Water Boiling Heat Transfer in Vertical Jacketed Pipe: A CFD Model

Mohammad Amin Abdollahi[1], Abtin Ataei[2], Mohammad Heydari[3]

***Abstract*—**In order to improve process performance and to prevent equipment damages for some special cases in industry, keeping temperature of a media unchanged is necessary. A possible approach to achieve this aim is using saturated water or steam as a jacket around pipes or tube bundles at a constant temperature. This leads the heat exchange to happen in constant temperature. In this study, a two-phase-Eulerian CFD modeling was carried out for a special jacketed pipe using ANSYS FLUENT software. Results of this modeling were compared with predictions of theoretical correlations which indicate its approval. Finally extended results were extracted and reviewed for further studies.

***Keywords*—** Jacketed Pipe, Boiling Heat Transfer, CFD, Critical Heat Flux.

I. Introduction

HEAT transfer of boiling is an important phenomenon which is widely used as a key solution for critical applications. BWR type light-water nuclear reactors, fixed-bed catalyst reactors coolant and Column Reboilers are some possible illustrations of this usage. Applications of boiling heat transfer in vertical thermosyphon reboilers, Fixed-bed Fischer-Tropsch reactors and shell-side thermosyphon reboilers are similar in some respects. Generally the heat flux (heat transferred per unit area) and heat transfer coefficients are as following.

Kettle < horizontal Thermosyphon < vertical Thermosyphon

So, it is not surprising that the most common design of reboiler is the vertical thermosyphon. [1]

When a vapor–liquid mixture flows through a circular tube, a number of different flow regimes can occur which depend on the vapor fraction, flow rate, and orientation of the tube. For vertical tubes, flow regimes could be categorized as [2].

- Bubbly flow: At low vapor fractions, vapor bubbles are dispersed in a continuous liquid phase.
- Slug flow: At moderate vapor fractions and relatively low flow rates, large bullet-shaped vapor bubbles flow through the tube separated by slugs of liquid in which smaller bubbles may be dispersed. A percolating coffee pot exemplifies this type of flow.
- Churn flow: At higher flow rates, the large vapor bubbles present in slug flow become unstable and break apart, resulting in an oscillatory, or churning, motion of the liquid upward and downward in the tube.
- Annular flow: At high vapor fractions and high flow rates, the liquid flows as a film along the tube wall while the vapor flows at a higher velocity in the central region of the tube. Small liquid droplets are usually entrained in the vapor phase, and vapor bubbles may be dispersed in the liquid film as well. At sufficiently high liquid flow rates, the droplets coalesce to form large streaks or wisps of liquid entrained in the vapor phase. This condition, which is characteristic of flows with a high mass flux, is referred to as wispy annular flow.
- Mist Flow: At very high vapor fractions, the liquid phase exists entirely as droplets entrained in a continuous vapor phase.

Generally, Two models of two-phase flow are used, the homogeneous flow model and the separated flow model. In the first one,, both phases are supposed to have an equal velocity, however, In the separated flow model, it is assumed that the phases flow in separate zones in annular and stratified regimes and so have different velocities, but with interaction capability to each other.

When the fluid enters the tube; by increasing the vapor fraction, the flow regime changes from bubbly to slug flow, and then to annular flow (the churn flow regime is omitted here) while the heat-transfer coefficient is continuously ascending. Eventually, the amount of liquid is reduced to the point where dry spots begin to appear on the tube wall, and the heat-transfer coefficient begins to decrease. This will continue until the tube wall is completely dry and all remaining liquid is in the form of droplets entered in the vapor phase (mist flow regime). The heat-transfer coefficient remains relatively constant as the droplets are gradually vaporized and the vapor becomes superheated. In the final stages, heat is transferred solely by gas-phase convection.

In vertical thermosyphon reboilers, the vapor fraction is kept sufficiently low that the mist flow regime does not occur and the sharp drop in heat-transfer coefficient is avoided. Nevertheless, the rate of heat transfer can vary greatly over the length of the reboiler, and an incremental analysis is therefore required to accurately predict performance. [2]

Many correlations have been developed to predict and formulize the convective heat transfer coefficient and so the

Mohammad Amin Abdollahi[1], Senior Student of Mechanical engineering, Department of Energy Engineering, Graduate School of the Environment and Energy, Science and Research Branch, Islamic Azad University, Tehran, Iran (corresponding author: e-mail: abdollahi_m@srbiau.ac.ir, tel: +989123431069).

Abtin Ataei[2], Associate Professor, Department of Energy Engineering, Islamic Azad University, Tehran, Iran (e-mail: a.ataei@srbiau.ac.ir).

Mohammad Heydari[3], Senior Student of Mechanical engineering, Department of Energy Engineering, Graduate School of the Environment and Energy, Science and Research Branch, Islamic Azad University, Tehran, Iran

heat flux in vertical tube-side and shell-side thermosyphon reboilers. The correlation developed by Chen [3] is the most widely used method for calculating heat-transfer coefficients in convective boiling. He assumed that the convective and nucleate boiling sections of heat transfer can be added to each other, however, it is known that convection tends to suppress nucleate boiling. Chen attributed this effect to the steepening of the temperature gradient near the wall with increasing flow rate, which reduces the effective temperature difference between the tube wall and bubbles growing outward of the wall. Therefore he introduced a suppression factor, SCH that is the ratio of the effective temperature difference for bubble growth to the overall temperature difference, Tw −Tsat. The convective coefficient, hL, can be calculated using any appropriate correlation for forced convection in pipes and ducts, such as the Seider–Tate equation. [2] However, Chen [3] used the Dittus–Boelter equation, a predecessor of the Seider–Tate equation for turbulent flow, as follows:

$$h_L = 0.023\left(\frac{K_L}{D_i}\right) Re_L^{0.8} Pr_L^{0.4} \qquad (1)$$

Chen used the Forster–Zuber correlation for the nucleate boiling heat-transfer coefficient which is usually stated by: [4]:

$$h_{nb} = 0.00122 \frac{k_L^{0.79} C_{pL}^{0.45} \rho_L^{0.45} g_c^{0.25} \Delta T_Q^{0.24} \Delta P_{sat}^{0.75}}{\sigma^{0.5} \mu_L^{0.29} \lambda^{0.24} \rho_V^{0.24}} \qquad (2)$$

Although the Chen correlation was developed using a limited amount of experimental data (for water, methanol, benzene, cyclohexane, pentane, and heptane, all in vertical tubes), it has a sound physical basis and reduces correctly in the limiting cases of zero flow rate (SCH =1), infinite flow rate (SCH →0), and zero vapor fraction [F (Xtt) =1.0], characteristics not always found in later correlations. [2].

In The correlation of Gungor and Winterton [5] the nucleate boiling and convection terms are additive like Chen's one. In this correlation a nucleate boiling suppression factor is included, along with a convective enhancement factor, EGW. It can be expressed as:

$$h_b = S_{GW} h_{nb} + E_{GW} h_L \qquad (3)$$

$$S_{GW} = (1 + 1.15\times 10^{-6} E_{GW}^2 Re_L^{1.17})^{-1} \qquad (4)$$

$$E_{GW} = 1 + 24000 Bo^{1.16} + 1.37 X_{tt}^{-0.86} \qquad (5)$$

The nucleate boiling heat-transfer coefficient is calculated by use of the Cooper correlation, and the Dittus–Boelter equation is used for the forced convection coefficient. [2].

The correlation developed by Liu and Winterton [6] is described by the following equations:

$$h_b = [(S_{LW} h_{nb})^2 + (E_{LW} h_L)^2]^{1/2} \qquad (6)$$

$$S_{LW} = (1 + 0.055 E_{LW}^{0.1} Re_L^{0.16})^{-1} \qquad (7)$$

$$E_{LW} = [1 + x Pr_L (\frac{\rho_L - \rho_V}{\rho_V})]^{0.35} \qquad (8)$$

More complex correlations have been also developed by Shah [7], Kandlikar [8], Steiner and Taborek [9], and Kattan et al. [10]. [2].

Reboilers are designed to operate below the peak flux, as beyond it either the heat flux would be lower, or much higher temperate difference would be required. Design is normally restricted to have a heat flux less than 70% of the critical flux. [1].

Mostinski [11] gave a correlation for the estimation of the critical heat flux for single tubes as follows:

$$q_{cr} = 3.76\times 10^4 P_c \left(\frac{P}{P_c}\right)^{0.35} \left[1 - \frac{P}{P_c}\right]^{0.9} \qquad (9)$$

Other correlations are also introduced by Palen [12], Katto and Ohno [13].

II. Material And Methods

In this paper, a model for a jacketed pipe is developed. Figure 1 depicts a schematic of the pipe. It contains a high temperature inner tube and an annulus which saturated water flows in it. To analyze heat transfer, the CFD method is used and then it is compared to the theoretical correlations.

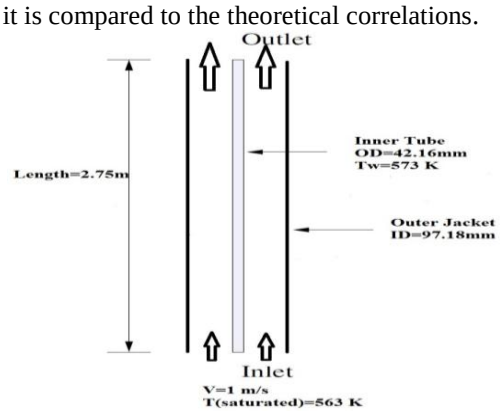

Fig. 1 Schematic Diagram of the model

Properties of saturated water/steam are obtained from material measurement laboratory of NIST (National Institute of Standards and Technology) and reported in table 1. [14]:

Table I
Saturated Water/Steam Properties At 563 K

	Liquid Phase	Vapor Phase
Pressure (Mpa)	7.455	
Density (kg/m^3)	732.19	39.05
Enthalpy (KJ/kg)	1289.2	2766.9
C_p (J/kg.K)	5.4897	5.5736
Viscosity (cP)	0.08971	0.019147
Thermal Conductivity (W/m^2.K)	0.56521	0.064641
Surface tension (N/m)	0.01669	

A 3D full model is created considering the fluid flow inside the annulus of jacketed pipe. It is meshed with 4160 quadrilateral concurrent cells. The set-out is made according to ANSYS FLUENT tutorial [15] with below exceptions:

- The geometry/mesh from pipe 2D arrangement is changed to jacketed pipe 3D arrangement.
- The heat flux to the fluid is omitted, instead the temperature of inner tube is considered to be constant and equal to 573 K.
- It is considered that the fluid is saturated at entrance of the jacket and will leave it saturated.

Considering different values for x (vapor fraction) and by using Chen's relevant heat transfer coefficient, the void

fraction and equivalent length of jacketed pipe are calculated.

a) By curve fitting for scatters of jacketed pipe length against vapor fraction, an implicit correlation is derived with a 5th degree polynomial curve. (y = 37500x5 - 29394x4 + 6854.2x3 - 420.58x2 + 48.771x + 0.0217, R2=1), by this correlation, the relevant vapor quality at the end of the pipe (length = 2.75m) is observed as 6.36%.

b) By curve fitting for scatters of heat transfer coefficient against jacketed pipe length, an implicit correlation is derived with a 3rd degree polynomial curve. (y = 643.75x3 - 3599.8x2 - 9575.9x + 93047, R2=1), Table 2 indicates the new data obtained by this correlation.

c) By curve fitting for scatters of void fraction against jacketed pipe length, an implicit correlation is derived with a 6th degree polynomial curve. (y = -1E-06x6 + 7E-05x5 - 0.0016x4 + 0.0178x3 - 0.1136x2 + 0.4154x - 0.0126, R2=1).

TABLE II
MODIFIED NEW DATA BY CURVE FITTING

Jacketed pipe length (m)	Heat transfer coefficient($W/m^2.^0k$)	Heat flux (W/m^2)	Void fraction
0.1-2.7	92054-53620	920540-536200	0.0278-0.5558

The resulted curves for void fraction and heat flux against length are illustrated by Figures 2 and 3.

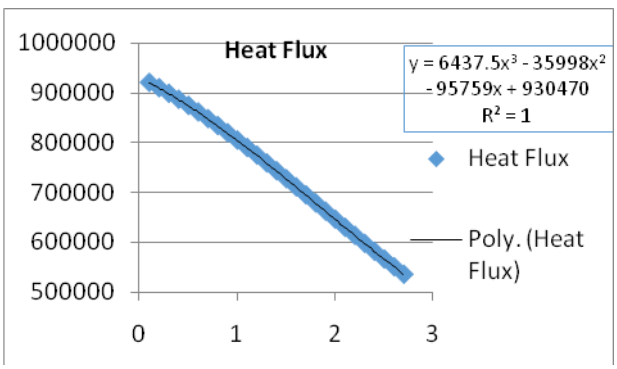

Fig 2 Modified new data for Heat Flux by curve fitting

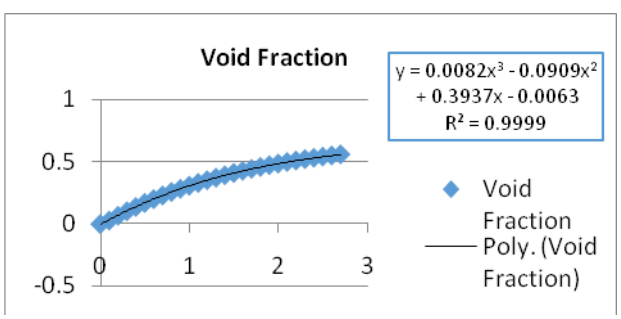

Fig. 3 Modified new data for void fraction by curve fitting

It is also necessary to calculate the critical heat flux Using Mostinski's correlation. By assuming the critical pressure as 22.064 MPa, the critical heat flux will be 3820000 W/m2.

III. RESULTS

By comparing the results of the CFD method with the ones of theoretical correlations, the plots of FLUENT for Heat Flux and Void Fraction are obtained and illustrated by Figures 4 and 5. As it could be seen there is a good consistency between the results of CFD and theoretical correlations. Also due to the concept of CFD method, seems that the data reported by this modeling could be more reliable and closer to what happens in actual situation.

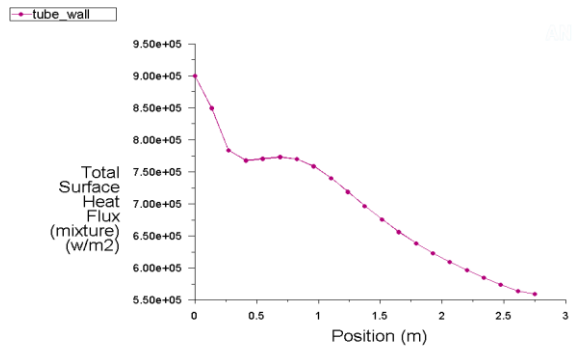

Fig 4 Results of Heat Flux against pipe length

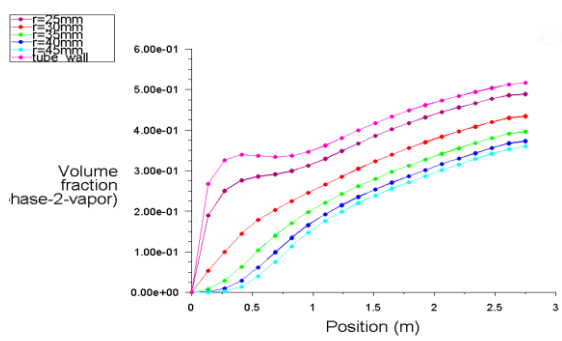

Fig 5 Results of Void Fraction against pipe length

For reviewing other results of the CFD modeling, Mass transfer rate from liquid phase to vapor phase is indicated by Figure 6 and. Figure 7 shows the rate of pressure loss through the jacketed pipe and Figure 8 shows the changes in static pressure through it.

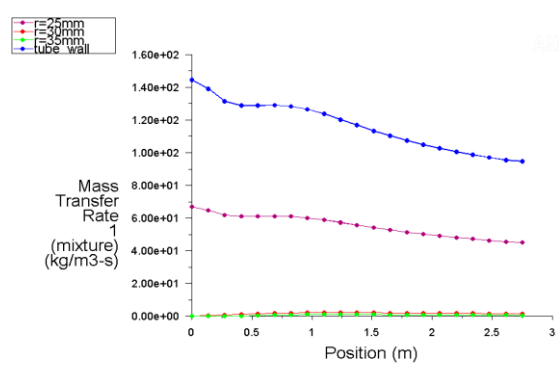

Fig 6 Results of mass transfer rate against pipe length

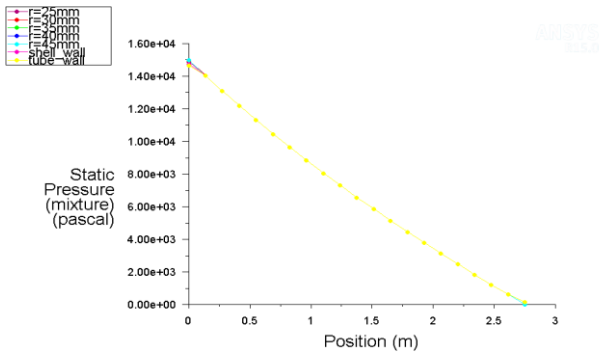

Fig 8 Results of static pressure against pipe length

The liquid velocity profile at outlet of the pipe is illustrated in Figure 9 as contours and vectors. As it is shown, there is a reasonable change in the velocity, since, the fractions of the liquid are transferred to vapor phase and therefore increased the velocity of the liquid.

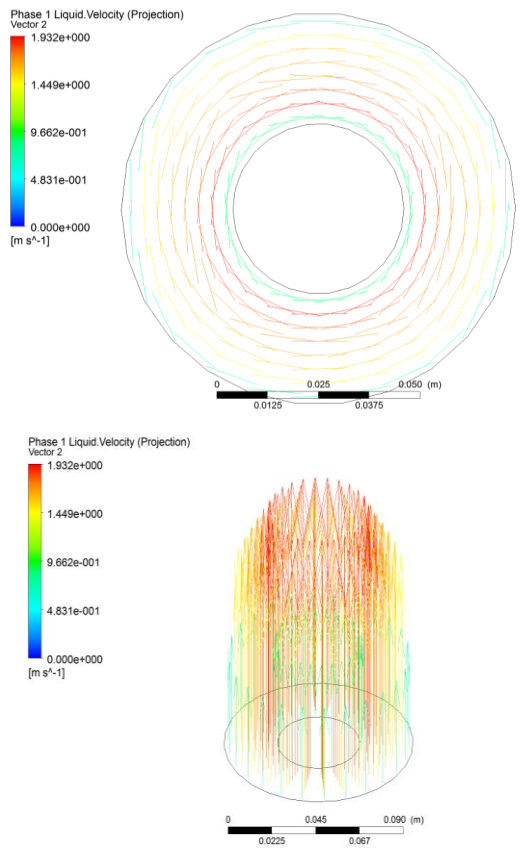

Fig 9 Contour and Vectors of Velocity at Outlet of jacketed pipe

IV. Conclusion

The CFD model developed in this article, presents reasonable results regarding to compatibility of general trends of heat flux and void fraction with predictions of theoretical correlations. It is notable that theoretical correlations just produce average figures irrespective of the radial distance in annulus and flow velocity curve generated from hydraulic effects. The capability of predicting hydraulic and heat transfer behavior of the fluid in all directions and in all sections of the geometry of the problem makes this CFD method a powerful tool for design and simulation. Although the data produced by CFD method are comparable to theoretical correlations (which are based on experimental data too), but necessity of validating the results of the CFD method with at least simpler experimental data, seems obvious.

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