Copyright IPCO-2014
Vol.2
ISSN : 2356-5608

Numerical Investigation of the Heating Distribution Effect on the Boiling Flow in the Bubble Pumps

Raoudha Garma [#1], Youssef stiriba *[2],Mahmoud bourouis *[3],Ahmed Bellagi [#4]

[#]: *U.R .Thermique et thermodynamique des procédés industriels,*
Ecole Nationale d'ingénieur de Monastir, E.N.I.M, Av. Ibn Jazzar, 5060 monastir, Tunisie

*: *Center CREVER, Universitat Rovira i Virgili,*
43006 Tarragona, Spain

[1]garma_raoudha@yahoo.fr
[4]a.bellagi@enim.rnu.tn

***Abstract*— In this paper, numerical simulations of the heating repartition effect on the boiling flow in vertical tube was conducted with the commercial CFD (Computational Fluid Dynamics) package FLUENT. The Eulerian multiphase flow framework model was used to model the phase's interaction. User-Defined Functions (UDFs) are provided to compute the wall heat transfer and to calculate inter-phase heat and mass transfer. Total and partial heating are treated to approach to two different configurations of the bubble pump which considered as one of the most important components of the diffusion absorption machine.**

In the simulation results we present preliminary calculations with pure water as working fluid to compare void fraction distribution, liquid-vapor velocities arrangement and temperature profiles.

***Keywords*— Diffusion absorption refrigeration, Modelling, Simulation, Bubble pump, heating, boiling flow, void fraction**

I. Introduction

Based on multiple good features such as the absence of moving part which cause vibration and noise and specially the possibility to operate with waste heat and solar energy, the diffusion absorption refrigerator (DAR) invented by Platen and Munters [1] in the1920s has been experimentally and numerically investigated over the years. In these absorption systems, the generator is considered as one of the most important components and improving its performance will automatically contribute to the whole absorption system performance improvement.

Detailed thermodynamic models for three DARs generator configuration and bubble pump were developed and analyzed by computer simulation by A. Zohar et al. [2] to study their effect on the system performance. It was found that the configuration that integrated both the generator and the bubble pump is of great interest. A mathematical model for the forced convective boiling of refrigerant-absorbent mixtures in vertical tubular generator is developed by B. Pasupathy et al. [3]. The generator is made up of two stainless steel coaxial tubes of 1m length. A parametric analysis has also been performed to study the effect of various parameters on the performance of the generator. Two-fluid model was employed by Zheshu Ma et al. [4] to describe the two-phase flow and heat transfer processes in a two-phase closed thermosyphon. It was found that numerically predicted flow patterns and distribution of parameters under different conditions show a good agreement in the most part with experimental results. The some model was used by A. Bemhmiden et al. [5, 6] to investigate the influence of heat input on the uniformly heated bubble pump for different operating conditions. The optimum heat flux was correlated as a function of the tube diameter and mass flow rate, and the minimum heat flux required for pumping was correlated as a function of the tube diameter. The same bubble pump configuration (uniformly heated tube) was numerically investigated by R. Garma et al. [7] using the commercial CFD (Computational Fluid Dynamics) package FLUENT. It was found that the onset boiling point is reduced from 0.43 to 0.0016 m and the void fraction at tube's outlet is increased from 0.01 to 0.7 when increasing the wall heat input from 25 to 150 kW/m^2.

An experimental investigation of an air-cooled diffusion-absorption machine operating with a binary light hydrocarbon mixture (C_4H_{10}/C_9H_{20}) as working fluids and helium as pressure equalizing inert gas is presented by Ben Ezzine et al. [8]. A new concept of generator consisting in a separated boiler and bubble pump instead of the usual combined generator is tested. The bubble pump is a single vertical tube (height: 800 mm, diameter: 8/10 mm) to which the heat input is restricted to a small zone in the bottom. The experimental results show that the bubble pump exiting temperature as well as those of the major components of the machine but the absorber is very sensitive to the heat power inputs to the bubble pump. For bubble pump heat inputs from 170 to 350 W, the driving temperature varies in the range of 120-150 °C.

Experimental study and the theoretical thermodynamic simulation of the same absorption refrigerator prototype with methylamine-water-helium were carried out by Mazzouz Souha et al. [9]. The test results showed that cold at -5°C is effectively produced by the machine for heating temperature of about 105°C and an ambient air of 22°C. The comparison of the experimental data with the theoretical model showed a good agreement. Jakob et al. [10] reported that the indirectly heated generator with its bubble pump is the main new feature of a solar heat driven ammonia-water diffusion-absorption cooling machine and that all the prototypes constructed performed well.

II. DARs Working Principal

To explain the working principal of the diffusion absorption refrigerator, we refer to the simplified schematic of a DAR depicted in figure 1. The machine consists of a generator including a separated bubble pump and a boiler, a rectifier, a condenser, an evaporator, a gas heat exchanger (GHX), an absorber and a solution heat exchanger (SHX). The working fluids mixture is composed of ammonia as refrigerant, water as absorbent and helium as pressure equalizing inert gas. The strong solution coming from the receiver tank flows to the generator via a solution heat exchanger. The single-phase solution is heated at the bottom of the bubble pump causing some ammonia to evaporate and then small vapour bubbles begin to appear on the tube wall. As the solution is further heated continuously by the heat flux Q_{pb}, the vapour bubbles grow and merge forming gas plugs that push liquid slugs in the pump-tube up. In this way, the two-phase solution is lifted up to the gas–liquid separator. When the two-phase solution is further heated by the additional heat flux Q_B provided to the boiler, the solution gets further concentrated and supplement quantity of vapor is generated.The resulting ammonia poor solution returns back to the absorber through the SHX. The vapour leaving the bubble pump contains a small quantity of the solvent water which is removed by condensation in the rectifier. The purified ammonia vapour is then condensed in the air cooled condenser and then introduced in the evaporator. As this evaporator is charged with helium, the partial pressure of the liquid refrigerant decreases and it evaporates at low temperature producing cold. While proceeding to the evaporator exit, the ammonia continues to evaporate and its partial pressure rises. The resulting ammonia-helium gas mixture becomes heavier as the ammonia still evaporating and so it drops down in the receiver think located at the bottom of the air cooled absorber. There, the weak solution arriving from the absorber's top absorbs the refrigerant and as a result the vapour ammonia-helium mixture becomes more and more lighter and it rises up to the top. The rich solution resulting at the bottom flows down toward the generator.

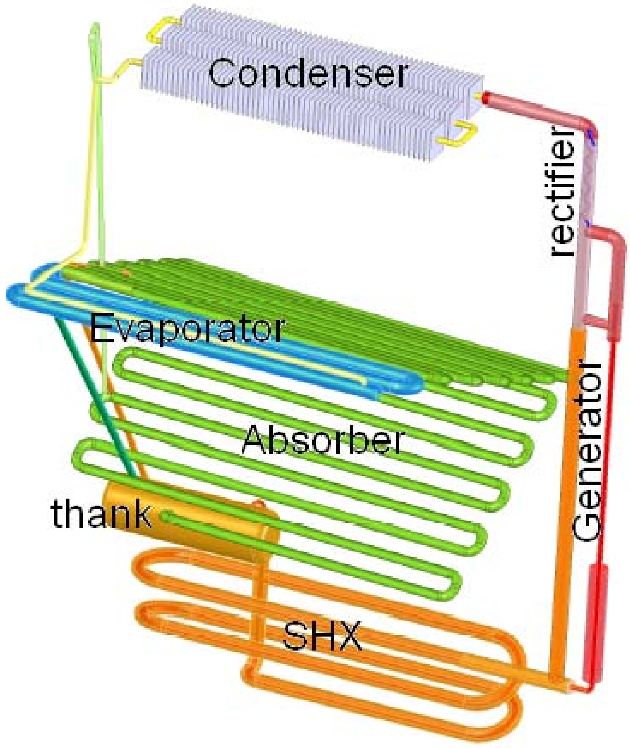

Fig.1 Schematic representation of a DARs

III. CFD Modelling and Resolution

The commercial CFD code FLUENT was used as a frame to perform the simulation. The interfacial forces models and the wall boiling model described previously were implemented in the code through User-Defined Functions (UDFs). The Eulerian multiphase flow framework model was used to model the phase's interaction. GAMBIT software was used to build and mesh a two-dimensional computational domain (2-D geometry) with 5x500 uniform rectangular cells. Only half of the pipe is considered due to the symmetry at the pipe axis. Unsteady state calculations with a time step of 0.1 s were fixed for all cases. SIMPLE algorithm was carried out for the calculations of the pressure velocity-coupling and the first order upwind calculation scheme was performed for the discretization of momentum, energy and volume fraction equations.

The saturation temperature is fixed to 425.15 K corresponding to the operating pressure (pressure outlet 15 bars). Fully-developed profile of velocity and sub-cooling temperature (T_{sub}=5K) is applied at the inlet. The gravity acceleration is 9.8 m/s2. Wall thickness is fixed to 2mm.

IV. Bubble Pump Heating Configurations

Total and partial heating are treated to demonstrate the heating repartition effect on the boiling flow in the bubble pumps. In the two cases the bubble pump consists of a single vertical tube of 1 m height and 10 mm outer diameter. The sub-cooled water (420.15 K) enters the system at the bottom, and then boils due to the heat fluxes supplied from the pipe wall. The same amount of heat flux, Q_{bp} = 2.827 kW, is applied to the stainless-steel made tube wall as follow:

- In the first case we have a uniformly heated bubble pump as demonstrated in Fig.2 and,
- In the second case the heat flux is supplied to 30 cm of the tube length (≈ 1/3 of the total tube's length) as indicated in fig.3.

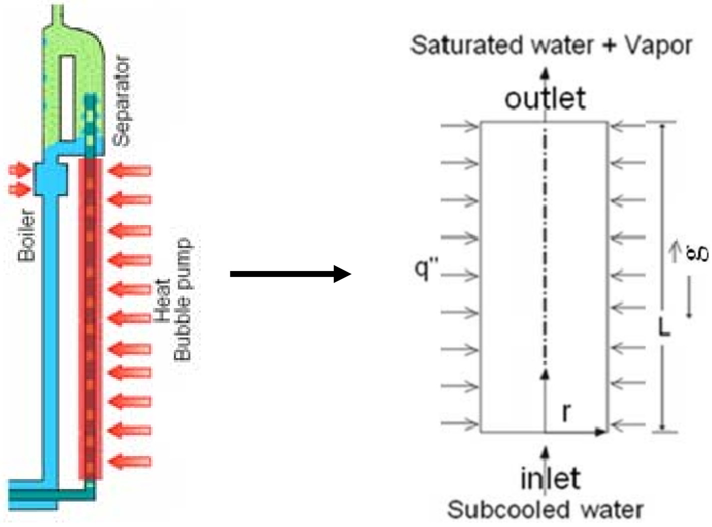

Fig.2. First bubble pump configuration: Uniformly heated bubble pump

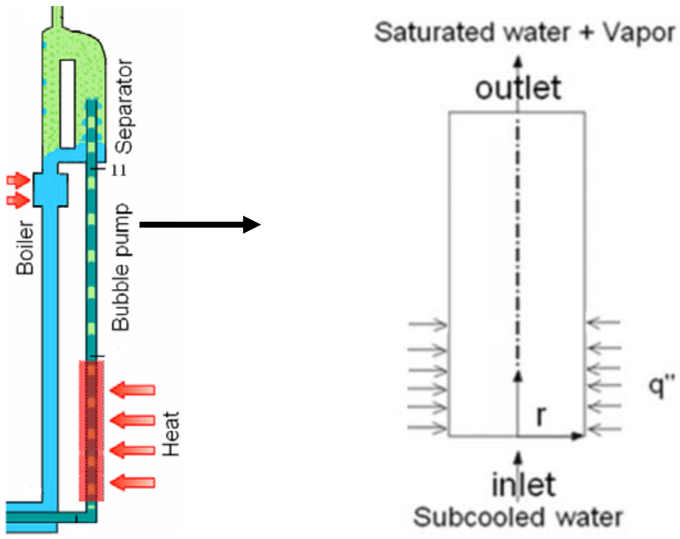

Fig.3. Second bubble pump configuration: Partially heated bubble pump

V. Results and Discussion

In the simulation results we present preliminary calculations with pure water as working fluid to compare void fraction distribution, liquid-vapor velocities arrangement and temperature profiles corresponding to heat input of Qbp = 2.827 kW .

Fig.4.a. illustrates the axial void fraction evolution for total and partial heating cases. One can see that the boiling onset in the partially heated tube is clearly lower than that for total heated one and this can be explained by the higher heat density in the bottom of the tube. One can also observe that for uniformly heated bubble pump the void fraction increases slightly over the whole length of the tube, attains a value of

0,1 when L=30 cm (L is the tube´s length) and a maximum of 0,55 at the tube outlet. This behavior is completely different from that of the partially heated pump. In fact, for this case the void fraction rises sharply, reaches a value in the order of 0,6 when L=30cm and then it remains approximately constant over the remaining length of the tube.

The bubble pump, consisting on a vertical heated tube, may present more than one flow regime. Identifying these flow regimes can be determined by referring to the limit values of the void fractions. The critical void fractions corresponding to the bubbly-slug, slug-churn and churn-annular transition are respectively 0.3, 0.55 and 0.8 [11, 12]. Fig. 4.b shows the limits of flow regimes from the curve of the void fraction evolution. One can see the effect of the heat distribution on the flow régime repartition along the tube.

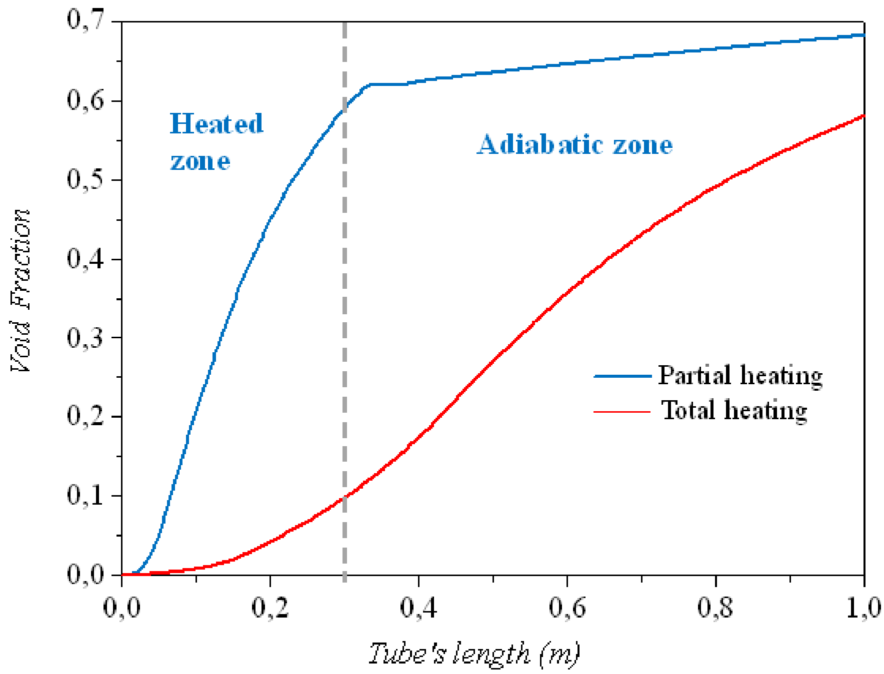

Fig.4.a. Axial void fraction evolution for total and partial heated tube

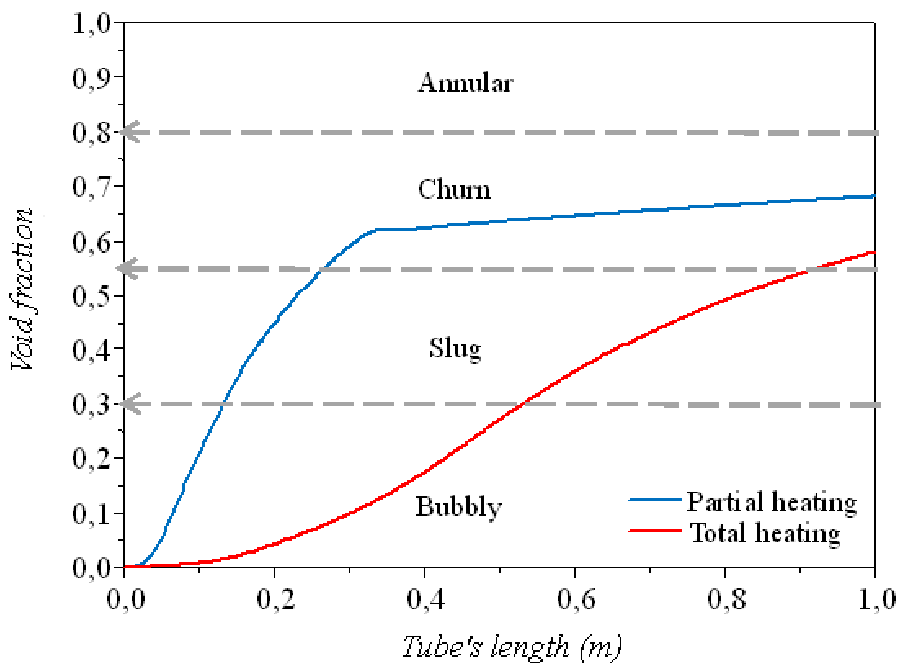

Fig.4.b. Flow pattern prediction

Radial temperature profiles at different tube locations for total and partial heating are respectively depicted in Figure 5.a and fig.5.b. One can observe for the first case (fig.5.a) that the liquid temperature profiles show the same behavior for the various locations. Actuality, the liquid at the center of the tube is subcooled even at the outlet (z = 1 m) and it is superheated near to the wall. We can see also that the subcooling level is decreasing throughout the tube's length. Referring to fig.5.b, we can remark the same temperature evolution

As the total heated tube until L=0.3 m (heated zone) except that the superheat near to the wall is clearly higher in this case. Whereas for L upper than 30 cm (the riser) the temperature remain constant (adiabatic zone)

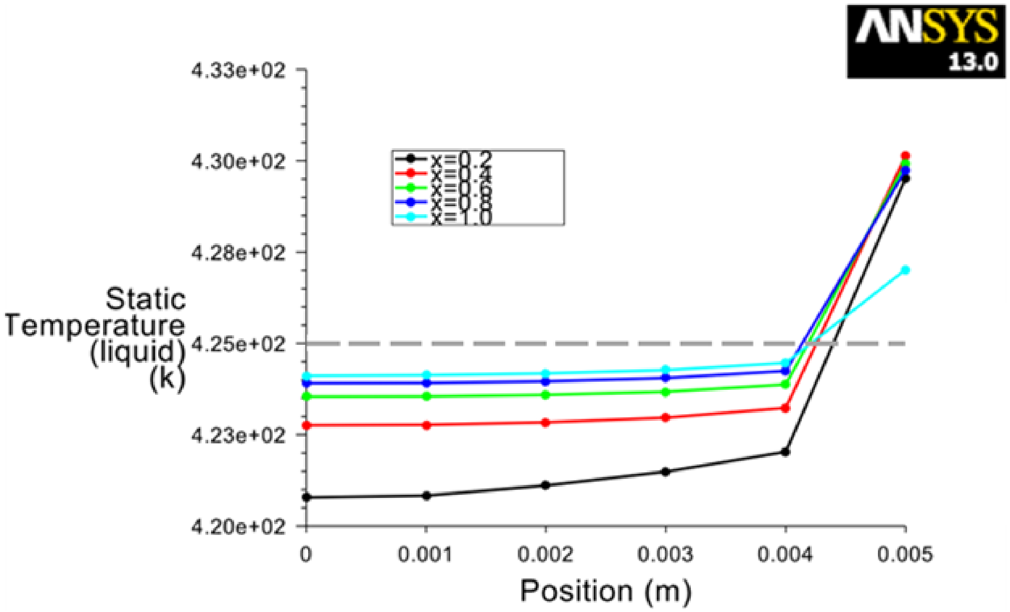

Fig. 5.a: Radial liquid temperature profile for total heating

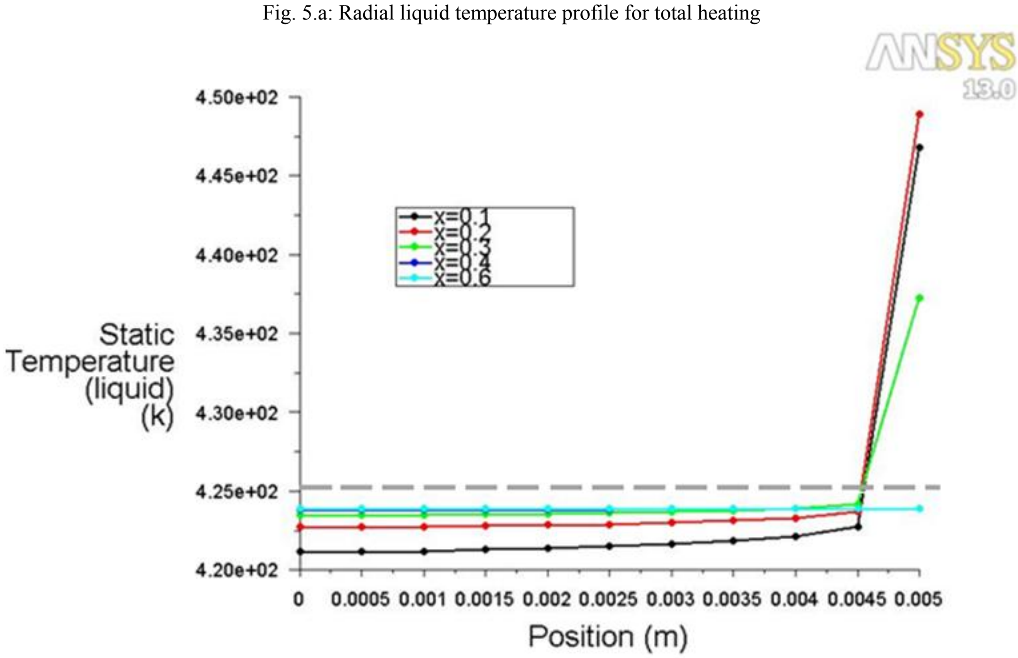

Fig. 5.b: Radial liquid temperature profile for partial heating

Liquid and vapor velocities arrangement for total and partial heating are respectively illustrated in Figure 6.a and fig.6.b. In the first case, one can remark that the liquid velocity increases slightly, it goes from 1.2 to 1.3 m/s from the entrance to 0.4 m, then increases sharply to the rest of the tube, it reached 2.9 m/s at the exit. The steam follows almost the same profile as the liquid. In the second case, the liquid-vapour velocity follows the same behaviour of the void fraction. In fact, it increases rapidly in the heated area then it increases slightly (almost constant) on the rest of the tube.

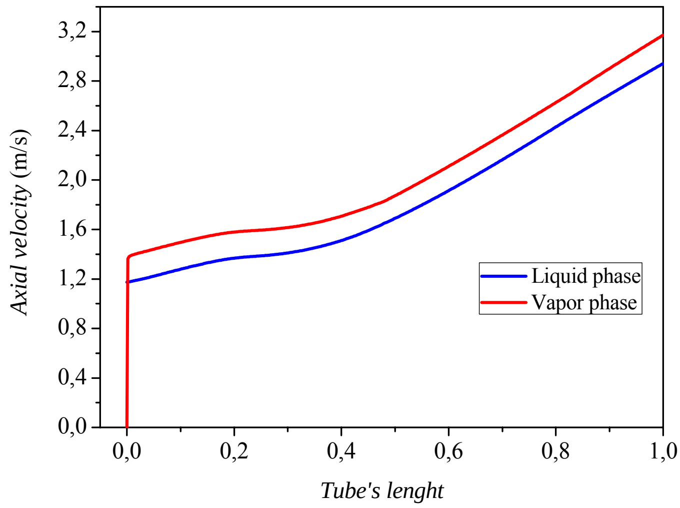

Fig. 6.a: Liquid and vapor velocities arrangement for total heating

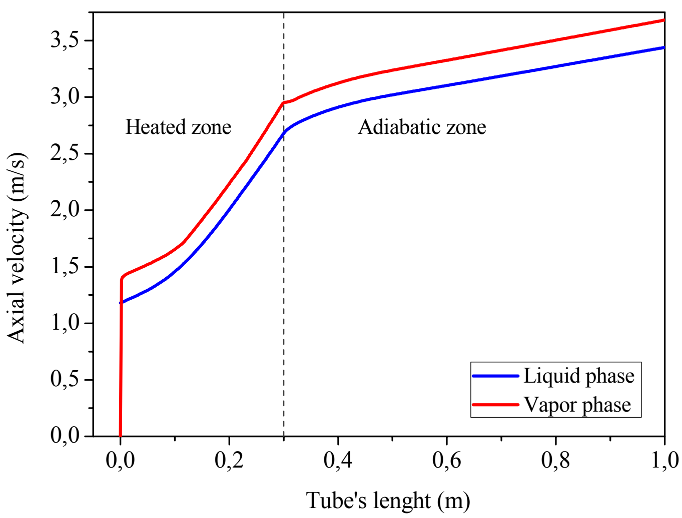

Fig. 6.b: Liquid and vapor velocities arrangement for partial heating

VI. Conclusions

Numerical simulation of the heating distribution effect on the boiling flow of water in vertical tube was carried out with the commercial CFD (Computational Fluid Dynamics) package FLUENT. User defined Functions (UDFs) are employed to model the boiling phenomena. Void fraction distributions, liquid-vapor velocities arrangement and radial temperature profiles are calculated and discussed. It was found that the the void fraction is higher when heating partially the wall. Flow regimes repartitions are identified referring to the void fraction variation along the tube. The liquid temperature in the middle part of the tube is still sub-cooled along the pipe and it is superheated near to the wall so bubbles migrate from the heated wall to the center of the tube and it still constant along the riser when the tube is partially heated.

References

[1] Von Platen BC, Munters CG. 1928. US Patent 1: 685-764.

[2] A. Zohar, M. Jelinek, A. Levy, I. Borde. 2008. The influence of the generator and bubble pump configuration on the performance of diffusion absorption refrigeration (DAR) system. International journal of refrigeration 31: 962–969

[3] Balamurugan PASUPATHY, Mani ANNAMALAI. 2010. Numerical Studies on Vertical Tubular Generator in Vapour Absorption Refrigeration System. International Refrigeration and Air Conditioning Conference at Purdue, July 12-15, 2010

[4] Zheshu Ma, Ali Turan, and Shengmin Guo. 2009. Practical Numerical Simulations of Two-Phase Flow and Heat Transfer Phenomena in a Thermosyphon for Design and Development. G. Allen et al. (Eds.): ICCS 2009, Part I, LNCS 5544, pp. 665–674, 2009.© Springer-Verlag Berlin Heidelberg 2009

[5] A. Benhmidene, B. Chaouachi, M. Bourouis, and S. Gabsi. 2011. Effect of Operating Conditions on the Performance of the Bubble Pump of Absorption-Diffusion Refrigeration Cycles. THERMAL SCIENCE, Vol. 15, No. 3, pp. 793-806

[6] A. Benhmidene, B. Chaouachi, M. Bourouis, and S. Gabsi. 2011. Modelling of heat flux received by a bubble pump of absorption-diffusion refrigeration cycles. Heat Mass Transfer. 47: 1341–1347

[7] R. Garma, Youssef stiriba, M. Bourouis, and A. Bellagi. 2012. Numerical Investigation of Nucleate Boiling Flow in Vertical Tubes, 2ème Colloque International sur l'Énergie CIE 12,Tozeur (Tunisie) Le 26-27-28 mars 2012

[8] N. Ben Ezzine, R. Garma, M. Bourouis, A. Bellagi. 2010. Experimental Studies on Bubble Pump Operated Diffusion Absorption Machine Based on Light Hydrocarbons for Solar Cooling, Renewable Energy, 35 (2010), 2, pp. 464-470

[9] S. Mazouz, N. Ben Ezzine, R. Garma, M. Bourouis, A. Bellagi. 2008. Experimental Investigation and Theoretical Model of a Diffusion Solar Absorption Machine International Sorption Heat Pump Conference, 23-26 September, 2008, Seoul, KOREA