V European Conference on Computational Fluid Dynamics ECCOMAS CFD 2010 J. C. F. Pereira and A. Sequeira (Eds) Lisbon, Portugal, 14–17 June 2010

CFD MODELING OF ISOTHERMAL AND NON-ISOTHERMAL BUBBLY FLOW IN VERTICAL CONDITION USING ANSYS CFX 12

Filippo Pellacani^{*}, Silvana Matturro Mestre[†], Sergio Chiva Vicent[†] and Rafael Macian Juan^{*}

*Department of Nuclear Engineering Technische Universität München Garching, 85748 Germany e-mail: pellacani@ntech.mw.tum.de e-mail: macian@ntech.mw.tum.de

[†] Department of Mechanical Engineering and Construction Universitat Jaume I Castelló de la Plana, 12080 Spain e-mail: schiva@emc.uji.es

Key words: Fluid Dynamics, Bubbly Flow, Interfacial forces, Subcooled boiling, Wall Heat Partition

Abstract. Upward isothermal and non-isothermal turbulent bubbly flow in tubes is numerically modeled using ANSYS-CFX v12 with the aim of creating the basis for the reliable simulation of a vertical channel of a nuclear reactor as long term goal. The interfacial non-drag forces are investigated first in the isothermal simulations and then are also included in the model for the non-isothermal simulations is used for the wall lubrication force. A Tomiyama model modified by Frank [5] and the Antal [4] model with different coefficients is used for the wall lubrication force. The lift force was calculated in two ways, with the Tomiyama model [3] and based on a constant value. The subcooled boiling simulation is based on the RPI wall boiling model developed by Kurul and Podowski [1]. The interfacial non-drag forces, previously investigated, are included in the model. The simulation results for the isothermal case are compared against experimental data of Hibiki [19]. The liquid superficial velocity is in the range between 0.5 and 1 (m/s) and the void fraction average varies from 5 up to 20%. The void fraction axial profiles for high pressure subcooled boiling in tubes are compared against the experimental data of Bartolomej [20]. The pressure varies from 3 up to 4.5 *MPa*. The models give predictions in close agreement with experimental results. In case of adiabatic bubbly flow the radial profiles of the void fraction obtained from the simulation are in good agreement with the experimental results. The main difficulties for the simulation are observed for flow in transition to flow regimes with high void fraction, when the bubbly flow is not able to maintain the spherical condition of the bubbles, which is a requirement of the boiling models.

1 INTRODUCTION

Two-phase flow occurs in a wide range of industrial application. It is important among many other systems in water-cooled nuclear reactors. Here the presence of bubbles influences the density of the moderator and so the reactivity response of the system. In these situations knowledge of the two-phase flow conditions is paramount for determining the reaction kinetics.

Bubbly flow is generated in subcooled boiling flow condition. It occurs when the local wall temperature during the heating of a subcooled liquid is above the saturation temperature and sufficiently high for bubbles nucleation to occur. Subcooled boiling is characterized by bubbles formations at the heated wall: It could appear under the form of isolated bubbles or as a bubbly layer along the wall. The relative motion between the phases generated by internal or, if present, external forces sweep the bubbles into the subcooled liquid core and then condense.

Two-phase flow dymanics and boiling have been studied extensively during the last decades. In case of boiling these studies have mostly lead to very useful and well working empirical correlations. Their limit is that they fails when applied to situations that go beyond the experimental range over which experimental data have been collected. Kurul and Podowski [1], proposed their own modifications of the two-fluid model and closure laws based both on empirical and mechanistic considerations. In case of two-phase flow dynamics, the development of new measurement techniques led to better descriptions of the physical phenomena and of the forces acting on the phases. The improvement of the physical models helped the setting up of interfacial forces model that able to reproduce the distribution of the phases in a given system describing the interfacial forces such has the drag force [2], the lift force [3], the wall lubrication force [4, 5].

The interfacial forces models and subcooled boiling model have been applied in this contribution to the simulation of upward flow in a vertical pipe for an adiabatic bubbly flow, [19], and for subcooled boiling at high pressure in turbulent flow [20] conditions. The general-purpose computational fluid dynamics (CFD) code ANSYS CFX 12 was used for solving the two-fluid model and the relevant closure relations.

2 MATHEMATICAL MODELS

2.1 Two-fluid model

Mass and Momentum Conservation Equations

The simulations presented in this paper are based on the two-fluid model Eulerian– Eulerian approach. The liquid phase is considered as the continuous phase and the gas phase is considered as dispersed. The constitutive equation of the two-fluid approach presented by Ishii [6] and Drew and Lahey [7] can be written as

$$\frac{\partial(\rho_i \alpha_i)}{\partial t} + \nabla \cdot \left(\rho_i \alpha_i \vec{U}_i\right) = 0 \tag{1}$$

For the mass equation and for the momentum equation for the two-phase mixture can be expressed as follows:

$$\frac{\partial \left(\rho_{i} \alpha_{i} \vec{U}_{i}\right)}{\partial t} + \nabla \cdot \left(\rho_{i} \alpha_{i} \vec{U}_{i} \vec{U}_{i}\right) = -\alpha_{i} \nabla P + \rho_{i} \alpha_{i} \vec{g} + \nabla \cdot \left[\mu_{i} \alpha_{i} \left(\nabla \vec{U}_{i} + \left(\nabla \vec{U}_{i}\right)^{T}\right)\right] + F_{i}$$

$$\tag{2}$$

The term F_i in eq. (2) represents the total interfacial force acting on the phases. Closure laws are needed to calculate the momentum transfer of the total interfacial force.

Modelling of the interfacial Forces

Four interfacial forces have been considered during the analysis. The drag force $F_{\rm D}$, has been modeled using the Grace model [2]. The non-drag forces considered are the lift force $F_{\rm L}$, the wall lubrication force $F_{\rm WL}$, and the turbulent dispersion force $F_{\rm TD}$. The virtual mass force was neglected since tests conducted by Frank et al. [5] showed that its influence is of minor importance in comparison with the amplitude of the other drag and non-drag forces.

$$F_{i} = F_{D} + F_{L} + F_{WL} + F_{TD}$$
(3).

Drag Force

The drag force accounts for the drag of one phase on the other and the coefficient that has been used is that of Grace et al. [2].

$$\vec{F}_D = \frac{3}{4} C_D \frac{r_\beta \rho_\alpha}{d} \left| \vec{U}_\beta - \vec{U}_\alpha \right| \left(\vec{U}_\beta - \vec{U}_\alpha \right)$$
(4)

Lift Force

Due to velocity gradient, bubbles rising in liquid are subjected to a lateral lift force. This is modeled according to the Tomiyama [3] formulation

$$\vec{F}_{L} = -C_{L} \cdot r_{\beta} \rho_{\alpha} \cdot \left(\vec{U}_{\beta} - \vec{U}_{\alpha}\right) \cdot \overrightarrow{rot} \left(\vec{U}_{\alpha}\right)$$
(5)

For the evaluation of the lift coefficient C_L two methods have been used, namely, a constant value of 0.06 and a value calculated according to Tomiyama [5].

For Tomiyama model the evaluation of the lift coefficient is based on the definition of the modified Eötvös number,

$$Eo_d = \frac{g.(\rho_l - \rho_g).d_h^2}{\sigma} \tag{6}$$

where d_h is the maximum horizontal bubble dimension that is calculated using this empirical expression:

$$d_h = d_b \sqrt[3]{1 + 0.163 \cdot Eo^{0.757}} \tag{7}$$

The lift coefficient proposed by Tomiyama has this form

$$C_{L} = \begin{cases} \min \left[0.2888 \cdot \tanh(0.121 \cdot Re), f(Eo_{d}) & Eo_{d} < 4 \\ f(Eo_{d}) & 4 < Eo_{d} < 10 \\ -0.27 & Eo_{d} > 10 \end{cases}$$
(8)

Where $f(Eo_d)$, the Eötvös number function, is defined as

$$f(Eo_d)0.00105Eo_d^3 - 0.0159Eo_d^2 - 0.0204Eo_d + 0.474$$
(9)

The behavior of the Tomiyama lift coefficient is a function of the bubble diameter d_b . A change of sign occurs, for air-water at atmospheric conditions, when the bubble reaches

the critical diameter of 5.8 mm. Bubbles with a diameter smaller than this value will be pushed toward the wall. Bubble with a diameter bigger than 5.8 mm will be moved toward the pipe centerline. At higher pressures the critical diameter becomes smaller.

Wall Lubrication Force

Due to surface tension, a lateral force appears to prevent bubbles attaching on the solid wall. The wall lubrication force has been modeled as it follows:

$$F_{WL}^{fluid} = -F_{WL}^{gas} = -\alpha_d \cdot \rho_c \cdot \frac{1}{d_p} \cdot C_{WL} \cdot \left(\vec{U}_{rel} - \left(\vec{U}_{rel} \cdot \hat{n}_w \right) \cdot \hat{n}_w \right)^2 \cdot \hat{n}_w$$
(10)

The wall lubrication coefficient C_{WL} in the Antal formulation [4] has the following expression:

$$C_{WL} = max \left\{ C_1 + C_2 \frac{d_p}{y_w}, 0 \right\}$$
(11)

The values used for C_1 and C_2 are -0.0064 and 0.016 as proposed by Krepper and Prasser [68] Also the pair of value (-0.025, 0.075) proposed by Krepper et al. [9] has been tested. The implementation of the wall lubrication force is necessary for the adiabatic two-phase flows, as it plays an important role to reproduces the void fraction peak near the wall [10]. Krepper [9] reports that its use at high-pressure wall boiling conditions may be questionable, but also that further research is necessary to improve the existing wall force models. In the adiabatic case also the model of Tomiyama modified by Frank [5] was tested. In this case the wall lubrication coefficient C_{WL} has the following expression:

$$C_{WL} = C_{W3}(Eo) \cdot max \left\{ 0, \frac{1}{C_{WD}} \cdot \frac{1 - \frac{y_W}{C_{WC} \cdot d_p}}{y_W \cdot \left(\frac{y_W}{C_{WC} \cdot d_p}\right)^{p-1}} \right\}$$
(12)

where

$$C_{W3} = \begin{cases} e^{-0.993Eo + 0.179} & 1 \le Eo \le 5\\ 0.00599Eo + 0.0187 & 5 < Eo \le 33\\ 0.179 & 33 < Eo \end{cases}$$
(13)

This formulation, as well as the Antal formulation (eq. 11) is geometry independent but leads to results in general with a higher absolute value.

Turbulent dispersion force

A turbulent dispersion force has been considered to take into account the turbulence assisted bubble dispersion. The turbulent dispersion force model that has been used is the Favre averaged Drag force (FAD) [11]. This force is modeled as:

$$F_{TD} = -C_{TD} \cdot C_{cd} \cdot \frac{v_{t,c}}{s_{ct,c}} \left(\frac{\nabla r_d}{r_d} - \frac{\nabla r_c}{r_c} \right)$$
(14)

where C_{cd} is the momentum transfer coefficient for the interphase drag force. The model depends on the details of the drag correlation used. Sc_t is the turbulent Schmidt number for continuous phase; it is taken to be 0.9. C_{TD} is a multiplier. Its value is unity.

Bubble induced turbulence

The Bubble induced turbulence has been taken into account according to Sato's [12] model. This model introduce a new term in the viscosity of the liquid to take in account the effect of turbulence enhancement produced by the bubbles.

2.2 Modeling of the subcooled boiling

The wall partitioning model proposed in the work of Kurul and Podowski [1] is based on the division of the heat flux applied on the heated surface into three different terms, convective, quenching and evaporative.

$$Q_w = Q_e + Q_c + Q_q \tag{15}$$

For the definition of each term, closure relationships are required. At the actual state of development they are based essentially on empirical correlations rather than on physical, mechanistic models.

Convective heat flux

The convective heat flux is modeled as it follows:

$$Q_c = h_c \cdot A_1 \cdot \left(T_s - T_{w,l}\right) \tag{16}$$

In this expression the calculation of h_c following Kurul and Podowski [1] is based on a one-dimensional Stanton number correlation obtained from the Reynolds analogy:

$$h_c = St \cdot \rho_l c_{p,l} u_l \tag{17}$$

where the Stanton number is defined as:

$$St = \frac{Q_w}{Nu \cdot \text{Re} \cdot \Delta T}$$
(18)

This formulation has the problem to be dependent on the position of the location closest to the wall. Following Egorov and Menter [13] the heat transfer coefficient h_c is calculated based on the wall turbulent function and is independent on the actual calculation grid.

The heat partitioning model considers the whole wall surface as being separated into two fractions: A_1 is the area fraction influenced only by the single-phase convection heat transfer and the fraction A_2 is the area fraction of bubble influence and is defined as:

$$A_2 = \min\left(\pi \cdot F \cdot d_d^2 \cdot N_a, 1\right) \tag{19}$$

F is a correction factor that is considered per default equal to 2. N_a is the wall nucleation site density. The correlation to determine the bubble departure diameter, d_d , is based on the work of Tolubinski and Kostanchuk [14]:

$$d_{d} = \min\left(d_{ref} \cdot \exp\left(-\frac{\Delta T_{sub}}{\Delta T_{ref}}\right), d_{\max}\right)$$
(20)

where d_d , d_{ref} , ΔT_{ref} and d_{max} are chosen to fit pressurized water data.

Evaporative heat flux

The evaporative heat flux is modeled as it follows:

$$Q_e = N_a \cdot f \cdot \left(\frac{\pi \cdot d_d^{-3}}{6}\right) \cdot \rho_g \cdot h_{fg}$$
(21)

The bubble release frequency f is calculated based on the Cole's [15] empirical correlation:

$$f = \sqrt{\frac{4g(\rho_l - \rho_g)}{3d_d \rho_l}}$$
(22)

This expression has been employed rather successfully to simulate high pressure boiling flows as reported by Yeoh et al. [16], but the applicability to other pressure ranges remains debatable. N_a according to Končar et al. [17] is the nucleation site density:

$$N_a = 210 (T_s - T_{sat})^{1,805}$$
(23)

Quenching heat flux

The quenching heat flux is modeled as it follows:

$$Q_q = h_q \cdot A_2 \cdot \left(T_s - T_{w,l}\right) \tag{24}$$

The quenching heat transfer coefficient h_q is modeled using the expression of Del Valle and Kenning [18]. In CFX, the characteristic temperature $T_{w,l}$ is taken from the reconstructed temperature profile using the temperature wall function and the given value of the non-dimensional distance y+ from the wall. A constant value of 250 is used for y+. The heat flux at the wall is transferred by the quenching mechanism during the so called bubble waiting time t_w . This is the time interval between departures of consecutive bubbles. The formulation of Tolubinski and Kostanchuk [14] is choosen to determine t_w :

$$t_w = \frac{0.8}{f} \tag{25}$$

The area fraction A_1 is defined as

$$A_1 = 1 - A_2 \tag{26}$$

3 VALIDATION OF THE MODELS

To show the adequacy of the calculated results, the interfacial forces models and the boiling model described previously were implemented to simulate, for the adiabatic case, the experiment of Hibiki 2001 [19] and for the non-isothermal case, the experiment of Bartolomej 1967 [20]

3.1 Isothermal:

As we commented above, we have use the work performed by Hibiki et al. [19] in order to check the capabilities of the models proposed for isothermal conditions. In that work the time-average radial profile of the void fraction, interfacial velocity, interfacial area concentartion and Sauter mean diameter were measured at three specific axial locations (z/D=6, z/D=30.3, z/D=53.5) and 15 radial locations (r/R=0 to 0.95) in upward water air two-phase flow using a double-sensor conductivity probes. In the test section water enters at 20°C and adiabatic conditions have been maintained. The pipe

internal diameter is 50.8 mm, the vertical test section is 3061 mm long. The flow conditions covered most of a bubbly flow region, including finely dispersed bubbly flow and bubbly-to-slug transition flow regions. The experimental points considered in this contribution are indicated in red in Figure 1 (Points-A,B,C,D).



Figure 1: Maps of phase distribution patterns at z=D=53.5 [19]

Values of the inlet average flow conditions for the four experimental points under analysis are reported in Table 1.

	j_{f} (m/s)	$j_{g,0}$ (m/s)	$\alpha_{z/D=53.5}$ (%)
Α	0.491	0.0275	4.9
В	0.491	0.0556	9.2
C	0.491	0.129	19.2
D	0.986	0.113	10.8

Table 1: Flow Conditions.

Several grids were tested in order to set up the computational domain. After the mesh sensitivity analysis the shortest computational time and also independency of the results from the calculation grid were provided by the model composed by approximately 110000 nodes that represents one eighth of a vertical pipe using symmetric boundary conditions for both axial cut planes. On the radial direction the number on node is 22. The first node near the wall was set at a distance to obtain a value of the y+ in the range [30-40], in order to avoid numerical oscillations and for an accurate wall lubrication force modeling. The bottom boundary conditions for the volume fraction of the gas phase and the velocity of the liquid phase are the available experimental profiles at z/D=6. An approximation of the diameter of the dispersed phase is obtained from the available profiles of the Mean Sauter Diameter at the lower and upper axial locations. At the outlet section a constant pressure condition is considered, and in all the cases the value given is the atmospheric pressure. A RANS turbulence model, based on the SST, for the liquid phase is considered. A zero equation turbulence model for the gas phase is used. The lift force was modeled using the Tomiyama model [4]. The Grace model [2] model for the drag force is used.

In the different calculation series the wall lubrication force was modeled based on Antal [5], with coefficients from Krepper et al. [8], and Frank et al. [5] (see Table 3 for a

detailed overview). For this case not only the original coefficients [5] have been used; other two sets of coefficients have been tested with the aim of reducing higher absolute value of the wall lubrication force and also to reduce its action at the near wall region (see Figure 2). In Figure 2, the behavior of the C_{WL} (eqs. 11 and 12,) in function of the wall distance y_w , is shown for the different sets of coefficients tested in this contribution, the value of the bubble diameter, the Eötvös number and of the pipe diameter have been kept constant and the value are indicated on the diagram itself. The values are calculated departing from a value of y_w equal to 0.3 mm.

The Tomiyama model, modified by Frank [5], and with the original coefficients, lead to very high values when the distance of the first node at the wall is very small. This could lead to numerical instabilities during the calculation and increasing the value of the y+ is needed. For the Experimental Point A and D also the poly-dispersed MUSIG (Multi Size Group) approach has been tested (Series 4 and 5). Table 3 resumes the different models and coefficients for the wall lubrication force used for the calculation.

Series	WLF	C1	C2	-	MUSIG
1	Antal	-0.0064	0.016	-	NO
5	Antal	-0.0064	0.016	-	YES
Series	-	Cwc	Cwd	р	MUSIG
2	Frank	8	8	1.2	NO
3	Frank	10	6.8	1.2	NO
4	Frank	8	8	1.2	YES
6	Frank	10	6.8	1.7	NO

Table 3: Parameters of the Wall lubrication force tested in the simulations.





When, (Fig. 4 A,B,C - Series 1), the Antal model with coefficients from Krepper and Prasser [8] is used, it leads to an underestimation of the wall lubrication force effect. The void fraction distribution along the radius presents a peak that is not in accordance to the experimental data. Simulating the wall lubrication force using the model that was modified by Frank [5] with original coefficients (Fig. 3 A - Series 6), leads to the overestimation of the force effect. This trend is also predicted by the diagram of Figure 2 representing the C_{WL} in function of the wall distance. The new sets of coefficients

tested for this last model lead both to similar results (Figure 3 A,B,C,D - Series 2 and 3). The position of the void fraction peak, in these cases, is in good agreement with the experimental profiles. The homogeneous polydispersed approach (Homogeneous MUSIG) was also tested (Fig3. A - Series 4 and 5; Fig. 1 D - Series 4) This approach, for this case, led to results in line with the monodispersed approach (Series 2 and 3).



Figure 3: Dependence of α on r and j_g.

3.2 Non-ishothermal-

For the non-isothermal case, since it was commented, the experiments performed by Bartolomej [20] were considered. The set-up of this experiment is illustrated schematically in Fig. 4. In these experiments the average cross-section void fraction was measured at one specific axial location in upward water flow location. In the test section sub-cooled boiling occurred and steam was generated. The pipe internal diameter is 15.4 mm, the test section length is 2 m long and it is completely heated.

Values of the system pressure P, mass flux G_{in} , wall heat flux q'' and inlet subcooling T_{sub} for this experiments are presented in Table 2.



 $\begin{tabular}{|c|c|c|c|c|} \hline Variable & Value \\ \hline P (MPa) & 3 (Series 2) \\ \hline 4.5 (Series 3) \\ \hline r (mm) & 7.7 \\ \hline Gin (kg/(s m^2)) & 900 \\ \hline q^{\prime\prime} (MW/m^2) & 0.38 \\ \hline T_{sub} (K) & \sim 20 \ K \\ \hline \end{tabular}$

Figure 4: Schematic illustration of the Bartolomej's 1967 test section.

Table 2: Parameters of the experimental set-up.

In our work several models concerning turbulence, drag and non drag forces models have been tested. We have used two different turbulence models for the simulation of upward subcooled flow boiling, k- ϵ turbulence model, like in [17, 21, 22], and the SST model. Results are presented and discussed. The Grace model [2] for the drag force is considered. The lift force was calculated in two ways, with the Tomiyama model [3] and based on a constant value. Further, two different set of coefficients for the wall lubrication force from the literature [8, 9] have been tested and the results are presented and compared. The purpose of this work is to show the influence of the proposed model assumptions in the accuracy of the simulations by comparing them to the experimental data.

Several grids were tested in order to set up the computational domain. After the mesh sensitivity analysis the shortest computational time and also independency of the results from the calculation grid were provided by the model composed by approximately. 70000 nodes that represents one eighth of a vertical pipe using symmetric boundary conditions for both axial cut planes. On the radial direction the number on node is 20. The first node near the wall was set at a distance to obtain a value of the y+ around 70, in order to avoid numerical oscillations. A screenshot of the 2D grid perpendicular to the flow direction is shown in figure 5. The temperature bottom boundary condition is a mean value extrapolated from the available experimental profile. For the upper part, a constant pressure condition is considered, and in all the cases the value given is the experimental system pressure. A RANS turbulence

model, based on the SST model and k- ϵ model, for the liquid phase is considered. A zero equation turbulence model for the gas phase is used.



Figure 5: 2d grid used for the definition of the computational domain normal to the main flow direction

Figure 6 shows the results obtained when the subcooled boiling is modeled with and without the presence of a lift force (figures 6B and 6A respectively).



Figure 6: Dependence of α on x - effect of the lift force model of Tomiyama [3].

In general the void fraction versus the thermodynamic quality is always over predicted, but more when the lift model is considered. Since the governing system parameters (*P*, *q*'', *G_{in}*) allow the generation of bubbles that remain far below the critical value of 5.8 mm for the lift force; the lift coefficient has a constant positive value of 0.288 (in equation 5 we are in the range $Eo_d < 4$). The lift force, with its high value, prevents to the bubbles generated at wall to move to the centerline (Fig. 6B) where they could condense since in this region the liquid is still subcooled. In fact, during subcooled boiling in the region where the thermodynamic quality is negative (x<0) the steam condensation process takes place.

The bubble diameter was modeled according to Kurul and Podoswki [23], with the bubble size dependent on liquid subcooling. Exactly, the bubble diameter is inversely proportional to the liquid subcooling. The results of Fig. 6 (A) and (B) are comparable when saturation conditions are reached (x>0). The lines labeled "*Poly*." represent polynomial fits of the experimental results as a means of better assessing the quality of the simulation results.

In the simulations results shown in Fig. 7 the wall lubrication force model of Antal with the coefficients from Krepper et al. [8] was added. The wall lubrication force under these conditions has a very low effect on the flow and is concentrated in the wall near region were the temperature of the liquid is near of above the saturation. The results are similar to the case of Fig. 6 (B).



Figure 7: Dependence of α on x - effect of the wall lubrication force (Antal with coefficients from Krepper et al. [7]).

In Fig. 8, a constant lift coefficient was set with a value of 0.06 with the aim to reduce the effect of the lift force that prevented the bubble to move to the centerline. The Tomiyama coefficient was used in the previous calculations (Fig. 6 (B) and Fig. 7). The two lubrication force coefficients were adjusted in order to obtain two effects. Achieve a higher absolute value of the wall lubrication force and also to extend its action not only at the near wall region. With this modifications a lower calculated axial mean value of the void fraction are achieved, but they are still slightly over predicted (Figure 8)



Figure 8: Dependence of α on x - effect of the wall lubrication force (Antal with coefficients C1=-0.025 C2=0.075).



Figure 9: Dependence of α on x - effect of the sst turbulence model.

The SST turbulence model was also tested (see Fig. 9) and the effect on the results was the increase of the void fraction for the same thermodynamic quality. The SST model resulted in a better turbulent mixing of the flow which lowered the temperature difference between the wall and the flow centerline and reduced the collapse of the bubbles. Comparing the turbulence kinetic energy of the liquid at the same level of thermodynamic quality (see Fig. 10), the turbulent energy content of the flow is higher when the SST model is used compared to the results obtained with k- ϵ model.



Figure 10: Turbulence kinetic energy along the radius in function of the turbulence model for x=-0.03 [-] for the series 3.

4 CONCLUSIONS

In the present work two test cases were analyzed with the specific goal of assessing the current models used for the forces, turbulence, bubble diameter and the inter-phase energy exchange .In general good qualitative agreement was obtained between the experimental data and the simulations results. The models give predictions in close agreement with experimental results.

In case of adiabatic bubbly flow the radial profiles of the void fraction obtained from the simulation are in good agreement with the experimental results produced by Hibiki et al. [19]. The assessment of the interfacial force models used in this work has been carried out based on water air data at low pressure and room temperature. In the literature, experimental data at high pressure with detailed radial profiles for the most important physical parameter like phase velocities, temperatures, void fraction, are not easily found. For these reasons radial profiles for the above mentioned parameters in case of high pressure conditions are not shown in the present work

In case of subcooled boiling the agreement is good if the axial average values are considered. The main difficulties for the simulation are observed for flow in transition to saturated boiling flow with higher void fraction at the end of the subcooled boiling region.

.The enlargement of the experimental database for bubbly flow and subcooled boiling at high and low pressure with an adequate level of resolution is required for further development. Radial profile distributions of void fraction, gas and liquid velocities and liquid temperature will allow the comparison of calculation results with experimental data to assess and validate further models to enlarge the range of applicability of CFD codes in the field of the two-phase flow simulations.

NOMENCLATURE

Roman Letters

- *A* fraction of area
- C coefficient [-]

Specific heat at constant pressure $[kJ/kg \cdot K]$ c_p d diameter [m] Eo Eötvös number Eo_d modified Eötvös number Fcorrection factor f bubble release frequency [1/s] acceleration due to gravity $[m/s^2]$ g mass flux $[kg/(m^2 \cdot s)]$ G heat transfer coefficient $[W/(m^2 \cdot K)]$ h latent heat of vaporization/condensation [kJ/kg] h_{fg} normal vector п nucleation site density $[1/m^2]$ N_a Nu Nusselt number coefficient [-] р Р pressure [MPa] heat flux $[W/m^2]$ Q radius [m] r Re **Reynolds** Number Sc Schmidt number St Stanton number Т temperature [K] t time [s] Uvelocity [m/s] velocity [m/s] и thermodynamic quality [-] x y^+ non-dimensional distance from the wall [-]

Greek Letters

- α volume fraction [-]
- ρ density [kg/m³]
- σ surface tension [kg/s²]

Subscripts

- *b* bubble
- *c* continuous, convection
- *cd* momentum transfer due to the drag force

- *d* departure
- d dispersed
- D drag
- e evaporation
- g gas
- h horizontal
- in inlet
- L lift
- *l* liquid
- *max* maximum
- *p* particle
- q quenching
- *ref* reference
- *rel* relative
- s surface
- sub subcooling
- t turbulent
- *TD* turbulent dispersion
- w wall, waiting
- WL wall lubrication
- α generic phase indicator
- β generic phase indicator

ACKNOWLEDGEMENTS

Part of this research was supported by the "Plan Nacional de I + D + I", Project EXPERTISER ENE2007-68085-C02-02/01.

REFERENCES

[1] Kurul, N. and Podowski, M.Z., On the modeling of multidimensional effects in boiling channels, ANS Proc. 27th National Heat Transfer Conference, Minneapolis, MN. 1991

[2] Grace J.R., Clift R., Weber M.E., Bubbles, Drops and Particles, Academic Press, 1978.

[3] Tomiyama, A., Struggle with computational bubble dynamics, 3rd International Conference on Multiphase Flow, ICMF-2004, Lyon, France, June 8–12, 1998, pp. 1–18.

[4] Antal, S.P., Lahey, R.T., Flaherty, J.E. Analysis of Phase Distribution in Fully Developed Laminar Bubbly Two-Pahse Flow, Int. Journal of Multiphase Flow, Vol 17, 635-652, 1991

[5] Frank Th., Zwart P.J., Krepper E., Prasser H.-M., Lucas D., Validation of CFD models for mono- and polydisperse air–water two-phase flows in pipes, Nuclear Engineering and Design 238 (2008) 647–659

[6] M. Ishii, Thermo-fluid Dynamic Theory of Two-phase Flow, Eyrolles, Paris, 1975

[7] D.A. Drew and R.T. Lahey Jr., Application of general constitutive principles to the derivation of multidimensional two-phase flow equation, International Journal of Multiphase Flow **5**, 1979, pp. 243-264

[8] E. Krepper, Lucas D., Prasser H.-M., On the modelling of bubbly flow in vertical pipes, Nuclear Engineering and Design 235 (2005) 597–611

[9] Krepper E., Končar B., Egorov Y., CFD modelling of subcooled boiling—Concept, validation and application to fuel assembly design,Nuclear Engineering and Design 237 (2007) 716–731

[10] Lucas, D., Shi, J.-M., Krepper, E., Prasser, H.-M., Models for the forces acting on bubbles in comparison with experimental data for vertical pipe flow. In: 3rd International Symposium on Two-Phase Flow Modelling and Experimentation, Pisa, Italy, 2004.

[11] Burns, A.D.B., Frank, Th., Hamill, I., and Shi, J-M., Drag Model for Turbulent Dispersion in Eulerian Multi-Phase Flows, 5th International Conference on Multiphase Flow, ICMF-2004, Yokohama, Japan.

[12] Sato, Y. and Sekoguchi, K.,Liquid Velocity Distribution in Two-Phase Bubbly Flow, Int. J. Multiphase Flow, 2, p.79, 1975.

[13] Egorov, Y. and Menter, F., Experimental implementation of the RPI boiling model in CFX-5.6, Technical Report ANSYS / TR-04-10., 2004.

[14] Tolubinski, V. I. and Kostanchuk, D. M., Vapour bubbles growth rate and heat transfer intensity at subcooled water boiling, 4th. International Heat Transfer Conference, Paris, France, 1970.

[15] Cole, R., A photographic study of pool boiling in the region of CHF, AIChEJ, 6 pp. 533-542, 1960.

[16] Yeoh G.H., Cheung C.P., Tu J.Y., Ho K.M., Fundamental consideration of wall heat partition of vertical subcooled boiling flows, International Journal of Heat and Mass Transfer, Volume 51, Issues 15-16, 15 July 2008, Pages 3840-3853

[17] B. Končar, I. Kljenak, B. Mavko, Modeling of local two-phase parameters in upward subcooled flow boiling at low pressure, Int. J. Heat Mass Transfer 47 (2004) 1499–1513.

[18] Del Valle, V. H. and Kenning, D. B. R., Subcooled flow boiling at high heat flux, Int. J. Heat Mass Transfer, 28 p. 1907, 1985.

[19] Takashi Hibiki, Mamoru Ishii, Zheng Xiao, Axial interfacial area transport of vertical bubbly flows, International Journal of Heat and Mass Transfer, Volume 44, Issue 10, May 2001, Pages 1869-1888

[20] Bartolomej, G.G., Chanturiya, V.M., Experimental study of true void fraction when boiling subcooled water in vertical tubes, Thermal Engineering, vol. 14, pp. 123–128 1967

[21] Chen E., Li Y., Cheng X., Wang L., Modeling of low-pressure subcooled boiling flow of water via the homogeneous MUSIG approach, Nuclear Engineering and Design 239 (2009) 1733–1743

[22] Chen E., Li Y., Cheng X., CFD simulation of upward subcooled boiling flow of refrigerant-113 using the two-fluid model, Applied Thermal Engineering 29 (2009) 2508–2517

[23] Kurul, N., Podowski, M.Z., Multi-dimensional effects in sub-cooled boiling. In: Proc. 9th Heat Transfer Conference, Jerusalem, 1990