THERMAL ANALYSIS OF 1D TWO PHASE FLOW IN A HORIZONTAL PIPE USING THE TWO-FLUID MODEL

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Abstract. *In this work, a non-isothermal intermittent air-water two phase flow in a circular pipe is studied. A 1D slug-capturing code based on the two-fluid formulation was extended to include thermal effects. Simulation results are compared with experimental data and the impact of the boundary conditions in the results are presented. An evaluation of the pipe length upstream of the heat transfer test section and the introduction of a white noise signal in the thermal and hydrodynamic parameters are conducted. The predicted pressure and temperature drop presented a reasonable agreement with the experimental data. Furthermore, it was shown that it is possible to use a white noise signal to shorten the required pipe length, thus reducing the computational domain and reducing the computational power required.*

Keywords*: thermal analysis, 1D, horizontal pipe, two-fluid model*

1. INTRODUCTION

The study of multiphase flow is of particular interest to the offshore oil industry. This phenomenon normally occurs on well streams, and has considerable implications in the production system design and operational strategy – especially in deep waters or long tie-back scenarios, where flow assurance issues are critical. The challenge with multiphase flow is the highly complex interaction between phases. The flow has a highly transient, turbulent and three-dimensional nature, which makes it difficult to model and to perform representative measurements.

The Two-Fluid Model (Ishii, 1975) represents one of the most used approaches for the simulation of multiphase flows. It is based on the solution of conservation equations of mass, momentum and energy for each phase. As mentioned above, even though multiphase flow has a three-dimensional nature, the behavior of the flow along a pipeline (usually with a length of several kilometers) is the main concern in practical engineering applications. Furthermore, the costs of 3D (or even 2D) simulations normally exceed by far present computational power. Therefore, one-dimensional approach is often employed.

Issa and Kempf (2003) have shown that the solution of the transient one-dimensional Two-Fluid Model equations on a high resolution mesh is capable of accurately predicting the evolution of long waves at the gas-liquid interface and the transition to slug flow is a natural outcome of the simulations. The methodology has been named "Slug-Capturing", in contrast to other approaches such as "Slug-Tracking" or "Unit-Cell Model", which require to some extent the incorporation of empirical correlations describing characteristics of the slug pattern. Cazarez-Candia et al. (2011), for example, presented a non-isothermal Two-Fluid Model for slug flow using the Unit Cell concept. However, to the knowledge of the authors, non-isothermal Slug-Capturing simulations have not been reported so far in the literature.

In this article, a non-isothermal 1D slug-capturing code based on the two-fluid model is presented, and simulation results are compared with experimental data. In addition, the effects of the pipe length and the introduction of a white noise signal in the thermal and hydrodynamic parameters were investigated.

2. MATHEMATICAL MODEL

In the present work, the Two Fluid Model equations were solved based on the following hypothesis: (i) horizontal or nearly horizontal pipelines; (ii) No mass transfer between phases; (iii) Constant density for the liquid phase; (iv) Ideal gas law for gas phase; (v) No differential pressure between phases; (vi) Negligible axial viscous diffusion fluxes (vii) Negligible axial heat flux; (ix) Constant specific heat and thermal conductivity for both phases. Considering these assumptions, the conservation equations of mass, momentum and energy are:

$$\frac{\partial(\alpha_k \rho_k)}{\partial t} + \frac{\partial(\alpha_k \rho_k U_k)}{\partial x} = 0 \tag{1}$$

$$\frac{\partial(\alpha_k \rho_k U_k)}{\partial t} + \frac{\partial(\alpha_k \rho_k U_k^2)}{\partial x} = -\alpha_k \frac{\partial P}{\partial x} - \alpha_k \rho_k g \frac{\partial h_L}{\partial x} \cos\beta - \alpha_k \rho_k g \sin\beta - \tau_{wk} \frac{S_k}{A} \pm \tau_i \frac{S_i}{A} \tag{2}$$

$$\frac{\partial(\alpha_k \rho_k H_k)}{\partial t} + \frac{\partial(\alpha_k \rho_k H_k U_k)}{\partial x} = \alpha_k \frac{DP}{Dt} + \frac{S_k}{A} q''_{wk} \pm \frac{S_i}{A} q''_i - \tau_{wk} \frac{S_k}{A} U_k \tag{3}$$

where U_k is the velocity, α_k is the void fraction, ρ_k is the density and H_k is the specific enthalpy. The subscript k indicates the phase. P is pressure, A is the cross sectional area of the pipe, β is the pipe inclination. τ_{wk} is the shear stress of phase k with the wall, τ_i is the gas-liquid interfacial shear stress, S_k is the phase k wet perimeter, S_i is the interface perimeter, h_L is the local liquid height, g is gravity, q''_{wk} is the wall heat flux of phase k, q''_i is the interfacial heat flux.

The closure equations are:

$$\tau_{wk} = \frac{f_k \rho_k |U_k| U_k}{2} \quad \tau_i = \frac{f_i \rho_G |U_G - U_L| (U_G - U_L)}{2}, \; q''_{wk} = ht_{k-eff}(T_k - T_\infty), \; q''_i = ht_i |T_G - T_L| \tag{4}$$

$$\frac{1}{ht_{k-eff}} = \frac{1}{ht_k} + \frac{\ln(r_e/r_i) r_i}{K} + \frac{(r_e/r_i)}{ht_{ext}} \quad ; \quad H_k - H_{k-ref} = cp_k (T_k - T_{ref}) \tag{5}$$

where f_k is the friction factor for phase k and f_i is the friction factor for the interface. Both coefficients are function of Reynolds number (Carneiro et al. 2011). The local heat transfer coefficient of phase k, ht_k, and the interface heat transfer coefficient, ht_i, were determined with Dittus & Boelter heat transfer correlation. r_i and r_e are the internal and external radius, K is the thermal conductivity of the pipe wall, ht_{ext} is the external heat transfer coefficient and cp_k is the specific heat at constant pressure of phase k.

The numerical solution was performed using the finite-volume method (Patankar, 1980), using the totally implicit scheme for time integration and upwind scheme for the convective terms. The conservation equations are solved sequentially and the system of algebraic equations for each variable is solved with the TDMA algorithm. At the present work, the discretized energy equations for each phase were implemented in an existing isothermal code (Ortega e Nieckele, 2005), for which extensive isothermal simulations were performed and validated with laboratory results (Carneiro et al. 2011).

3. RESULTS

Lima (2009) performed various measurements for intermittent air-water flow regime in the elongated bubble flow region, with the gas superficial velocity in the range of 0.22~0.8 m/s and the liquid superficial velocity in the range of 0.58~1.38 m/s. A horizontal pipe was employed, with a tubular concentric heat exchanger of 6.071 m length and 0.052 m of diameter, preceded by an insulated upstream pipe of 17.740 m length. The multiphase fluid was cooled down in the heat exchanger by a cold water stream. Figure 1 depicts the pipe configuration, with L_{up} as the insulated upstream pipe length.

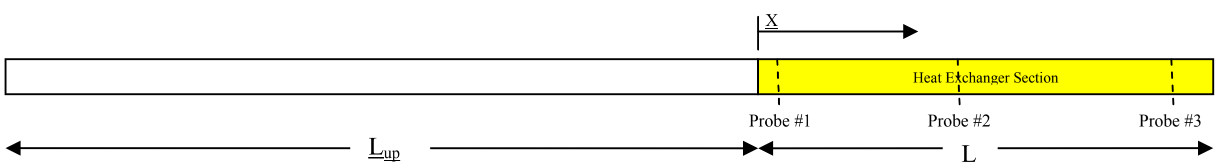


Figure 1. Pipe Configuration

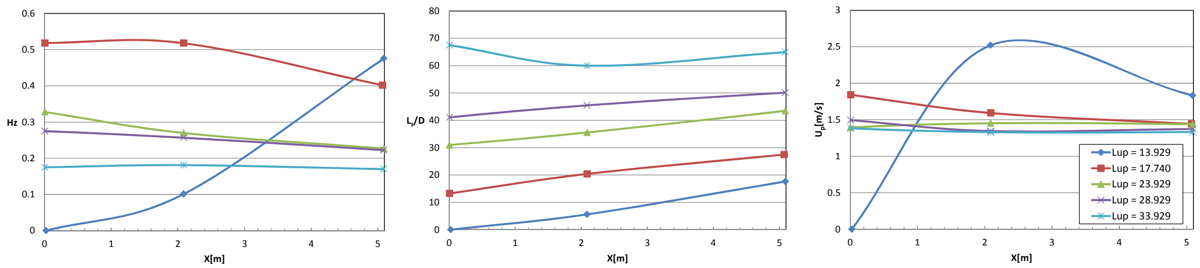
At the present work, one case was selected to be investigated (gas mass flow of 0.0007 kg/s and liquid mass flow of 1.47 kg/s). The outlet pressure was estimated from the experiment data and it was kept constant, equal to 135.15 kPa. To guarantee a developed slug regime at the entrance of the heat exchanger test section, several simulations were conducted to evaluate the impacts of the numerical boundary conditions. The parameters evaluated are: (i) Pipe Length and (ii) Effects of a white noise signal in the pipe entrance void fraction.

4.1 Pipe Length Evaluation

The stratified flow is the starting point for attaining the slug regime. As a consequence, the slug formation process consumes a significant length of the pipe entrance, and thus the flow may not achieve the heat exchanger section fully developed. In this sense, various simulations were performed, varying the upstream insulated pipe length, to investigate the impact in the hydraulic and thermal results.

Although the lengths were varied, all simulations used a fixed Δx/D of 0.385. In addition, all simulations were performed using the equilibrium liquid hold up at the pipe entrance.

Figure 2 shows the results for the slug parameters (frequency, length and velocity), measured along the heat exchanger section for different lengths of the upstream section, L_{up}. It is possible to observe in Fig. 2c, that the slug velocity stabilizes around 1.5 m/s for L_{up} greater than 23.929 m. As observed in Fig. 2a, the slug frequency decreases as the pipe lengths increases. This behavior was also found by Al-Safran (2009) for cases with high liquid velocities and high liquid holdup, as in the present case. It can also be seen that the slug length increases with the pipe length (Fig. 2b). This behavior was reported by many authors in the literature (Carneiro et al. 2011). The reported causes for the growing slug length are the merging process and the wake effect. Although slug frequency and length did not stabilize to a specific value, their variation along the heat exchanger section decreased with larger L_{up}.



(a) slug frequency (b) slug length (plug) (c) slug velocity (plug)

Figure 2. Simulation results for the slug parameters.

Table 1 shows the position of the initial slug formation along the pipeline, as well as the temperature and pressure drop long the test section, for several pipe lengths. For all cases the first slug appeared at 16 m from the pipe entrance. As a result, for the 20 m pipe simulation, slugs started within the heat exchanger section, which affected the quality of the results. This result agrees with the experiment, since a longer section was needed before the heated test section to guarantee the presence of developed slugs. The upstream length section did not influence significantly the predicted temperature drop, with a difference of approximately 20% for all cases in relation to the experimental data. On the other hand, the pressure drop in the heat exchanger section seems to be influenced by L_{up}, what can be caused by the changes in the slug parameters (growth of the slug length and decrease of the slug frequency with bigger L_{up}).

Table 1. Effects of Pipe Length

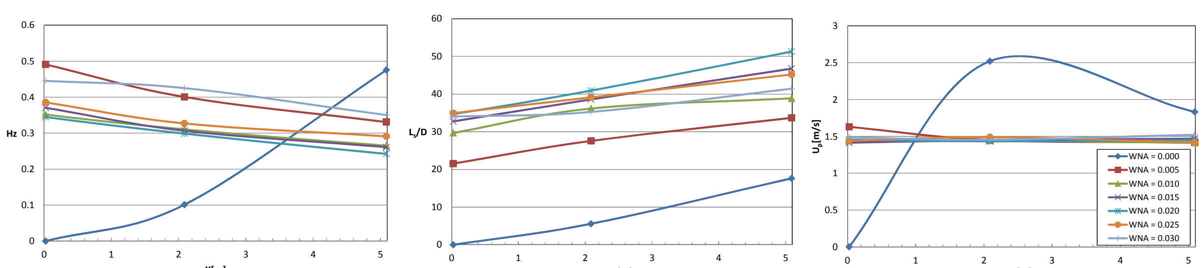
L+ L_{up} [m]	**L_{up} [m]**	**Slug Formation Position [m]**	**Error in ΔT [%]**	**Error in ΔP [%]**
20.00	13.93	2.07	-22.60	7.57
23.81	17.74	-1.74	-21.97	18.53
30.00	23.93	-7.93	-18.91	23.46
35.00	28.93	-12.93	-19.69	23.70
40.00	33.93	-17.93	-19.42	29.37

Note: Measurements taken at the heat exchanger section after flow stabilization.

4.2 Effects of white noise signal

The addition of a white noise signal is one possible way for shortening the slug formation length. The biggest benefit of this approach is to reduce the computational domain, thus reducing the computational power required. Furthermore, variations in the inlet boundary conditions are likely to happen in real applications (e.g. flow variations caused by pumps, gas lift compressors, etc…), thus making the noise addition fairly realistic.

In these simulations, the white noise was inserted in the void fraction up at the pipe entrance, though maintaining a constant mass flow. Figure 3 shows the results for various white noise maximum amplitude in a 20 m pipe, with a heat exchanger section of 6.071 m (the same as the experiment) and with Δx/D of 0.385. Note that the influence of the white noise in the slug parameters is similar to that of increasing the pipe length.



(a) slug frequency (b) slug length (plug) (c) slug velocity (plug)

Figure 3. - Simulation results for the slug parameters.

Table 2 presents the influence of the white noise in the flow parameters. Comparing Figs. 2 and 3, with Tables 2 and 3, it can be seen that the slug parameters for L_{up} = 23.93 m are quite similar to the results for L_{up} = 13.93 with the addition of white noise signal with maximum amplitude (WNA) of 0.01. This noise amplitude corresponds to 5% of the mean gas void fraction in the heat exchanger section. In addition, both simulations have similar slug formation positions and percentage errors on ΔP and ΔT. On the other hand, larger WNA values did not present correspondence with bigger pipe length cases, indicating that large WNA must be avoided. In fact, WNA values above 0.015 are over 10% of the mean gas void fraction, and thus excessive noise is present. Therefore, noise levels above 10% of the mean gas void fraction should be avoided.

Table 2. Effects of White Noise Amplitude in the Slug Formation

Maximum White Noise Amplitude (WNA)	**Slug Formation Position [m]**	**Mean Gas Void Fraction**	**Error in ΔT [%]**	**Error in ΔP [%]**
0.000	2.071	0.122	-22.60%	7.57%
0.005	-2.929	0.167	-21.48%	20.92%
0.010	-4.929	0.170	-20.95%	25.94%
0.015	-6.929	0.164	-21.41%	23.77%
0.020	-7.929	0.157	-21.68%	28.59%
0.025	-8.429	0.142	-22.74%	25.79%
0.030	-8.929	0.127	-23.70%	26.74%

Note: measurements taken at the heat exchanger section after flow stabilization

5. CONCLUSION

A non-isothermal air-water two phase flow in a circular pipe was studied with a 1D slug-capturing code based on the two-fluid model. The influence of the upstream pipe length and the levels of the white noise signal at the inlet were investigated and found to affect the overall hydrodynamic behavior significantly, while the error levels on the thermal solution were less influenced. For the upstream pipe length, care should be taken to avoid small lengths, assuring that the slugs are formed before the heat exchanger section. For the white noise amplitude, care should be taken to avoid excessive noise levels, which could distort the results.

It was also shown that is possible to shorten the simulated pipe length by adding a white noise signal. For a white noise signal with maximum amplitude of 0.01, similar results were obtained when compared to a simulation using a larger unheated upstream section (10 m longer). Based on the results, noise levels above 10% of the mean gas void fraction are not recommended.

Future work should be directed, for example, to the investigation of different flow conditions and the influence of different correlations for the heat transfer coefficients.

6. ACKNOWLEDGEMENTS

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8. RESPONSIBILITY NOTICE

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