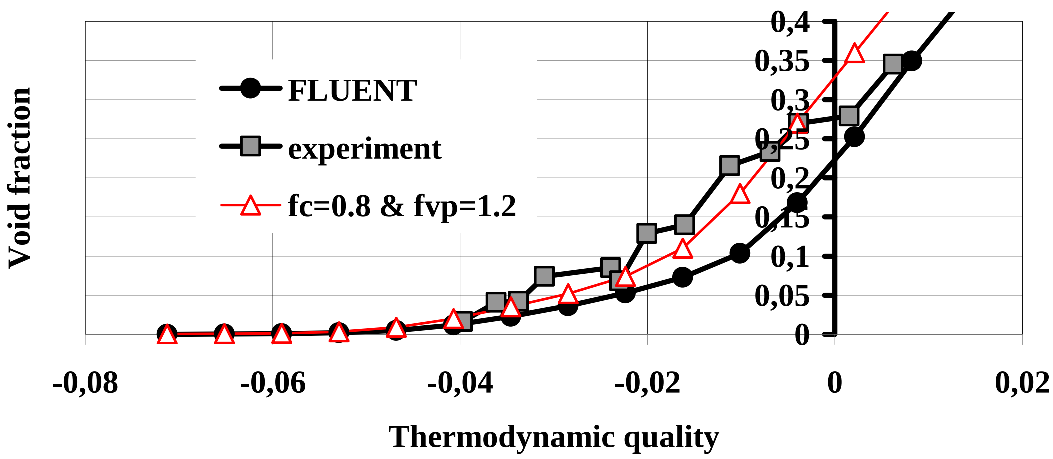

Figure 9. Void fraction versus thermodynamic quality for 30 bar (f_c=0.8 and f_{vp}=1.2).

The tendency of the void fraction calculations at a pressure of 30 bar showed similar results to the pressure of 15 bar, but in this case, the influence of the correction factors was greater.

4.3. Pressure of 45 bar

Similarly to the previous case, in pressure of 45 bar, Figs. 10 to 12 show the graphs of the void fraction versus thermodynamic quality for the different correction factors. In Fig. 10, f_c=1.0 and f_{vp}=1.2; in Fig. 11, f_c=0.8 and f_{vp}=1.0; and in Fig. 12, f_c=0.8 and f_{vp}=1.2.

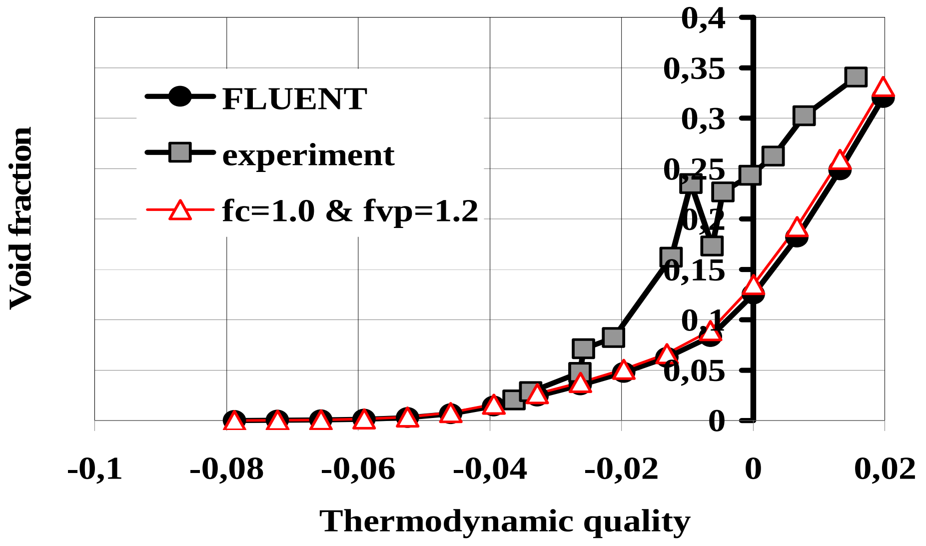

Figure 10. Void fraction versus thermodynamic quality for 45 bar (f_c=1.0 and f_{vp}=1.2).

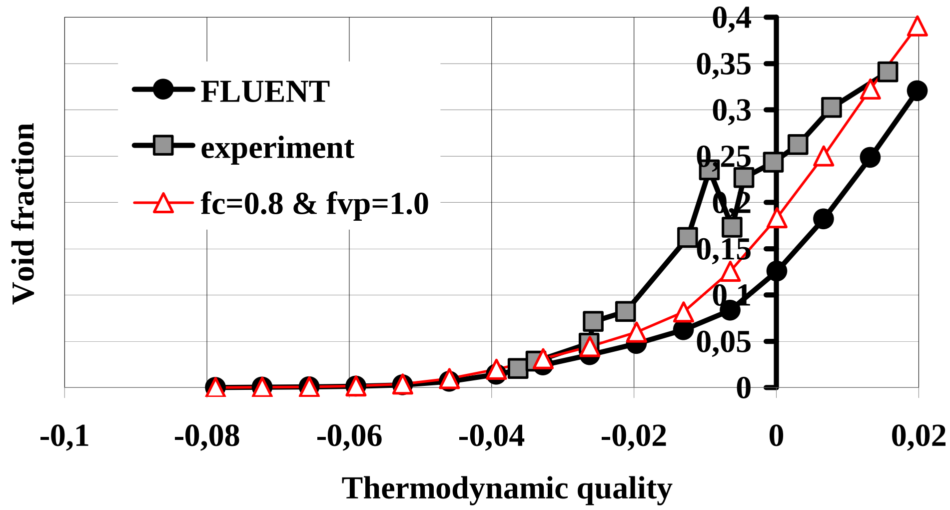

Figure 11. Void fraction versus thermodynamic quality for 45 bar (f_c=0.8 and f_{vp}=1.0).

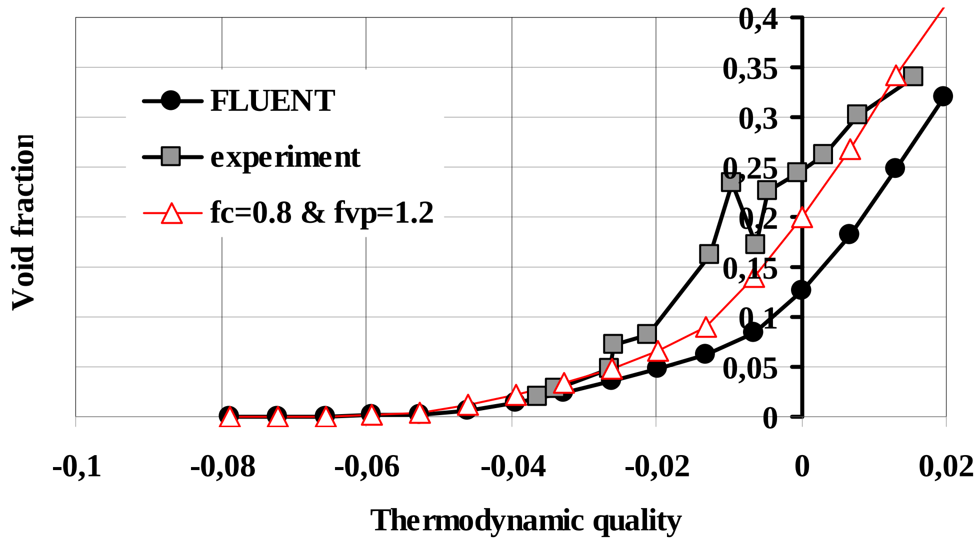

Figure 12. Void fraction versus thermodynamic quality for 45 bar (f_c=0.8 and f_{vp}=1.2).

It was clear that with increasing pressure, the influence of the correction factors is more noticeable. Although the modification of both factors simultaneously, f_c and f_{vp}, has approximate more the results of the experimental data, the tendency is that the void fraction calculation move away from the experience in the nucleate boiling region.

5. CONCLUSIONS

In an attempt to improve the results of the void fraction calculation by FLUENT, a sensitivity analysis of the mass balance equation terms, vapor production and condensation, was proposed.

For all three evaluated pressures (15 bar, 30 bar and 45 bar) the obtained results presented the same trends. The influence of the factor related to the vapor production, f_{vp}, was small, but the factor applied to the condensation term, f_c, was efficient. When both factors were applied at the same time the results were better in subcooled boiling region, but the results began to pull away of the experimental data in nucleate boiling region.

The condensation term showed the greatest influence on the fit of the void fraction calculations to the experimental data. This term is a function of the coefficient of heat transfer between the phases, $H_{R\text{-}M}$, and interfacial area density, A_i. These parameters are obtained from empirical correlations and they will be the subject of future research, in order to improve the results.

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