STRATIFIED FLOW PATTERN MODELS AND THEIR PERFORMANCE WITH REFRIGERANTS

Paulo Eduardo Lopes Barbieri
Centro Federal de Educação Tecnológica de Minas Gerais, Dpto de Eng. Mecânica, Av.Amazonas 7675, Belo Horizonte – MG.
barbieri@des.cefetmg.br

José Maria Saiz Jabardo
Escuela Politécnica Superior, Universidad de la Coruña, Mendizabal s/n Esteiro, 15403 Ferrol, Coruña, Spain
mjabardo@cdf.udc.es

Enio Pedone Bandarra Filho
Faculdade de Engenharia Mecânica – Universidade Federal de Uberlândia – Av. João Naves de Ávila, 2160, Uberlândia- MG.
bandarra@mecanica.ufu.br

Abstract. *The convective boiling of refrigerants in direct expansion evaporators is characterized by the occurrence of the following typical flow patterns: stratified, intermittent and annular. Their occurrence depends on the operating conditions, with the mass velocity being the most important physical parameter in the development of a particular flow pattern. The stratified flow pattern occurs under medium to low mass velocities, typical in numerous refrigeration applications. The purpose of the present paper is to perform a comparative study of stratified flow dynamic models of the literature. Since most of the published investigations aim at applications other than refrigeration, their use for refrigerants and tube diameters typical of the refrigeration field is questionable. The application of these models in the evaluation of pressure drop requires the use of friction factors and, in some cases, the shape and size of the liquid-vapor interface. By performing a comparative analysis of the different analytical models it has been determined that reasonable pressure drop results under convective boiling conditions could be obtained from some of the aforementioned models if both adequate friction factors are used and the liquid-vapor interface is correctly modelled.*

Keywords*: two-phase flow, stratified flow, refrigeration, pressure drop*

1. INTRODUCTION

Stratified flow patterns are of common occurrence in horizontal two-phase flow applications. They are characterized by slower flow of liquid at the bottom of the channel with the separated fast moving gas phase flowing in the upper region. The interface may assume different configurations depending on the relative velocities of the phases, as illustrated in the pictures of Fig. 1 for the flow of refrigerant R-134a. In smooth stratified flow pattern, such as the one of Fig. 1(a), the interface is clearly free of waves and is typical of flows at reduced mass velocities. At elevated mass velocities the interface becomes rough with the presence of waves of sizes varying with the relative velocity between the phases. The picture of Fig. 1(b) illustrates this type of interface configuration.

Figure 1. Pictures showing two interface conditions of the stratified flow pattern for the flow of Refrigerant R-134a at the temperature of 5°C in tubes of 12.6 mm of internal diameter. (a) Smooth: G=50 kg/s.m2; x=17%; (b) wavy: G=150 kg/s.m2, x=60%. (Barbieri (2005)).

The main thrust to the research of stratified flow patterns has been given by the petrochemical industry where two phase pipe flows are of common occurrence. As a result, modeling of this type of two-phase flow pattern has been developed in connection with flow conditions and tube diameters typical of petrochemical applications. Under these circumstances, the use of those models under different operational conditions, fluids and tube diameters could be questionable. This would be the case of refrigerating applications where the stratified flow pattern is of common occurrence in evaporators and condensers.

The objective of the present paper is to report both a literature survey of the stratified flow pattern modeling and a summary of results from modified models of the flow of refrigerants in typical diameter tubes.

2. A BRIEF LITERATURE SURVEY

The early models related to the evaluation of pressure drop under two-phase flow conditions appeared in the 40's in connection with the fairly new nuclear industry. Worth mentioning is the investigation performed by Martinelli and co-workers, Lockhart and Martinelli (1949). Their empirical two-phase flow multipliers, consisting on correcting the single-phase pressure drop, were well received by the scientific community. Though simple, their use required empirical adjustments that limited the range of their applications. In recent years mathematical models have been developed for the stratified flow pattern that took into account the specific interface topology of this pattern, an aspect not considered by early models. Among those models, worth of mentioning in chronological order are the ones by Russel *et al.* (1974), Taitel and Dukler (1976), Cheremisinoff and Davis (1979), Hart *et al.* (1989), Chen *et al.* (1997), Vlachos *et al.* (1999), Barbieri (2005), and Ullmann and Brauner (2006). An aspect that distinguishes these models from each other is the approach to the interface configuration. Thus, whereas the more recent papers include the concavity of the interface, the older ones (those from the 70's) assume a flat interface "ad hoc", an aspect that certainly limits their range of applications. Several configurations have been assumed in the literature for the topology of the liquid-gas interface, among them the most representative are the ones shown in Fig.2.

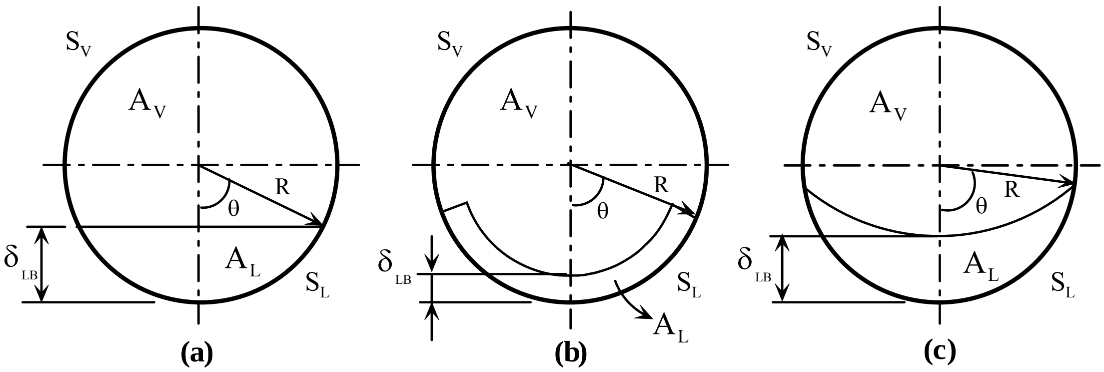

Figure 2. Typical representations of the liquid-gas interface in stratified flow. (a) Flat; (b) Curved concave with constant liquid thickness; and (c) Concave.

Taitel and Dukler, in their classic 1976 paper, aimed at developing a generalized two-phase flow map based on modeling the different patterns transitions. Though assuming a flat interface stratified pattern, their model, based on the continuity and momentum equations for both phases, has been a common reference for most of the more recent investigations. Among these, there has been found a common concern regarding the improvement of stratified flow pattern models: the need to introduce effects of both the interface curvature and roughness. Thus all the aforementioned papers from the 90's on have tried to introduce effects of interface curvature and roughness on the two-phase momentum equations.

In what follows, a summary of these developments will be presented along with results obtained from modified models applied to the flow of refrigerants. In the development of the following sections reference will be made to the contributions of other authors not mentioned so far.

3. INTERFACIAL FRICTION FACTOR

Some of the models for the momentum transfer at the liquid-gas interface and between the phases and the tube wall in stratified flow pattern use single phase expressions for the friction factor with the characteristic dimension of the Reynolds number being the flow area hydraulic diameter of each phase. This is the case of early models such as the ones by Lockhart and Martinelli (1949) and Taitel and Dukler (1976). The use of this procedure for the momentum transfer at the interface can lead to inadequate results. Andritsos and Hanratty (1987) suggested empirical correction factors to the gas phase friction factor. Others , like Hart *et al.* (1989), raised empirical correlations that include the effect of the interface roughness. It must be noted that interface roughness may affect not only the interfacial friction factor but also the liquid to wall and gas to wall friction factor. According to Kowalski (1987), the effect is more significant in the case of liquid to wall momentum transfer. Interfacial friction factor correlations have also been proposed by Vlachos *et al.* (1999) and Barbieri (2005). In the following paragraphs an outline of interfacial friction factor models will be presented, stressing the results from the one proposed under the present investigation.

The one dimensional momentum equation for both phases can be written as follows:

$$-A_v \frac{dP}{dz} - \tau_{pv} S_v - \tau_i S_i = 0 \qquad (1)$$

$$-A_l \frac{dP}{dz} - \tau_{pl} S_l + \tau_i S_i = 0 \tag{2}$$

where "A" is the flow area, "τ" the shearing stress, "S" the perimeter, and "dP/dz" the pressure gradient, which is assumed equal for both phases. The subscripts stand for: "v" gas phase, "l" liquid phase, "pv" gas-wall, "pl" liquid-wall and "i" interface. Figure 2 displays a schematic representation of the geometry of stratified flow and may be used as reference for the momentum equations, Eqs. (1) and (2).

Shear stresses can be defined in terms of the Fanning friction factor in each phase as follows:

$$\tau_{pv} = f_v \frac{\rho_v u_v^2}{2}; \quad \tau_{pl} = f_l \frac{\rho_l u_l^2}{2}; \quad \tau_i = f_i \frac{\rho_v \left(u_v - u_l\right)^2}{2} \tag{3}$$

where "f" stands for the Fanning friction factor, "ρ" for the density, and "u" for the average velocity. The phase average velocities can be defined in terms of the corresponding superficial velocities as follows:

$$u_v = \frac{J_v}{\alpha}; \quad u_l = \frac{J_l}{\left(1-\alpha\right)} \tag{4}$$

where "J" stands for the superficial velocity and "α" for the void fraction. The superficial velocities can be written in terms of the mass velocity of the liquid-gas mixture, "G", and the quality, "x":

$$J_v = \frac{Gx}{\rho_v}; \qquad J_l = \frac{G\left(1-x\right)}{\rho_l} \tag{5}$$

Liquid and gas friction factors are usually written in terms of single phase typical correlations, one of them is that due to Blasius, with the following general form:

$$f_v = C_v \, Re_v^{-n}; \qquad f_l = C_l \, Re_l^{-m} \quad \Rightarrow \quad \begin{cases} Re < 2100 \Rightarrow C = 16 \text{ e } m,n = 1,0 \\ Re \geq 2100 \Rightarrow C = 16 \text{ e } m,n = 0,2 \end{cases} \tag{6}$$

It must be noted that Reynolds number for each phase is defined as:

$$Re_v = \frac{\rho_v D_v u_v}{\mu_v}; \quad Re_l = \frac{\rho_l D_l u_l}{\mu_l} \tag{7}$$

where "D_v" and "D_l" are gas and liquid phases hydraulic diameters, given by:

$$D_v = \frac{4A_v}{S_v + S_i}; \quad D_l = \frac{4A_l}{S_l} \tag{8}$$

The flow areas of each phase are related to the void fraction by the following equations:

$$A_l = \left(1-\alpha\right)\frac{\pi D^2}{4}; \qquad A_v = \alpha \frac{\pi D^2}{4} \tag{9}$$

where "D" is the pipe diameter. The hydraulic diameter definition is adjusted to the relative velocity between phases. In general, the interface is considered still with respect to the faster phase and free surface for the slower one. In addition, it must be noted that the individual phase transition Reynolds number is significantly lower under two-phase flow conditions, as pointed out by Hurlburt and Newell (1999). This should be expected given the liquid-gas momentum interaction in two-phase flow.

Ullmann and Brauner (2006) assumed a still interface with respect to both phases and neglected the relative motion, though the phase interaction was introduced as a correction factor in their model equations.

Kowalski (1987) argued that the use of a Blasius type of equation for the friction factors in stratified two-phase flow can lead to inadequate results, especially in the liquid phase. However, the Taitel and Dukler (1976) model, with friction factors given in terms of the Blasius correlation, correlated well Kowalski experimental gas phase shear stress data. Regarding the liquid phase friction factor, Kowalski suggested the following equation due to Agrawal apud Kowalski (1987):

$$f_l = 0,263\left[\left(1-\alpha\right)Re_l^+\right]^{-0,5} \quad ; \qquad Re_l^+ = \frac{\rho_l J_l D}{\mu_l} \tag{10}$$

Andritsos and Hanratty (1987) modified the Cheremisinoff and Davis (1979) model for the liquid-wall shear stress. According to the Cheremisinoff and Davis model, τ_{pl} is given in terms of a non-dimensional liquid film thickness, $\delta_{LB}{}^+$, defined as:

$$\delta_{LB}^+ = \frac{\delta_{LB} u^*}{\upsilon_l} \tag{11}$$

where u^* is the "shear stress velocity", $\sqrt{\tau_c \rho_l}$, written in terms of a "characteristic shear stress", τ_c, defined as:

$$\tau_c = \frac{2}{3}\tau_{pl}\left(1-\frac{\delta_{LB}}{D}\right)+\frac{1}{3}\tau_i \tag{12}$$

Andritsos and Hanratty arrived at the following expression relating δ_{LB}^+ with δ_{LB}/D and Re_l:

$$\delta_{LB}^+ = \left\{\left(1.082\,Re_l^{0.5}\right)^5 + \left[\frac{0.098\,Re_l^{0.85}}{\left(1-\frac{\delta_{LB}}{D}\right)^{0.5}}\right]^5\right\}^{0.2} \tag{13}$$

The liquid-wall shear stress can then be determined through an iterative procedure, what makes its use rather difficult to implement.

Andritsos and Hanratty (1987) suggested that the interfacial friction factor depends on two non-dimensional parameters, that is:

$$\frac{f_i}{f_v} - 1 \sim \psi\left(\frac{\delta_{LB}}{D}, \frac{J_v}{u_t}\right) \tag{14}$$

where "J_t" is the so called "transition superficial velocity", defined as:

$$J_t = 5\left(\frac{\rho_{vo}}{\rho_v}\right)^{0,5} \tag{15}$$

where "ρ_{vo}" is the gas density at atmospheric pressure. Finally, Andritsos and Hanratty curve fitted equations to the general correlation, Eq. (14), arriving to the following expressions:

$$\frac{f_i}{f_v} = 1 \quad \text{para} \quad J_v \le J_t \tag{16}$$

$$\frac{f_i}{f_v} = 1 + 15\left(\frac{\delta_{LB}}{D}\right)^{0,5}\left(\frac{J_v}{J_t}-1\right) \quad \text{para} \quad J_v > J_t \tag{17}$$

where "f_v" is determined from Eq. (6). It can be noted that for the range of lower gas phase velocities the interfacial friction factor is equal to the gas phase one, a result that seems reasonable from the physical point of view. At higher gas velocities, the liquid film thickness affects the interfacial friction factor.

The model proposed by Andritsos and Hanratty (1987) correlates reasonably well experimental results though it does not include the curvature of the interface. According to Chen *et al.* (1997) the curvature of the interface is affected mainly by the gas velocity, and it can affect the liquid-wall contact area with possible effects over the interfacial and liquid friction factors. Chen *et al.* (1997) modified the former Andritsos and Hanratty (1987) correlation to include the interface curvature, arriving at the following equation:

$$\frac{f_i}{f_v} = 1 + 3,75\left[\frac{(1-\alpha)}{\Theta}\right]^{0,20}\left[\frac{J_v}{u_t} - 1\right]^{0,08} \tag{18}$$

where "Θ" is the liquid wetting fraction of the tube wall. The transition superficial velocity is the one proposed by Taitel and Dukler (1976):

$$(J_t)_C = \left[\frac{4\,\upsilon_l(\rho_l - \rho_v)g}{s\,\rho_v u_l}\right]^{0,5} \tag{19}$$

The parameter "s" is a "sheltering coefficient", introduced by Jeffreys (1925) apud Taitel and Dukler (1976). This coefficient constitutes the necessary condition for the wave formation at the interface. Andritsos and Hanratty (1987) assumed "s" as being equal to 0.06, a value introduced by Chen *et al.* (1997) into their model.
The model proposed by Chen *et al.* (1997) requires the determination of the wetted tube wall fraction, Θ. They proposed the following correlation from Hart *et al.* (1989):

$$\Theta = 0,52\,(1-\alpha)^{0,374} + 0,26\,Fr^{0,58} \tag{20}$$

where the modified Froud number, "Fr", is defined as:

$$Fr = \frac{\rho_l u_l^2}{(\rho_l - \rho_v)g\,D} \tag{21}$$

The investigation of the friction factors in stratified flow has been limited to specific conditions such as air-water mixtures and tube diameters generally larger than 25 mm. The proposed models and correlations are of limited extent since most of them are the result of curve fitting procedure. The model proposed by Andritsos and Hanratty (1987) is a worth to mention exception, since it is mostly based on physical arguments.

Recently Barbieri (2005) developed correlations for the interfacial friction factor that include the curvature of the interface and are applicable to fluids and tube diameters different from most of the ones considered in the literature. Figure 3(a) displays experimental data from Barbieri (2005) in terms of the interfacial friction factor against the parameter ($J_V/J_{t,i}$). Curve fitted results from data shown in this plot along with others not shown for the liquid friction factor obtained during the same investigation resulted in the following two correlations:

$$f_l = 0,6\left[\frac{u_l}{u_{r,l}}\right]^{-1,25} \tag{22}$$

$$f_i = 12,5\left[\frac{J_v}{J_{t,i}}\right]^{-1,05} \tag{23}$$

where "$u_{r,l}$" is the so called "reference velocity", defined as:

$$u_{r,l} = \left[\frac{4\upsilon_l(\rho_l - \rho_v)g}{\rho_v\,u_l}\right]^{0,5} \tag{24}$$

and "$J_{t,i}$" is the "transition superficial velocity", defined as:

$$J_{t,i} = \left[\frac{4\upsilon_l \left(\rho_l - \rho_v\right) g}{\rho_v \left(u_v - u_l\right)} \right]^{0,5} \tag{25}$$

A comparison between the interfacial friction factors, given in terms of the gas superficial velocity, from Barbieri's model with that proposed by Chen *et al.* (1997) is shown in Fig. 3 (b). It can be noted that the friction factor of the present model diminishes with the superficial velocity whereas Chen et al results present a rather constant behavior. Both models compare very well with each other in the range of gas superficial velocities higher than 3 m/s. Deviations in the lower range could be attributed to differences in flow conditions and two-phase fluids used as reference in modeling.

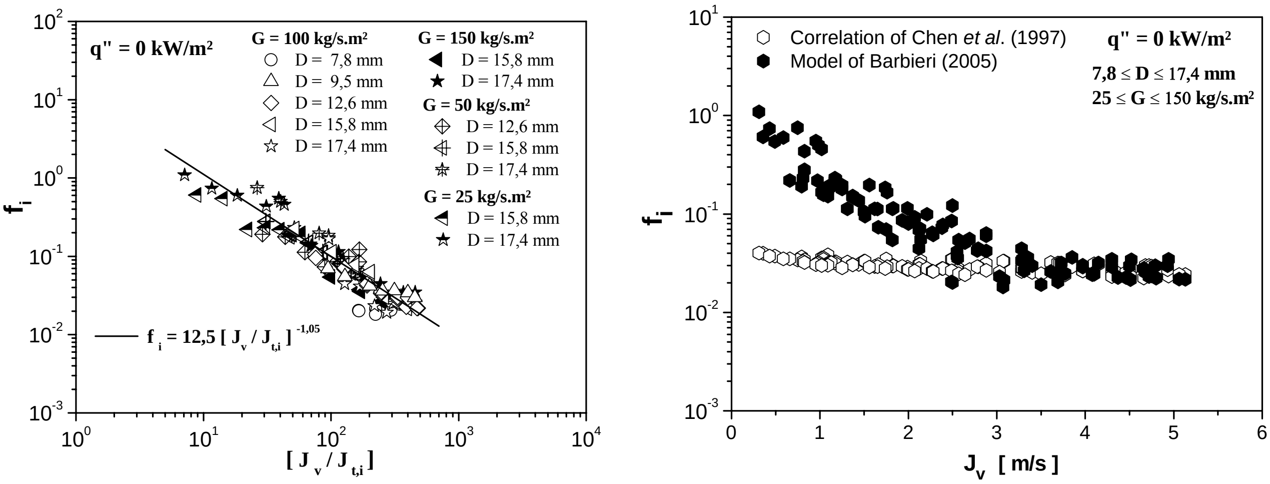

Figure 3. Interfacial friction factor. (a) Experimental data from the present investigation in terms of the parameter $(J_v/J_{t,i})$, Barbieri (2005); (b) a comparison between the interfacial friction factors obtained by Chen et al. (1997) and present model in terms of the gas phase superficial velocity.

Figure 4 has been included in order to compare the performance of the models of Fig. 3(b) in the determination of the pressure drop in pipe two-phase flow. The plot of Fig. 4 includes the calculated pressure drop in a 1.5 m long tube versus the experimental one. Flow conditions are included in the plot. The evaluation of the pressure drop has been done by using the model equations for the interface friction factor by Barbieri (2005) and Chen et al (1997), Eqs. (23) and (18), respectively. It can be noted that Barbieri's model correlated experimental data within $\pm 15\%$, whereas Chen et al results clearly under predict experimental data. However, as mentioned before, Chen et al model was based on two-phase flow data involving air – water and air – kerosene mixtures in pipes of higher diameter than those considered in the comparison of Fig. 4.

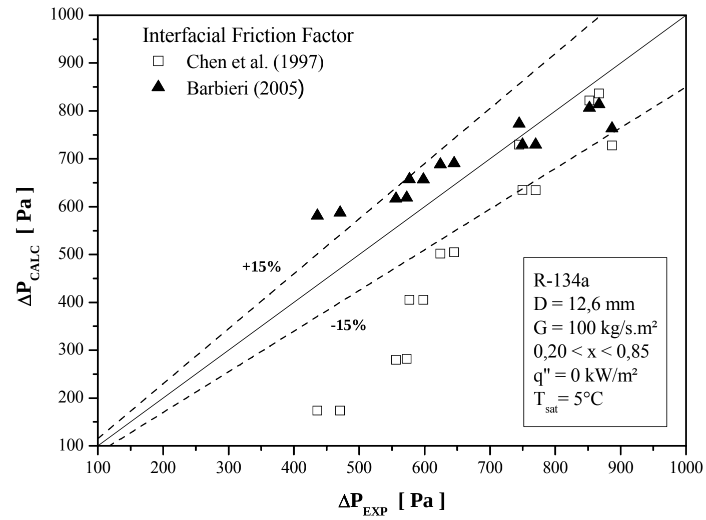

Figure 4. Calculated versus experimental results for the models by Barbieri (2005) and Chen et al (1997).

As already emphasized before, and Figs. 3(b) and 4 confirm, the use of empirical models developed for given set of conditions in a range outside the one for which they have been developed might is not recommended since the results might be inadequate.

4. INTERFACE GEOMETRY

In previous paragraphs the need to adequately evaluate the topology of the liquid-gas interface was emphasized. In this section an outline of the most recent interface models will be discussed.

Former models such as the ones proposed by Taitel and Dukler (1976), Cheremisinoff and Davis (1979), and Hart *et al.* (1989) assumed the interface geometry as starting point to the model development. Chen *et al.* (1997) proposed a model that adjusts the interface topology to the flow conditions. Their model is based on the "dual circle procedure", illustrated in Fig. 5. Chen *et al.* (1997) superposed a circle to the one that represents the tube geometry, as shown in Fig. 5. The centre and radius of the second circle are O_i and R_i. Its centre is located at a distance "y_{ci}" from the tube centre. Both the centre position of the second circle and its radius vary with flow conditions.

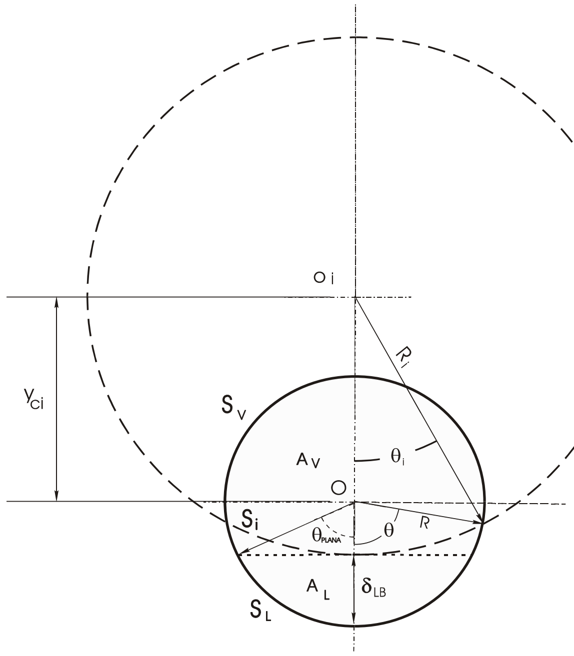

Figure 5. Schematic representation of the dual circle model, Chen *et al.* (1997).

The following correlations can be obtained from the dual circle model geometry:

$$S_v = (\pi - \theta)D \;; \quad S_l = \theta D \;; \quad S_i = \theta_i D_i \;; \quad S = \pi D; \;\; \Theta = \frac{S_l}{S} \tag{26}$$

$$D_i = D\frac{\text{sen}\,\theta}{\text{sen}\,\theta_i} \tag{27}$$

$$A_l = R^2\left(\theta - \frac{\text{sen}\,2\theta}{2} - \frac{\text{sen}^2\theta}{\text{sen}^2\theta_i}\theta_i + \frac{\text{sen}^2\theta}{\text{tg}\,\theta_i}\right) \tag{28}$$

$$A_v = A - A_l \tag{29}$$

$$y_{ci} = R\left(\frac{\text{sen}\,\theta}{\text{tg}\,\theta_i} - \cos\theta\right) \tag{30}$$

$$\delta_{LB} = R\left(1 - \frac{\text{sen}\,\theta}{\text{sen}\,\theta_i} + \frac{\text{sen}\,\theta}{\text{tg}\,\theta_i} - \cos\theta\right) \tag{31}$$

The following additional correlation can be obtained by introducing Eq. (9) into Eq. (28):

$$\theta_i - \left(\frac{\operatorname{sen}\theta_i}{\operatorname{sen}\theta}\right)^2 \left(\theta + \frac{\operatorname{sen}^2\theta}{\operatorname{tg}\theta_i} - \frac{\operatorname{sen}2\theta}{2} - \pi(1-\alpha)\right) = 0 \tag{32}$$

It must be noted that the angle "θ_i" is positive when measured in the counter-clockwise direction, with the centre O_i above the tube centre. The interface configuration varies with θ_i according to the following rules:

- $\theta_i \to 0 \Rightarrow$ flat interface;
- $\theta_i > 0 \Rightarrow$ concave interface;
- $\theta_i < 0 \Rightarrow$ convex interface.

The above set of equations can be transformed into the following for a flat interface, corresponding to $D_i \gg D$ and $\theta_i \to 0$:

$$S_v = (\pi - \theta_{plana})D \;;\quad S_l = \theta_{plana} D \;;\; S_i = D \operatorname{sen}\theta_{plana} \;;\quad S = \pi D \;;\; \Theta = \frac{S_l}{S} \tag{33}$$

$$A_{l,plana} = R^2 \left[\theta_{plana} - \left(\frac{\operatorname{sen}2\theta_{plana}}{2}\right)\right] \tag{34}$$

$$y_{ci} = -R\cos\theta_{plana} \tag{35}$$

$$\delta_{LB} = R\left(1 - \cos\theta_{plana}\right) \tag{36}$$

$$\theta_{plana} - \left(\frac{\operatorname{sen}2\theta_{plana}}{2}\right) - \pi(1-\alpha) = 0 \tag{37}$$

The shape of the interface can be determined by previously evaluating the values of α, θ, and θ_i. However, these parameters depend on the flow conditions, the type of fluid, and the tube diameter. In addition to these parameters, according to Sutharshan *et al.* (1995), the interface topology is also affected by the following physical mechanisms:

- secondary flows;
- droplets dispersion and deposition;
- wave effects;
- pumping action of large waves.

According to Brauner *et al.* (1996) and Ullmann and Brauner (2006), the interface topology can be determined from the "minimum energy principle" which is the state corresponding to the minimum total energy, potential and surface components, of the two-phase flow. The minimum energy is the total energy excess per unit of tube length with respect to that of the flat interface, Δe. The minimum energy can be determined from the following equation:

$$\Delta e = \frac{1}{8}(\rho_l - \rho_v) g D^3 \left\{ \begin{aligned} &\frac{\operatorname{sen}^3\theta}{\operatorname{sen}^2\theta_i}(\operatorname{ctg}\theta_i - \operatorname{ctg}\theta)\left(\frac{\operatorname{sen}2\theta_i}{2} - \theta_i\right) + \frac{2}{3}\operatorname{sen}^3\theta_{plana} + \\ &+\varepsilon_o \left[\frac{\operatorname{sen}\theta}{\operatorname{sen}\theta_i}\theta_i - \operatorname{sen}\theta_{plana} + \cos\xi\left(\theta_{plana} - \theta\right)\right] \end{aligned} \right\} \tag{38}$$

where "ε_o" is the Eotvös number, defined as $\varepsilon_o = 8\sigma/[(\rho_l - \rho_v)gD^2]$, "$\sigma$" is the surface tension, and "ξ" the contact angle of the liquid with the tube wall. It can be noted that Eq. (38) includes two components that incorporate the potential and the surface (interface) energies. The latter is related to the surface tension through the effect of the interface curvature variation with the contact area of the phases with the tube wall. The former is related to the variation of the centre of mass with the curvature of the interface. According to Brauner *et al.* (1996), when surface effects are dominant, the interface can be either concave or convex, depending on the wettability of the tube surface. On the other hand, when gravitational effects are dominant and secondary flows are unimportant, the interface tends to be flat. However, it has been determined that even when the surface effects are small, corresponding to low values of the

Eotvös number, the interface tends to acquire a concave configuration at reduced void fractions. As a general conclusion, Brauner *et al.* (1996) suggest that for Eotvös numbers lower than 10^{-3}, surface effects can be neglected. Under these circumstances, gravity effects are dominant and the interface acquires a flat configuration.

As part of the investigation reported herein, Barbieri (2005) combined the minimum energy procedure, Eq. (38), with the dual circle model by Chen *et al.* (1997) to develop a model of the topology of the interface for convective boiling of refrigerants. Barbieri compared his interface generating model with images obtained by Wojtan *et al.* (2004) from the flow of refrigerant R-410A at 5°C, flowing in a 13.6 mm tube diameter with a mass velocity of 70 kg/s.m^2 and an average quality of 20%. The comparison of model generated and experimental images is shown in Fig. 6. The pictures from Wojtan *et al.* (2004) were taken in a time interval of 1.66 seconds corresponding to the displacement of a wave through the viewing section. The time variation of the void fraction during this period is shown in Table 1. It must be stressed that transient effects associated to the large wave displacement through the viewing section have been neglected in the model application. It can be noted that the model generated interface deviates from the experimentally obtained one, showing a more pronounced concavity. The deviation is more significant in the region closer to the wall, though, in general, both images are very close, a clear indication of the adequacy of the proposed model.

Table 1. Time variation of the void fraction corresponding to the images of Fig. 6.

Time (s)	9.50	10.00	11.00	11.16
α	0.537	0.685	0.794	0.497

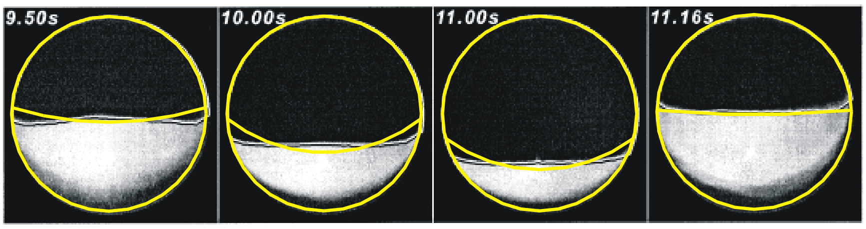

Figure 6. Images of a wavy stratified flow pattern overlaid with model generated ones. Refrigerant R-410A; T_{sat}=5°C; G=70 kg/sm^2, x_{av}=20%. (a) $\alpha = 0.537$; (b) $\alpha = 0.685$; (c) $\alpha = 0.794$; (d) $\alpha = 0.497$.

The model proposed by Hart *et al.* (1989) for the interface topology is based on a curve fitted correlation for the wetted fraction of the tube wall, Eq. (20). The other model, due to Hart *et al.* (1989), was developed for high void fractions, a condition that makes the assumption of constant film thickness adequate, an interface configuration illustrated in Fig. 2(b). The generalization of both models to conditions other than the ones for which they were developed is rather questionable due to their curve fitting character.

5. CONCLUSIONS

The objective of the present paper was to perform a literature survey of the models and procedures for the determination of the two-phase flow friction factors, especially the liquid and the interface ones, along with an analysis of the more recent model dealing with the interface topology of the stratified flow patter. In addition to the literature survey, an outline of the models developed under the reported investigation was presented comparing their results with those obtained elsewhere. The comparison revealed the adequacy of the proposed models. Finally, it must be noted that most of the proposed models are based on curve fitting procedures what makes their extension to different operating conditions rather questionable.

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