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HEAT TRANSFER INFLUENCE ANALYSIS ON LIQUID-GAS HORIZONTAL SLUG FLOW HYDRODYNAMICS USING A SLUG-TRACKING MODEL

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Abstract. *Gas-liquid two-phase slug flows in pipes occur over a wide range of gas and liquid flow rates. This kind of flow pattern is frequently found in oil and gas offshore transportation, concomitantly with transfer of heat between the transported fluids and the subsea waters. An accurate prediction of the heat transfer is essential in the design of production facilities and operation, as well as in flow assurance assessments due to the possibility of wax and hydrates formation and deposition. Also, the gas expansion or contraction due to temperature and pressure gradients affects some slug flow parameters such as the elongated bubble length and its translational velocity. It might also influence the slip velocity between the phases, whose role in liquid mass transfer across the slug flow structures is relevant and tends to decrease the liquid slug length along the pipeline. From this perspective, the present work investigates the influence of the heat transfer in the hydrodynamic parameters of this type of flow by using a transient slug tracking model. The predicted lengths and velocities of each slug flow structure under heating, cooling, isothermal conditions and different gas and liquid flow rates were obtained and analyzed.*

Keywords: *slug flow, slug-tracking, heat transfer.*

1. INTRODUCTION

Two-phase slug flows are quite common to the petroleum and nuclear industries alike, occurring inside heat exchangers, boilers, condensers and in pipelines transporting crude oil and natural gas mixtures. This kind of flow is typified by the alternate passage of two distinct flow structures: a liquid slug that may or may not contain dispersed bubbles in its body and an elongated bubble travelling over a liquid film. Seen as a whole, those two structures form that what is known a *unit cell* (Wallis, 1969). Either one of those two structures possesses lengths and phase fractions of intermittent characteristics, thus making their analysis and modelling complex and laborious. However, predicting the behaviour of their pressure and temperature distributions, as well as their velocity fields and phase distribution is of extreme importance to the optimisation of both the production operations and the design of the pieces of equipment involved in such operations.

The earlier models aimed at predicting the characteristic parameters of such flows did not take the intermittency into account, that is, elongated bubbles and liquid slugs would, according to those models, repeat successively in both time and space. This sort of approach was used by Dukler and Hubbard (1975) to model horizontal two-phase slug flows, whereas Fernandes *et al.* (1983) used it to model vertically upward flows. Taitel and Barnea (1990) also proposed a model for any pipe inclinations. Those models are known as stationary models.

The advances in digital computing encouraged the development of new models of higher complexity. This new generation of models would take the intermittent features of those slug flows into consideration. Amongst those models, that one known as slug tracking stands out (Nydal and Banerjee, 1995; Grenier, 1997; Taitel and Barnea, 2000; Franklin and Rosa, 2004; Ujang *et al.*, 2006; Rodrigues, 2009). This kind of modelling uses mass and momentum conservation equations, applying them to control volumes whose boundaries coincide with those of the elongated bubble and the liquid slug.

The models so far mentioned consider the flow as isothermal. However, the heat transfer must not be disregarded in most industrial applications, as it is the case of the extraction and transportation liquid and gas mixtures from oil wells, be those either offshore or onshore. Hot fluids from the reservoir exchange heat with the surroundings (e.g. the subsea waters) through the pipe wall on their way to a production platform or storage facility. The heat flow thus changes the properties of the phases, in special their viscosities and densities, which by their turn affect several other hydrodynamic parameters such as the pressure drops, liquid slug and elongated bubble lengths and the velocities of those structures. Temperature and pressure distribution along the two-phase mixture might also favour hydrate formation and deposition, thus increasing pressure losses or even blocking long stretches of the pipe. That demonstrates the importance of accurately predicting temperature distributions along a pipeline to the optimum operation of oil and gas production facilities.

Most studies on heat transfer in two-phase gas-liquid flows focus on the determination of the overall heat transfer coefficient. Deshpande *et al.* (1991) observed that the two-phase heat transfer coefficient in horizontal pipes varies along their perimeter due to the distribution of the phases. They claimed that the heat transfer coefficient depends mainly on the liquid superficial velocity, the slug frequency and the liquid slug length. Kim and Ghajar (2006) proposed a correlation for the heat transfer coefficient as a function of the ratio between phases' properties such as the viscosity and the Prandtl number, and validated their correlation with a large data bank of experimental data. As they also claimed that the heat transfer coefficient strongly depends upon the liquid holdup they, suggested a factor to account for the flow pattern in terms of the phase fraction. França *et al.* (2008) proposed a mechanistic correlation for the heat transfer coefficient from the characteristic lengths and wetted perimeters of each unit cell region.

The heat transfer problem remains nevertheless complex, and not too many works attempted at modelling two-phase slug flow and the interactions between the mass, momentum and energy balances. The objective of the present work is to analyze the influence of the heat transfer on the hydrodynamics of the gas-liquid two-phase slug flow by using a transient, lagrangian slug tracking model developed by Medina *et al.* (2015).

2. MATHEMATICAL MODEL

The transient slug tracking model developed by Medina *et al.* (2015) to predict the heat transfer in gas-liquid slug flows in horizontal pipes will be used in this work. His model will be briefly presented, as the present work is an extension of the results obtained by Medina *et al.* (2015). The complete model is found in the original work (Medina *et al.*, 2015).

The model of Medina *et al.* (2015) is one-dimensional and applies the mass, momentum and energy equations in their integral form to a lagrangian control volume that encapsulates the whole unit cell. The main hypotheses of this model are: (i) one-dimensional, horizontal flow, (ii) Newtonian fluids, (iii) the rear of the elongated bubble is flat, (iv) constant and homogeneous phase fractions in both the elongated bubble and liquid slug regions throughout time and space, (v) uniform pressure in the cross section of the pipe, (vi) constant external temperature, (vii) the momentum of the gas is negligible, (viii) the pressure loss within the mixing zone is negligible, (ix) negligible viscous dissipation, (x) kinetic energy negligible with regard to the internal energy, (xi) incompressible liquid and (xii) ideal gas.

The mass, momentum and energy balance equations are applied to the control volumes shown in Figure 1. This control volume is deformable and its borders travel along with the unit cell borders. The unit cell is composed of a liquid slug (*LS*) with small dispersed bubbles (*GS*) in its interior and an elongated bubble (*GB*), which slides over a liquid film (*LB*). The index *j* represents the current unit cell under analysis. The coordinates x_j and y_j delimit the control volume around the liquid slug; y_j is located at the elongated bubble nose whereas x_j is at the rear of the succeeding elongated bubble. The x_{j-1} and y_j coordinates delimit the control volume around the elongated bubble, the latter positioned at the tip of the bubble nose and the former at the bubble rear.

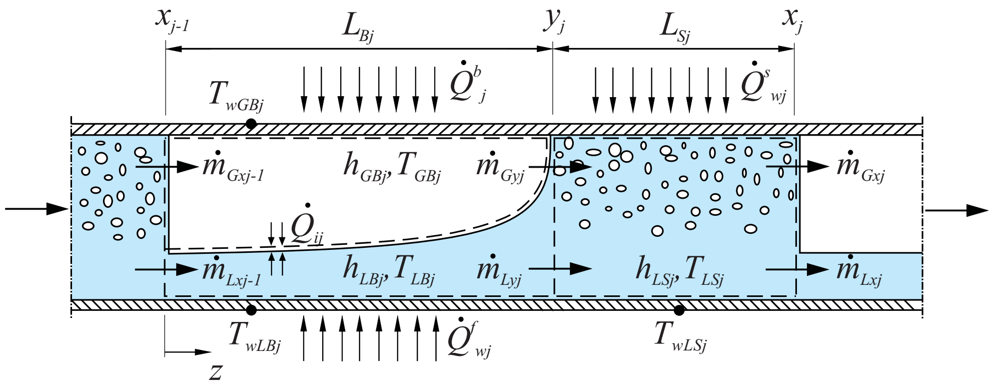

Figure 1. Slug tracking control volumes.

With the purpose of determining the liquid velocity in the slug, Medina *et al.* (2015) applied a mass balance on the liquid slug and the elongated bubble regions. The overall liquid mass balance in the entire unit cell then becomes:

$$U_{LSj-1}-U_{LSj}=\left(\frac{1-R_{LSj}}{R_{LSj}}\right)U_{DSj}-\left(\frac{1-R_{LSj-1}}{R_{LSj-1}}\right)U_{DSj-1}+\left(\frac{1}{P_{GBj}}\frac{dP_{GBj}}{dt}-\frac{1}{T_{GBj}}\frac{dT_{GBj}}{dt}\right)\times \\ \times\left[L_{Bj}\left(1-R_{LBj}\right)+\frac{L_{Sj}}{2}\left(1-R_{LSj}\right)+\frac{L_{Sj-1}}{2}\left(1-R_{LSj-1}\right)\frac{\rho_{GBj}}{\rho_{GBj-1}}\right] \tag{1}$$

where U_{LS} is the liquid slug velocity, U_{DS} the slip velocity of the bubbles dispersed within the liquid slug, R is the phase fraction, L the length of each unit cell region, ρ is the density, t is the time and T the temperature of the gas in the elongated bubble. The subscripts L, G, S and B refer to the liquid and gaseous phases, the liquid slug and the elongated bubble regions, respectively. Equation (1) expresses the differential velocity between two consecutive liquid slugs. As it can be observed, this differential velocity is affected by variations in the phase fractions and in the slip velocity between two adjacent unit cells (1st and 2nd terms in the RHS), and by the pressure and temperature gradients (3rd RHS term).

The pressure variation between two adjacent unit cells is obtained by applying the momentum balance to the unit cell shown in Fig. 1. This balance yields:

$$P_{GBj}-P_{GBj+1}=\tau_{LSj}\frac{S_{LSj}L_{Sj}}{A}+\tau_{LBj+1}\frac{S_{LBj+1}L_{Bj+1}}{A}+\rho_L R_{LSj}L_{LSj}\frac{dU_{LSj}}{dt} \tag{2}$$

where P is the pressure, τ is the shear stress, S the wetted perimeter and A the cross sectional area of the pipe. The equation above expresses the relationship between the pressures of two consecutive elongated bubbles as a function of the pressure losses due to the friction of the liquid slug and the liquid film on the pipe wall (1st and 2nd RHS terms), plus an acceleration term associated to the liquid slug (3rd RHS term).

Energy conservation is applied to the control volumes around the liquid slug, the elongated bubble and the liquid film so that the temperature in each of those regions can be found. Due to their low concentration in horizontal flows, the contribution associated to the dispersed bubbles within the liquid slug was neglected, thus yielding the following expression:

$$\overline{h_{LSj}^{G}}S_{LSj}L_{Sj}\left(T_{wLSj}-T_{LSj}\right)=\dot{m}_{Lxj}C_L T_{Lxj}-\dot{m}_{Lyj}C_L T_{Lyj}+\rho_L A C_L\left(T_{LSj}R_{LSj}\frac{dL_{Sj}}{dt}+R_{LSj}L_{Sj}\frac{dT_{LSj}}{dt}\right) \tag{3}$$

where $\overline{h^G}$ is the overall heat transfer coefficient which takes the thermal resistance associated to the wall and the external convection into account. T is the temperature and the subscript w refers to the wall. The liquid mass flow rates at the x and y boundaries originate from the differential velocity between the phases, a phenomenon known as *scooping* (Shoham, 2006). Those mass exchanges are given by $\dot{m}_{Lxj}=\rho_L A R_{LSj}\left|U_{LSj}-dx_j/dt\right|$ and $\dot{m}_{Lyj}=\rho_L A R_{LSj}\left|U_{LSj}-dx_j/dt\right|$, where dx_j/dt and dy_j/dt are the displacement velocities of the bubble rear and of the bubble front, respectively. Equation (3) expresses the relationship between the heat exchange and the wall (LHS), the temperature variation of the liquid at the liquid slug boundaries (1st and 2nd RHS terms) and the energy variations inside the liquid slug due to changes in the control volume length and in the temperature (3rd RHS term). By analogy, the energy balance can be applied to the liquid film and to the elongated bubble, thus giving:

$$\overline{h_{LBj}^{G}}S_{LBj}L_{Bj}\left(T_{wLBj}-T_{LBj}\right)+h_{ij}S_{ij}L_{Bj}\left(T_{GBj}-T_{LBj}\right)=c_L\dot{m}_{Lyj}T_{Lj-1}-c_L\dot{m}_{Lyj}T_{Lj-1}+ \\ +\rho_L A c_L\left(T_{LBj}R_{LBj}\frac{dL_{Bj}}{dt}+R_{LBj}L_{Bj}\frac{dT_{LBj}}{dt}\right) \tag{4}$$

$$\overline{h_{GBj}^{G}}S_{GBj}L_{Bj}\left(T_{wGBj}-T_{GBj}\right)-\overline{h_{ij}}S_{ij}L_{Bj}\left(T_{GBj}-T_{LBj}\right)=c_{P,G}\dot{m}_{Gyj}T_{Gyj}-c_{P,G}\dot{m}_{Gxj-1}T_{Gyj-1}+ \\ +c_{\mathrm{V},G}A\rho_{GBj}\left(T_{GBj}R_{GBj}\frac{dL_{Bj}}{dt}+T_{GBj}R_{GBj}\frac{L_{Bj}}{\rho_{GBj}}\frac{d\rho_{GBj}}{dt}+R_{GBj}L_{Bj}\frac{dT_{GBj}}{dt}\right) \tag{5}$$

where the gas mass flow rates at the boundaries are expressed by $\dot{m}_{Gxj}=\rho_{GBj}A\left(1-R_{LSj}\right)\left|U_{GSj}-dx_j/dt\right|$ and $\dot{m}_{Gyj}=\rho_{GBj}A\left(1-R_{LSj}\right)\left|U_{GSj}-dy_j/dt\right|$. Equations (4) and (5) make up a system of equations that must be simultaneously solved in order to determine the temperatures of the gas in the elongated bubble and of the liquid in the liquid slug.

Those equations are similar to the one that calculates the energy balance in the liquid slug, eq. (3), with an extra term to account for the heat exchange between the elongated bubble and the liquid film (2nd RHS term). There is yet another extra term accounting for energy variations in the elongated bubble due to gas compressibility effects (eq. (5), 3rd term in the RHS).

The model still requires closure and to achieve that the velocities at the boundaries y_j and x_j must be known. The velocity at y_j is the elongated bubble front velocity, given by an experiment-based and widely accepted correlation proposed by Nicklin (1962). As for x_j, it corresponds to the elongated bubble rear velocity and can be modelled through the gas mass balance within the unit cell, similarly to eq. (3). The velocities of the boundaries are then given by the following two equations:

$$\frac{dy_j}{dt} = U_{Tj} = \left(C_0 J + C_1\sqrt{gD}\right)\left(1+\hbar\right) \tag{6}$$

$$\frac{dx_{j-1}}{dt} = \frac{R_{GSj}U_{GSj} - R_{GSj-1}U_{GSj-1} + \left(\frac{1}{P_{GBj}}\frac{dP_{GBj}}{dt} - \frac{1}{T_{GBj}}\frac{dT_{GBj}}{dt}\right)\left(L_{Bj}R_{GBj} + \frac{L_{Sj}R_{GSj}}{2} + \frac{L_{Sj-1}R_{GSj-1}}{2}\right) + \left(R_{GBj} - R_{GSj}\right)\frac{dy_j}{dt}}{R_{GBj} - R_{GBj-1}} \tag{7}$$

where J is the mixture velocity, C_0 and C_1 constants experimentally calculated by Bendiksen (1984) and $\hbar$ the wake effect coefficient. This latter represents the acceleration of the elongated bubble due to the low-pressure (suction) zone developed at the rear of the elongated bubble travelling just downstream of the liquid slug. The wake effect coefficient is inversely proportional to the length of the liquid slug and is given by $\hbar = a_w exp(-b_w L_S / D)$, with a_w = 0.4 and b_w = 1.0. Liquid and gas mass balances in their respective regions determine the velocities of the liquid film and of the elongated bubble, as proposed by Taitel and Barnea (1990) in their stationary model.

Due to the intermittent character of the flow, the translational velocity of the j^{th} bubble might exceed that of the $(j+1)^{th}$ bubble. Bubble coalescence – a phenomenon that takes place whenever two consecutive bubbles blend into a single one of larger dimensions – may therefore occur. Whenever coalescence takes place, the properties of the newly-formed cell are averaged with the properties of the former individual cells.

The overall heat transfer coefficient is obtained from the thermal resistances associated to the external convection, the heat conduction through the pipe wall (be it a single- or multiple-layer wall) and the internal convection. For each cell region – namely the liquid slug, the elongated bubble and the liquid film – single-phase flow can be assumed. The Gnielinski (1976) single-phase correlation for calculating the heat transfer coefficient can therefore be used for each unit cell region, as proposed by França *et al.* (2008). França *et al.* (2008) also proposed that the heat transfer coefficient of the liquid slug region should be increased in 30%, due to the swirling in the recirculation zone behind an elongated bubble.

For the sake of convenience, the calculated temperatures of each region might be expressed as a function of a single temperature, T_{mj}, the mixture temperature of unit cell j. This is the temperature that is actually measured during the course of lab experiments since the thermocouple cannot determine what unit cell structure is passing by it whenever the temperature is read. Applying an energy balance in a unit cell and neglecting the gas contributions since $C_L \rho_L U_L \gg C_G \rho_G U_G$, the mixture temperature can be expressed as:

$$T_{mj} = \frac{U_{LSj}R_{LSj}T_{LSj} + U_{LBj}R_{LBj}T_{LBj}}{U_{LSj}R_{LSj} + U_{LBj}R_{LBj}} \tag{8}$$

Finally, the two-phase heat transfer coefficient must be predicted. Similarly to the analysis developed by França *et al.* (2008), comes:

$$\overline{h_{TPj}} = \frac{\overline{h_{LSj}}L_{Sj}\left(T_{wiLSj} - T_{LSj}\right) + \frac{\overline{h_{LBj}}S_{LBj}L_{Bj}\left(T_{wiLBj} - T_{LBj}\right) + \overline{h_{GBj}}S_{GBj}L_{Bj}\left(T_{wiGBj} - T_{GBj}\right)}{\pi D}}{L_{Sj}\left(T_{wiLSj} - T_{LSj}\right) + L_{Bj}\left(T_{wiLBj} - T_{LBj}\right)} \tag{9}$$

where $\overline{h_{LS}}$, $\overline{h_{LB}}$ and $\overline{h_{GB}}$ are, respectively, the average single-phase heat transfer coefficients of the liquid in the slug and in the film and of the gas in the elongated bubble. It must be pointed out that the average temperatures in each unit cell structure, namely T_{LSj}, T_{LBj} e T_{GBj}, are given by the averages of each phase within their respective boundaries.

3. SOLUTION METHODOLOGY

The mass and momentum balances, eqs. (1) and (2), and the liquid energy balances, eqs. (3) and (5), were discretized according to the Crank-Nicholson finite difference scheme. Since this is a forced-convection problem, the hydrodynamic and the heat transfer models are decoupled and can thus be solved separately. Equations (1) and (2) form a $2n$-order system for the pressure-velocity, where n is the number of unit cells inside the pipe. Equations (3) and (5) make up another $2n$-order system, this time for the temperatures of the liquid at the boundaries of a unit cell. The linear systems were solved using the TDMA (*Tridiagonal Matrix Algorithm*) method. The gas and the mixture temperatures as well as the heat transfer coefficient can be found after solving the two systems of equations.

The model was implemented as a Fortran95 objected-oriented code. The transient slug tracking simulator for solving two-phase gas-liquid slug flows with heat transfer requires the following input data: the temperature of the liquid at the inlet; the pressure at the pipe outlet; the temperature and heat transfer coefficient of the external medium; the pipe geometry; and either the superficial velocities of the phases or their mass flow rates. Moreover, the simulator needs the structures lengths and phase fractions at the pipe inlet, here assumed the same as the experimental data for validation. An alternative method is to use a stationary model, such as the Taitel and Barnea's (1990) one. As output results, the slug flow simulator yields the spatial and time distributions of pressure, velocity, temperature, heat transfer coefficient and structure lengths.

4. RESULTS AND DISCUSSION

The model was already validated trough comparison with experimental data in previous works. The hydrodynamic model, i.e., the pressure-velocity linear system, was validated for air-water horizontal, inclined and vertical flows by Rodrigues (2009) and Conte *et al.* (2011). The heat transfer model, i.e., the liquid temperature system, was validated for air-water horizontal flows by Medina *et al.* (2015). The mean accuracy found for the main parameters lied approximately on ±30% for the pressure gradient, the temperature gradient and the two-phase heat transfer coefficient. Further information can be found on the previous works about spatial and time distribution of the main parameters.

The slug tracking simulator is used here to analyze the influence of the heat transfer on the hydrodynamic parameters of the flow. Three different superficial velocity (j_G, j_L) pairs were picked for numerical simulation purposes. Those pairs were j_L = 1.38 m/s and j_G = 0.28 m/s (predominant liquid superficial velocity), j_L = 0.33 m/s and j_G = 1.20 m/s (predominant gas superficial velocity) and j_L = 0.53 m/s and j_G = 0.47 m/s (balanced liquid and gas superficial velocities). Those superficial velocity pairs were experimentally obtained by Rosa and Altemani (2006) for isothermal air-water flows in a 20-m long, 26-mm ID acrylic pipe. The measured lengths of the liquid slug and elongated bubble as well as their phase fractions were used as input data to the simulator. Three numerical simulations were run for each superficial velocity pair; one considering isothermal flow, one simulation for heating and the remaining one for cooling. As the experiments were carried out in constant temperature (isothermal flow) conditions, fictitious heat sources were considered in the heating and cooling simulation cases. A constant heat flux of ±10,000 W/m^2 through the pipe wall was adopted. This amount was assumed as being sufficiently large to allow an analysis of the influence of the heat transfer without phase change (that is, no vaporisation or condensation). The inlet temperature was assumed as constant and equal to 290K for all simulations.

Figures 2a, 2c and 2e (left column) show how the dimensionless elongated bubble length varies for the three superficial velocity pairs under isothermal, heating and cooling conditions. As it can be observed, under isothermal conditions the elongated bubble tends to grow from the pipe inlet to the outlet due to the decreasing pressure. When the heat flow is positive (heating), the temperature increase makes the gas to expand further, thus increasing the elongated bubble length. Cooling, on the other hand, counterbalances the pressure drop in a sort of competitive effect. When the liquid phase predominates ($j_L > j_G$), the pressure drop effects overcome the temperature drop ones, and the elongated bubble expands slightly. The opposite is observed whenever either the gas phase predominates ($j_G > j_L$) or a balance between the phases ($j_G \approx j_L$) exist.

Figures 2a, 2b and 2e clearly show that, for larger superficial liquid velocities, the pressure drops predominate over the temperature drops. The mechanisms behind the increase in the pressure and temperature drops must be analyzed so that this phenomenon can be explained. The equations that predict the pressure drop (Taitel e Barnea, 1990) and the heat transfer coefficient (Camargo, 1991) under stationary conditions provide a better understanding about that phenomenon:

$$\frac{dP}{dz} = \frac{\tau_{LS} S_{LS}}{A} \frac{L_S}{L_U} + \left(\frac{\tau_{LB} S_{LB}}{A} + \frac{\tau_{GB} S_{GB}}{A} \right) \frac{L_B}{L_U} \tag{10}$$

$$h_{TP} = h_{LS} \frac{S_{LS}}{S} \frac{L_S}{L_U} + \left(h_{LB} \frac{S_{LB}}{S} + h_{GB} \frac{S_{GB}}{S} \right) \frac{L_B}{L_U} \tag{11}$$

Assuming a negligible friction to the gas in the bubble and considering that $S_{LS} > S_{LB}$ and $U_{LS} > U_{LB}$, then $\tau_{LS} > \tau_{LB}$ and an increase in the L_S/L_B ratio will always increase the pressure drop. Likewise, should the heat exchange in the gas bubble be negligible and $U_{LS} > U_{LB}$, then heat transfer coefficient in the liquid slug is bigger than the one associated to the liquid film, $h_{LS} > h_{LB}$, thus an increase in the L_S/L_B ratio will always increase the two-phase heat transfer coefficient and therefore an increase in the temperature gradient will be observed. That leads to the conclusion that an increase in the liquid rate will always increase the pressure and temperature gradients. However, nothing can be affirmed about the predominance of the pressure drop effects over the temperature drop effects when it comes to the competitive effect affecting gas expansion/contraction and the resulting increase/decrease in the elongated bubble length.

A second mechanism that explains the increase in the pressure and temperature gradients with the increase in the superficial liquid velocity must therefore be considered. For a constant superficial gas velocity, the superficial mixture velocity increases with an increase in the superficial liquid velocity as $J = j_L + j_G$. This implies in an increase in the translational velocity of the elongated bubble, U_T, eq. (6). The translational velocity, by its turn, affects the mass balance and as a consequence the local phases' velocities in each region increase; in other words, U_{LS} and U_{LB} increase. Those augmented velocities increase the local Reynolds numbers and, for that reason, the shear stresses and the pressure drop also increase. The local Nusselt numbers also increase as they directly relate to the Reynolds number, and thus larger temperature drops are observed due to an increase in the heat transfer coefficients.

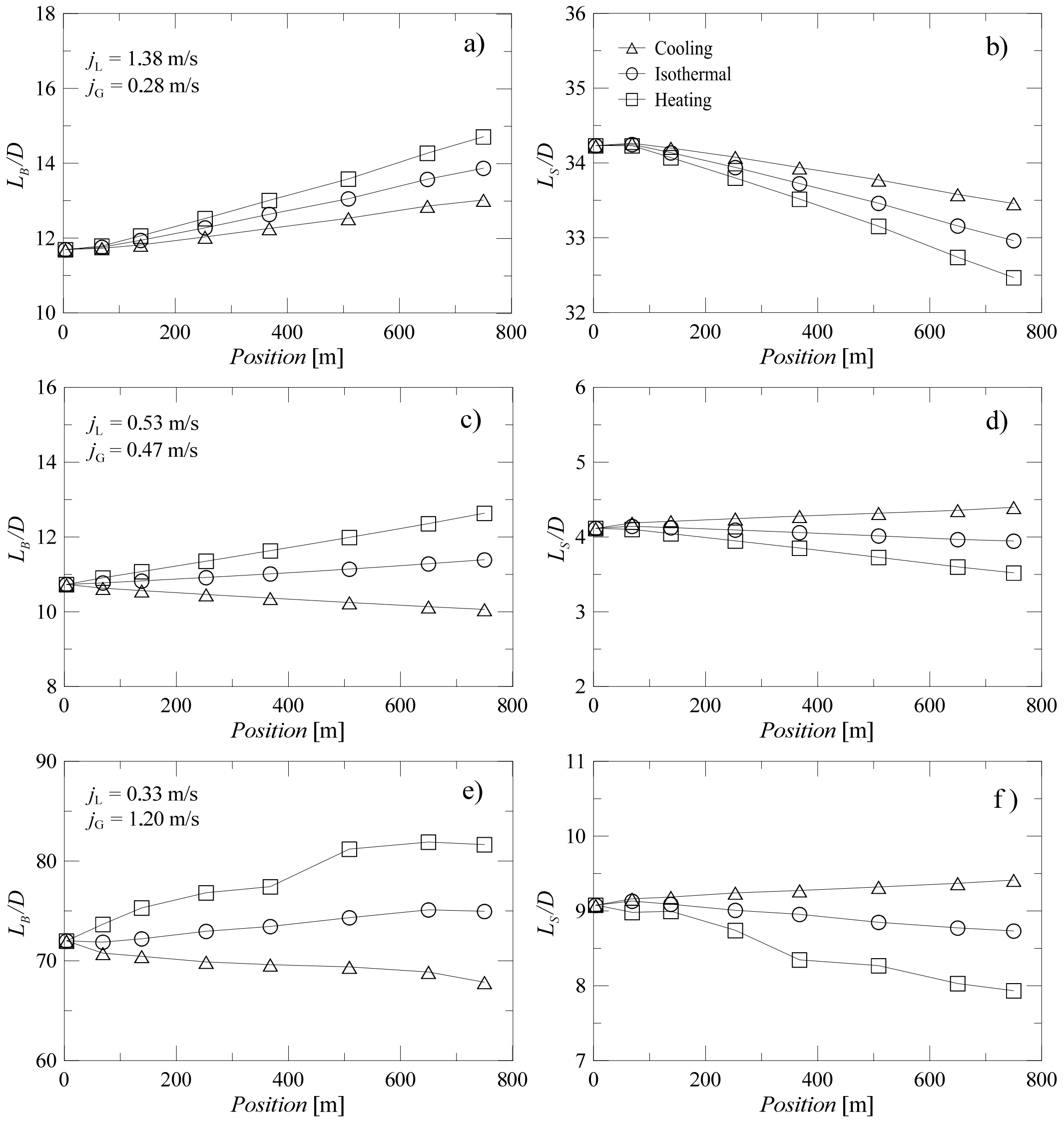

Figure 2. Heat transfer influence: a), c), e) on the elongated bubble length and b), d), f) on the liquid slug length.

Thus, an increase in the superficial liquid velocity increases both the pressure and the temperature gradients. It is clear, however, that in this second mechanism the increase in the temperature gradient is smaller than the increase in the pressure gradient, as the thermal mechanism follows a less straightforward path ($Re \rightarrow Nu \rightarrow h_{TP} \rightarrow dT_m/dz$, instead of

Re → τ → *dP/dz*). Therefore, an analysis shows that the pressure gradient overcomes the competitive effect on the elongated bubble length at higher liquid superficial velocities.

Finally, a last assertion about the behaviour of the elongated bubble length can be made, this time using the superficial gas velocity as a parameter. From Figures 2a, 2c and 2e it becomes evident that the variation of elongated bubble length, $\Delta L_B / \Delta z$, that is, the curve inclination, is steeper for larger superficial gas velocities. That can be explained on the grounds that a larger mass of gas is expanding at a constant superficial liquid velocity.

Once the behaviour of the elongated bubble due to the variation of its length is understood, the comprehension about the behaviour of the remaining hydrodynamic parameters becomes simple. Figures 2b, 2d and 2f (pictures at the right) show the distribution of the dimensionless liquid slug length, L_S/D. Such length behaves inversely to that of the elongated bubble. As the liquid is assumed incompressible and the phases' fractions are constant, any changes in the liquid slug length are due solely to the mass of liquid that is either captured or shed by the liquid film due to the variations in the elongated bubble length. Such a phenomenon is known as *scooping*/*shedding* (Shoham, 2006).

The distributions of the elongated bubble translational velocity are shown in Figures 3a, 3c and 3e (pictures at the left). The trend of this velocity is similar to that of the elongated bubble length, that is, it either increases or decreases due to the gas expansion/contraction effects caused by the pressure and temperature gradients. From Figure 3e, one can observe that the slug tracking simulator presents some numerical oscillations at higher gas flow rates. Previous works (Rodrigues, 2009; Medina; 2015) reported this behaviour. Those oscillations are stronger during heating, when the pressure and temperature gradients act together, favouring the expansion of the gas phase. This behaviour is also observed in all other variables in greater or lesser extent; see Figures 2e, 2f and Figure 3f.

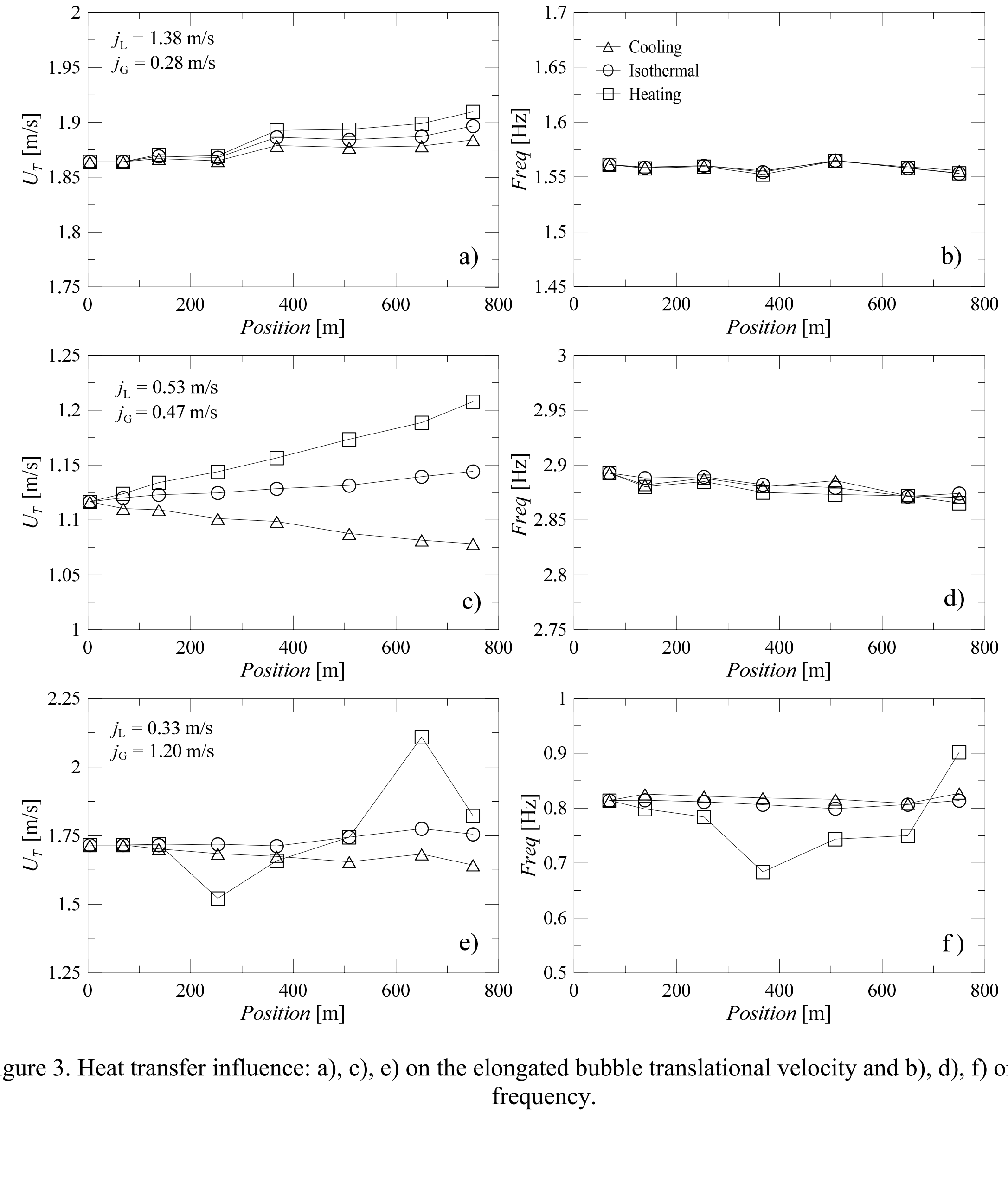

Figure 3. Heat transfer influence: a), c), e) on the elongated bubble translational velocity and b), d), f) on the flow frequency.

Figures 3b, 3d and 3f (pictures at the right) show the distribution of the slug flow frequency. This frequency is defined as the number of unit cells passing by a fixed point in the pipe per unit time. Since the unit cell travels with the elongated bubble translational velocity (as proposed in stationary models such as the one developed by Taitel and Barnea, 1990), the frequency is therefore given by $f = U_T/(L_S + L_B)$. The frequency behaviour is thus the behaviours of the elongated bubble translational velocity and the lengths of the liquid slug and elongated bubble, combined. For the given superficial velocity pairs and the pipe length/diameter, a noticeable frequency variation along the pipe was not observed; those frequencies variations stood in the range of 0.01 to 0.05 Hz. Larger variations were observed in the cases where numerical instabilities occurred, see the Figure 3f (heating). However, given the non-physical character of those instabilities, they should not be considered in the present analysis.

5. CONCLUSIONS

The influence exerted by heat transfer on the hydrodynamic parameters of two-phase gas-liquid slug flows was analyzed by using the slug tracking simulator developed by Medina *et al.* (2015). The thermal modelling implemented in the simulator was validated with experimental data obtained by Lima (2009), presenting an average accuracy of 30% for both the heat transfer coefficient and the temperature gradient. Isothermal, heating and cooling cases were simulated for flows with predominance of either the liquid or the gas, and for flows with a balanced mixture of the two fluids. In isothermal flows, pressure drops tend to increase the volume of the gas phase, whereas pressure and temperature work together to expand the gas when heating takes place. There is, however, a competitive effect when it comes to the expansion/contraction of the gas phase during cooling. It was also observed that the pressure gradient tends to overcome the temperature gradient effects when superficial liquid velocities are high. The velocities of the elongated bubble as well as its lengths are directly affected by gas expansion/contraction effects. As for the liquid slug length, its behaviour is opposite to that of the elongated bubble due to the phenomenon known as scooping or shedding, that is, a mass of liquid from the liquid slug that is captured by the liquid film and shed afterwards to the preceding liquid slug. Finally, the frequency does not present a significant variation along the pipe for any simulated heat exchange case.

6. ACKNOWLEDGEMENTS

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7. REFERENCES

Bendiksen, K., 1984. "An Experimental Investigation of the Motion of Long Bubbles in Inclined Tubes". *International Journal of Multiphase Flow.* Vol. 10, No. 4, pp. 467-483.

M.G. Conte, C.L. Bassani, C. Cozin, A.E. Nakayama, C.D.P. Medina e R.E.M. Morales (2011). "Numerical analysis of slug flow in inclined ducts using slug tracking model", XXXII CILAMCE - Congresso Ibero Latino Americano de Métodos Computacionais em Engenharia, Ouro Preto/MG.

Deshpande, S., Bishop, A., e Karandikar, B. (1991). "Heat transfer to air-water plugslug flow in horizontal pipes". *Ind. Eng. Chem Res*, 30(9).

Dukler, A. E. and Hubbard, M. G., 1975. "A model for gas-liquid slug flow in horizontal and near horizontal tubes". *Ind. Eng. Chem. Fundam.*, 14 (4), pp. 337-347.

Fernandes, R. C., Semiat, R., Dukler, A. E. (1983). "Hydrodynamic model for gas-liquid slug flow in vertical tubes". *American Institute for Chemical Engineers Journal*, 29(6), pp. 981-989.

França, F., Bannwart, A. C., Camargo, R. M. T., Gonçalves, M., 2008. "Mechanistic Modeling of the Convective Heat Transfer Coefficient in Gas-Liquid Intermittent Flows". Heat Transfer Engineering, 29(12), pp. 984-998.

Franklin, E. M., Rosa, E. S., 2004. "Dynamic slug tracking model for horizontal gas-liquid flow". Proc. of the 3rd International Symposium on Two-Phase Flow Modeling and Experimentation.

Gnielinski, V., 1976. "New equations for heat and mass transfer in turbulent pipe and channel flow". *International Chemical Engineering*, 16 (2), pp. 359-368.

Grenier, P., 1997. "Evolution des longueurs de bouchons en écoulement intermitttent horizontal". Ph.D. thesis, Institut National Polytechnique de Toulouse, France.

Kim, J. and Ghajar, A., 2006. "A general heat transfer correlation for non-boiling gas-liquid flow with different flow patterns in horizontal pipes". *International Journal of Multiphase flow*, 32, pp. 447-465.

Lima, I. N. R. C., 2009. *Estudo Experimental da Transferência de Calor no Escoamento Bifásico Intermitente Horizontal.* Master Thesis, University of Campinas, Campinas.

Medina , C. D. P., Bassani C. L., Cozin C., Barbuto F. A. de A., Junqueira S. L.M., Morales R.E.M., 2015. "Numerical simulation of the heat transfer in fully developed horizontal two-phase slug flows using a slug tracking method". *International Journal of Thermal Sciences*, 88 (2015) pp. 258-266.

Nicklin, D. J., Wilkes, J. O. e Davidson, J. F., 1962. “Two-phase flow in vertical tubes”. Trans. Instn. Chem. Engnrs, Vol. 40, pp 61-68.

Nydal, O. J., Banerjee, S. (1995). “Objected oriented dynamic simulation of slug flow”. Proc. of the 2nd Int. Conf. of Multiphase Flow, Kyoto, Vol. 2, pp. IF2_7-14.

Rodrigues, H. T., 2009. “Simulação numérica de escoamento bifásico gás-líquido no padrão de golfadas utilizando um modelo lagrangeano de seguimento de pistões”. Master thesis, Federal Technological University of Paraná, Curitiba.

Rosa, E. S., Altemani, C. A. C., 2006. “Análise de Escoamentos em Golfadas de Óleos Pesados e de Emulsões Óleo-água”. Quarto Relatório de Atividades e Resultados alcançados, Projeto UNICAMP / CENPES – PETROBRAS, Campinas, SP.

Shah, M., 1981. “Generalized prediction of heat transfer during two-component gas-liquid flow in tubes and other channels”. *AIChE Symp*. Ser., 77 (207), pp. 140-151.

Shoham, O., 2006. “Mechanistic Modeling of Gas-Liquid Two-phase Flow In Pipes”. *Society of Petroleum Engineers*. Richardson, TX, 396p.

Taitel, Y. and Barnea, D., 1990. “Two phase slug flow”. Advances in Heat Transfer, Hartnett J.P. and Irvine Jr. T.F. ed., vol. 20, 83-132, Academic Press.

Taitel, Y. e Barnea, D., 2000. “Slug tracking model for hilly terrain pipelines”, SPE Journal, Vol. 5, pp. 102-109.

Ujang, P. M., Lawrence, C. J. e Hewitt, G. F., 2006. “Conservative incompressible slug tracking model for gas-liquid flow in a pipe”. Proc. of the 5th BHRG North American Conference on Multiphase Technology, Banff, Canadá.

Wallis G. B., 1969. *One dimensional two-phase flow*. New York, McGraw-Hill.

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