A STUDY OF FLOW PATTERNS FOR GAS/LIQUID FLOW IN SMALL DIAMETER TUBES

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Abstract: Time resolved void fraction data and flow pattern information have been obtained for two-phase air/water flows in a small diameter (5 mm) vertical pipe using conductance probes. The time averaged void fractions are seen to agree with values measured using a completely different approach.

Analysis of high speed videos reveals that the probability density function (PDF) technique is inadequate for accurately delineating the transition between slug and churn flows but performs better for the churn to annular flow transition. Instead a novel approach has been developed for transitions between flow patterns using the velocity of structures. This gives good agreement with the present experiments. Additionally, a modification to the bubble-to-slug flow transition of Taitel et al. (1980) gives improved predictions in narrow passages.

A flow pattern specific method for void fraction prediction has been applied and its predictions are in good agreement with the measured mean void fraction. The slug flow model gives better predictions of void fraction in churn flow that the annular flow model.

The velocities of disturbance waves on the wall film in annular flow are well predicted using the model of Pearce (1979). However, pipe diameter dependence of one of the constants is required.

Keywords: two-phase flow; flow pattern; void fraction; small diameter; vertical pipe; conductance probes; compact heat exchanger; nuclear fuel bundles.

INTRODUCTION

Two-phase flow in one form or another exists in at least 60% of all process industry heat exchange equipment (Hewitt, 1981). However, most available data has been obtained for tube diameters typical of shell and tube exchangers. Two-phase flows in a small diameter tube have become important for compact heat exchanger applications and for nuclear reactors with tight passages. Compact heat exchangers are increasingly being applied to the chemical and process industries because of the significant benefits they offer. The national energy saving potential of this technology has been estimated to be worth 8 PJ year\(^{-1}\) (ETSU, 1994). They are used because of their greater effectiveness, smaller volume, improved safety and power savings when compared to most conventional heat exchanger types.

Much of the published information on two-phase flow characteristics have been confined to conduits with diameters greater than 20 mm. However, Oya (1971), Barnea et al. (1983), Wambsganss et al. (1992), Bao et al. (1994), Holt et al. (1999), Cheng and Lin (2001) and Okawa et al. (2004) are some of the researchers to have worked with tubes of smaller diameters. The recent review papers by Cheng and Mewes (2006) and Thome (2004) show that the challenges of exchanging heat with smaller surface areas such as those encountered in electronic devices are increasing research interest in fluid flow in small diameter channels. Barnea et al. (1983) found the pipe diameter effect to be minimal for upward air/water flow in vertical tubes with diameters ranging from 4 to 12.3 mm. Holt et al. (1999) used circular and non-circular channels with hydraulic diameters ranging from 3.9 to 10 mm to collect two-phase pressure drop and void fraction data in order to investigate typical conditions found in compact evaporators. The CISE (Premoli et al., 1970) and Lockhart/Martinelli correlations were found to accurately predict the circular void fraction data only. Although the results obtained were compared with a novel flow pattern specific method, the flow patterns transitions were not studied. Cheng and Mewes (2006) conclude that any future research should include studies of two-phase flow pattern transitions in small and mini channels.

Workers including Hibi et al. (2000), Okubo et al. (2000) and Sakashita et al. (1999) report...
that several nuclear installations expect to attain high conversion ratios by reducing the moderation of neutrons. This results in the decrease of the core water volume and corresponds to a reduction in the cross-sectional area of flow channels. Okawa et al. (2004, 2005) working with a tube whose inside diameter is 5 mm, obtained experimental data of the deposition rate and entrainment fraction in annular flow to investigate the onset of critical heat flux condition, which is essential to the aforementioned nuclear power plants.

All heat exchanger design involves optimizing heat transfer and pressure drop characteristics, both of which are affected by the various two-phase flow patterns. Although the pressure drop characteristics are often overlooked, the mechanical pumping power expended to overcome friction and gravity is sometimes equal to that consumed in heat transfer when dealing with gases. It has been reported (Kays and London, 1998) that this mechanical energy is worth four to 10 times its heat equivalent in most thermal systems. In essence, mechanical energy can be transformed into heat with maximum efficiency (e.g., through friction) but when converting heat into mechanical energy (e.g., in a power station) some of the energy will be lost and 100% efficiency is never attainable. As a consequence, knowledge of the void fraction is useful because it is required for determining the acceleration and gravity components of pressure drop, from which the frictional component can be inferred.

In the present study, void fraction data and flow pattern information have been obtained from a small diameter (5 mm) pipe. High-speed video pictures, time-varying void fraction data and probability density functions (PDFs) have been used to discriminate between the various flow patterns. An empirical flow pattern map for the conditions of our experiment has been developed and compared with the most commonly used correlations for flow pattern transitions in two-phase vertical upflow. The velocities of the various structures present in the flow were determined and have been compared with the relationship proposed by Nicklin et al. (1982) and Pearce (1979) for slug velocity and the velocities of disturbance waves on wall film respectively. Some of our data has been taken in similar conditions to those of Holt et al. (1999), who used the method of quick closing valves for void fraction measurements. The results from the two sets have been compared. The mean void fraction values obtained have been tested against a flow pattern specific method to assess its predictive capability.

FLOW PATTERNS

Flow Pattern Classification

The classification of flow patterns in co-current vertical gas–liquid systems, by Hewitt and Hall-Taylor (1970), into four major groups: bubble, slug, churn and annular flow is now widely accepted. These flow patterns are depicted in Figure 1.

In bubble flow, the gas phase is approximately uniformly distributed in a continuous liquid phase in the form of discrete bubbles. Slug flow occurs at slightly higher gas flow rates. In this flow pattern, most of the gas phase is situated in large bullet shaped bubbles referred to as ‘Taylor bubbles’. Successive Taylor bubbles are separated by liquid slugs that bridge the pipe and usually contain small gas bubbles. Churn flow is a more chaotic, frothy and disordered version of slug flow which occurs at still higher gas flows. The shape of the Taylor bubbles is distorted resulting in narrower bubbles and the direction of the liquid phase in churn flow is oscillatory or alternating. Annular flow is characterized by continuity of the gas phase along the pipe in the core, occurring at the highest gas flow rates. Liquid flows in the upward direction, both as a thin wavy film and as entrained droplets in the gas core.

Flow Pattern Transitions

The widely applied models by Taitel et al. (1980), Jayanti and Hewitt (1992) and Barnea (1986) for the bubble-to-slug, slug-to-churn and churn-to-annular flow transitions respectively are to be tested to determine their suitability for small channels of similar size to those found in compact heat exchangers.

Taitel et al. (1980) postulated that discrete bubbles combine into larger vapour spaces whose diameter approximates to the tube by agglomeration. This is assumed to occur as the void fraction reaches 0.25 and results in a transition to slug flow.

Jayanti and Hewitt (1992) concurred that the flooding mechanism appears to be the most likely cause of the transition from slug to churn flow in vertical tubes and suggested an improvement to the modelling of the flooding mechanism given by McQuillan and Whalley (1985) which was capable of predicting the transition over the full range of liquid rates. Watson and Hewitt (1999) found that the model of Jayanti and Hewitt (1992) gives better predictions of the slug-to-churn flow transition than the methods by McQuillan and Whalley (1985), Mishima and Ishii (1984) and Brauner and Barnea (1986) in a 32 mm diameter tube.

The model for the transition from annular to intermittent flows (slug or churn) developed by Barnea (1986) is proposed to occur when the gas core is blocked at any location by the liquid. There are two mechanisms by which this blockage may occur. The first mechanism which predominates at low liquid flow rates is results from the instability of the annular configuration while the second mechanism is caused by spontaneous blockage of the gas core as a result of axial transfer of liquid in the film.

Flow Pattern Specific Model for Void Fraction

Published models in the literature for predicting void fraction in slug and annular flows have been applied to the present work. For churn flow, which is the least understood flow pattern intermediate between slug and annular flows, both models have been applied to determine the most appropriate.
 Slug model

Slug flow is assumed to consist of a succession of identical unit cells. A unit cell comprises a liquid slug and a cylindrical Taylor bubble surrounded by a falling liquid film. A zero void fraction is assumed in the liquid slugs based on the observation of de Cachard and Delhaye (1996) in their 10 mm diameter vertical pipe. Ros (1961) has proposed that the condition for a non-aerated liquid slug is given by

\[
Bo = \frac{(\rho_l - \rho_g)gD^2}{\sigma} < 140
\]

This corresponds to a diameter of less than 32 mm for the low pressure systems being considered in this study. The assumption therefore appears to be reasonable.

The widely used expression by Nicklin et al. (1962) for the Taylor bubble velocity is

\[
u_b = 1.2(u_g + u_g) + u_0
\]

From equations (2) and (3), the average void fraction may be expressed by

\[
u_g = \frac{u_g}{1.2(u_g + u_g) + u_0}
\]

\[u_0\] is determined by the correlation of White and Beardmore (1962)

\[u_0 = \Gamma(gD)^{1/2}
\]

\[
\Gamma = 0.345(1 - e^{[-0.01 N_v^{0.345}]}(1 - e^{3.37 - Bo/r}))
\]

\[r = 10 \text{ when } N_v > 250
\]

\[r = 69(N_v)^{-0.35} \text{ when } 18 < N_v < 250
\]

\[r = 25 \text{ when } N_v < 18
\]

\[N_v = \sqrt{\frac{0.1((\rho_l - \rho_g)gD^3)}{\mu_L^2}} < 3.10^{5}
\]

Annular model

In equilibrium annular flow, the deposition rate, \(m_d\), equals the entrainment rate, \(m_e\).

\[m_e = m_d
\]

\(m_d\) is assumed to be proportional to the droplet concentration in the gas core, C.

\[m_d = k_d C
\]

where \(k_d\) is the mass transfer coefficient of droplet deposition. \(m_d\) can be approximated by

\[C \approx \frac{\mu_l E u_g}{u_g}
\]

\[u_0\] where \(E\) is the entrained fraction.

Assuming the liquid film is sufficiently thin and the droplet velocity relative to the gas phase is sufficiently small. Hence,

\[m_d = k_d C \approx k_d \frac{\mu_l E u_g}{u_g}
\]

Whalley and Hewitt (1978) and Ishii and Mishima (1989) show that increasing the interfacial shear force causes an increase in the amount of droplets in equilibrium but increasing the liquid surface tension has a reverse effect. The rate of droplet entrainment is thus assumed to be characterized by the dimensionless number, \(\pi_e\) which is defined as the ratio of interfacial force to the surface tension force acting on the liquid film surface.

\[
\pi_e = \frac{f_l \rho_l u_g^2}{\sigma/\bar{a}}
\]

The mean liquid film thickness, \(\bar{a}\) is determined from the balance between the interfacial shear force and wall friction force acting on the liquid film:

\[f_l \rho_l u_g^2 \approx f_w \rho_g u_g^2
\]

As the liquid film is thin, \(u_g\) and \(u_l\) are approximated by \(u_g\) and \(D/4\bar{a}u_g\) respectively. The interfacial friction factor \(f_l\) and the wall friction factor \(f_w\) are estimated using correlations by Wallis (1969):

\[f_l = 0.005 \left(1 + 300 \frac{\bar{a}}{D}\right)
\]

\[f_w = \frac{16}{Re_H} \text{ or } 0.005, \text{ whichever is greater}
\]

Furthermore, the volume of the entrained droplets is assumed to be proportional to the dimensionless term, \(\pi_e\).

\[\frac{m_e}{\rho_l} \propto \pi_e
\]

From equations (11) and (14), the following expression is obtained:

\[\frac{m_e}{\rho_l} = k_d E u_g \frac{u_g}{u_g}
\]

Okawa et al. (2004) correlated annular flow data by

\[k_d E u_g = k_o \pi_e^0
\]

where

\[k_o = 3.1 \times 10^{-2} \text{ ms}^{-1} \text{ and } n = 2.3 \text{ for } \pi_e < 0.0675
\]

\[k_o = 1.6 \times 10^{-3} \text{ ms}^{-1} \text{ and } n = 1.2 \text{ for } 0.0675 < \pi_e < 0.295
\]

\[k_o = 6.8 \times 10^{-4} \text{ ms}^{-1} \text{ and } n = 2.3 \text{ for } \pi_e > 0.295
\]

The use of the Okawa et al. (2004) correlation is justified as it was developed using experimental data obtained with the same fluid pairing as the present work (air/water) and in
pipes whose range of diameters (5–42 mm) includes that used in the present study. The void fraction is determined from

\[ \varepsilon_g = \left( \frac{D - 2\bar{a}}{D} \right)^2 - E \]  

(25)

**EXPERIMENTAL**

The vertical two-phase flow experiments were carried out on a specially designed facility in the School of Chemical, Environmental and Mining Engineering (SchEME). The test fluids used were air and mains tap water at ambient temperature. The gas and liquid superficial velocities ranged from 0.062 to 64.5 m s\(^{-1}\) and 0.03 to 0.65 m s\(^{-1}\) respectively at pressures ranging from atmospheric to 2 bara. These flow rates gave slug, churn and annular flow patterns. The flow loop is shown schematically in Figure 2.

Air was drawn from the compressed air main and supplied to the mixing unit where it combines with water drawn from a storage tank by means of a centrifugal pump. Inflow of air and water are controlled using separate banks of calibrated rotameters. The mixing unit consisted of an annular section surrounding a porous wall.

Water enters the main pipe from the periphery to form a film on the wall whilst the air passes along the middle. Downstream of the mixer, the two phases travel for 1.97 m up to the test section along a 0.005 m diameter stainless steel vertical pipe, where time varying, cross-section averaged void fractions were measured using a cascade of eight identical conductance probes placed sequentially along the pipe. Pressure measurements were taken with a pressure gauge mounted close to the test section. Beyond the test section, the two-phase flow travels for 0.67 m upwards, 1.07 m horizontally and finally a further 3.12 m vertically downwards. The outlet stream is collected in a cylindrical vessel of 100 mm diameter where the phases are separated. The pressure in the test section was controlled using a gate valve positioned at the separator inlet. Air is metered at the top of the separator vessel using a calibrated wet gas flow meter which allowed the flow rates to be measured over a period of time. The liquid level in the separator vessel was kept constant to ensure accuracy. Detailed calculations of the measurement errors were made by Omebere-Iyari (2006).

**Figure 2.** Schematic of the small diameter rig.

**Table 1.** Physical properties of air and water at test conditions.

<table>
<thead>
<tr>
<th>Fluids</th>
<th>Pressure (bara)</th>
<th>Liquid density (kg m(^{-3}))</th>
<th>Gas density (kg m(^{-3}))</th>
<th>Liquid viscosity (Pa s(^{-1}))</th>
<th>Gas viscosity (Pa s(^{-1}))</th>
<th>Surface tension (N m(^{-1}))</th>
</tr>
</thead>
<tbody>
<tr>
<td>Air–water</td>
<td>1–2</td>
<td>998</td>
<td>1.2–2.4</td>
<td>0.001</td>
<td>1.8 x 10(^{-5})</td>
<td>0.073</td>
</tr>
</tbody>
</table>

The physical properties of the fluid pairing are presented in Table 1. The air–water mixture has been chosen because of the minimal risk involved and its ease of handling whilst providing valuable flow pattern information. The conductance technique has been chosen to measure void fraction because water is an electrical conductor, albeit a poor one, while air is essentially resistive. In this technique, the cross-sectional averaged void fraction can be estimated once the relationship between electrical impedance and phase distribution has been established. Andreussi et al. (1988) and Tsochatzidis et al. (1992) are some of the researchers to have used this method successfully. The probes consist of two stainless steel ring electrodes mounted in acrylic resin housing (Figure 3). They were carefully manufactured so that the electrodes had the same diameter, $D_e$, as the test section (5 mm). The distance between the electrode plates, $D_e$, and width of each plate, $s$, are 1.7 and 0.5 mm respectively. The electronic circuitry employed for the conductance probe is similar to that of Fossa (1998) and Fossa and Guglielmini (1998).

The probes give a voltage output ($V_{out}$) which is proportional to the resistance of the two-phase mixture. This response is converted to dimensionless conductance ($G_e$) by referring to the value obtainable when the pipe is full of liquid ($V_{full}$). To account for the variation in water resistance, $V_{full}$ was measured before every experimental run and adjusted if a considerable drift was observed. The calibration procedure involved artificially creating instantaneous liquid fractions over the conductance probes using plastic plugs with cylindrical rods of known diameters. $G_e$ is determined from the voltage responses, $V_{out}$ and $V_{full}$ and correlated with the corresponding liquid/void fractions. A unique third order polynomial is then obtained for each probe with $G_e^*$ as a function of void fraction (Figure 4).

The measurements from the probes are acquired using a PC installed with a National Instrument (NI) DAQ card. A modified version of the Labview program by Guglielmini (2002) was used to sample and convert the voltage signals to void fraction. Cross correlation of consecutive time-series yielded vital structure velocity information. Sampling rates of 1 and 5 kHz were used with data being taken for 60 and 12 s respectively.

Although flow pattern observation with the naked eye is the simplest method, discrimination of flow features proves more difficult at higher flow rate. A KODAK HS 4540 High Speed Video System was positioned at the test section so that the flow pattern and related structures in the two-phase flow could be observed. Video footage for varying flow conditions corresponding to slug, churn and annular flow patterns were recorded for detailed study in slow motion. The video signals were downloaded both as a sequence of digital still pictures and in analogue VHS format. Proper illumination from a halogen lamp was necessary in order to obtain good quality pictures.

**RESULTS**

**Void Fraction**

The mean void fraction is plotted against the gas superficial velocity for constant liquid superficial velocity in Figure 5. As expected, the mean void fraction generally increases with gas flow rate. At the liquid superficial velocities of 0.1 m s$^{-1}$ and 0.4 m s$^{-1}$, we observe that an increase in pressure from atmospheric to 2 bara results in a decrease in mean void fraction for low gas superficial velocities but...
The effect of pressure is studied for several gas flow rates at two liquid velocities, i.e., 0.1 and 0.4 m s\(^{-1}\). An increase in the pressure from 1.1 to 2 bara, results in a slight reduction in the structure velocity whilst retaining the characteristic change in slope.

The time varying void fraction data show the simultaneous occurrence of huge waves and liquid slugs (Figure 9), confirming the findings made by Sekoguchi and Mori (1997) using advanced measuring devices for longitudinal distributions of the liquid hold-up along a tube. The void fraction of a structure must equal zero to be classified as a liquid slug.

**Visualization**

The need for a detailed flow visualization scheme arose when initial attempts to describe the flow by observations with the naked eye proved inconclusive due to the two-phase flow mixture moving too fast. Slug, churn and annular flows were observed with high speed video pictures at 1.1 bara, a constant liquid superficial velocity of 0.1 m s\(^{-1}\) for gas superficial velocities of 0.12, 1.02, 1.43, 2.00, 4.07, 6.10 and 20.3 m s\(^{-1}\). The observations were made at a speed of 750 frames s\(^{-1}\).

At the lowest gas flow rate, the flow is characterized by Taylor bubbles (whose diameters approximate to the tube diameter) followed by non-aerated liquid slugs. A thin liquid film present on the pipe wall appears to travel counter-current to the gas flow.

Increasing the gas flow rate to 1.02 and 1.43 m s\(^{-1}\), the flow behaviour is essentially the same albeit the liquid slugs occur less frequently. When the gas superficial velocity reaches 2 m s\(^{-1}\), the liquid slug becomes aerated and although the flow pattern is still slug flow, the rate of falling liquid on the wall is very much increased. At about 4 m s\(^{-1}\), features of churn flow are present. The process of a falling liquid film levitated only for limited height by the gas phase is repeated until the gas core possess sufficient momentum to transport the accumulated liquid further downstream. At time intervals averaging 0.8 s, a fast moving chaotic two phase mixture akin to a highly aerated liquid slug body is observed passing through the test section. At a gas velocity of 6.1 m s\(^{-1}\), the churn flow is faster and the occasional swift liquid body bears fewer semblances with a liquid slug. The characteristic oscillatory motion is a major feature of the flow. Increasing the gas velocity to 20.3 m s\(^{-1}\) gives fully developed annular flow. Most of the liquid phase travels on the tube wall and occasional bridging of the pipe by the liquid phase can be seen.

**Probability Density Function**

PDF is the rate of change of the probability that void fraction values lie within a certain range versus void fraction. The total area under the probability density function must equal unity. The probability density function is determined using the `hist` command in MATLAB.

The PDF of time varying void fraction has been used to classify the flow patterns in the same manner as Costigan and Whalley (1997). Typical examples for slug, churn and annular flows are given in Figure 10. This is for a liquid superficial velocity of 0.65 m s\(^{-1}\) and gas superficial velocities of 0.15, 1.05, 5.22 and 23.9 m s\(^{-1}\) respectively at 1.1 bara. The slug flow regime is characterized by twin peaks
corresponding to the liquid slug and the Taylor bubble. A single peak existing at a high void fraction is a feature of annular flow. In addition to a high void fraction peak, churn flow possesses a tail extending to low void fractions.

COMPARISONS WITH PREVIOUS WORK

In the present experiments, some void fraction measurements were taken at exactly the same conditions as those of Holt et al. (1999), which were taken at 1.5 bara in a 5 mm diameter vertical tube for liquid superficial velocities of 0.02, 0.08 and 0.12 m s\(^{-1}\). The results from both sources are given in Figures 11–13. The conductance probe technique was employed in our experiments whilst Holt et al. (1999) used the quick closing valves method. Most of the data lies in the annular flow regime, where the liquid phase can be transported both as a thin film on the pipe wall and as entrained droplets in the liquid core. The droplets in the core are insulated from the liquid film by the non-conducting air phase, and are therefore not accounted for by the conductance technique. Hence, the experimental void fraction (or more appropriately film fraction) obtained through conductance is expected to be greater than the actual void fraction measured using quick closing valves for annular flow where there is liquid entrainment.

At the lowest liquid superficial velocity of 0.02 m s\(^{-1}\), the flow pattern obtained is annular. Figure 11 shows that the mean void fractions from both sets of results are indistinguishable. This means that the entrainment of the liquid phase in the gas core is negligible. Confirmation is found in the requisite critical liquid film flux predicted by the relationship of Govan et al. (1988) for the onset entrainment. For these experiments, entrainment is predicted to occur at liquid superficial velocities of 0.084 m s\(^{-1}\) and above. When the liquid flow rate is increased to 0.08 m s\(^{-1}\), churn and annular flows are present. Figure 12 shows that at the higher gas flow rates where annular flow exists, the mean void fraction from Holt et al. (1999) is in good agreement with present experiments. However, at gas velocities below 10 m s\(^{-1}\), the values differ considerably. A possible reason for this is that critical liquid film flux which predicts the onset of entrainment is only applicable to annular flows. Figure 13 gives the data obtained at a liquid velocity of 0.12 m s\(^{-1}\). The correlation of Okawa et al. (2004) is used to estimate the equilibrium entrainment fraction, which is subtracted from the experimental void fraction to obtain an actual corrected void fraction value. Figure 13 shows that the correlation underpredicts the void fraction at gas velocities of up to 10 m s\(^{-1}\) and gives good agreement between 10 and 40 m s\(^{-1}\) whilst overpredicting the entrainment above 40 m s\(^{-1}\).
The reason for this underprediction is because the entrainment correlation has been developed for annular flows but the flow pattern at low gas superficial velocities is either slug or churn.

**DISCUSSION**

**Flow Pattern Transitions**

The comparisons of flow patterns identified using PDFs with observations from a high speed video camera for a constant liquid superficial velocity of 0.1 m s\(^{-1}\) at 1.1 bara are given in Table 2. The PDF plots agree well with visual observation of flow pattern transitions except for cases of slug flow near the boundary with churn flow. The presence of different structures within a given flow pattern might be responsible.

The empirical correlations by Barnea et al. (1983) for the slug-to-churn and churn-to-annular flow transitions and the bubble-to-slug flow transition of Taitel et al. (1980) are compared with our results at 1.1 bara in Figure 14. The transitions by Barnea et al. (1983) perform well for the slug-to-churn flow transition at low liquid flow rates only. It is less reliable for the transition to annular flow.

The correlation of Taitel et al. (1980) which assumes that the transition from bubble to slug flow occurs at a critical void fraction of 0.25 is in poor agreement with our experiments, as can be seen in Figure 14. Bubble coalescence which is the mechanism for this transition increases sharply beyond a certain critical void fraction.

Hence, an accurate estimate of the critical void fraction is essential for the correct prediction of the bubble-to-slug flow transition. Recent work by Song et al. (1995), Cheng et al. (2002) and Guet et al. (2002) on the effect of bubble size show that an inverse relationship exists between the critical or maximum voidage for bubble flow and the bubble/pipe diameter ratio (dimensionless bubble size) (Figure 15). Assuming a constant bubble size, the relationship suggests that critical voidage reduces as pipe diameter is decreased. Omebere-Iyari (2006) provides an explanation for this.

![Figure 11](image1.png)

Figure 11. Mean void fractions at \(U_{LS}\) of 0.02 m s\(^{-1}\).

![Figure 12](image2.png)

Figure 12. Mean void fractions at \(U_{LS}\) of 0.08 m s\(^{-1}\).

![Figure 13](image3.png)

Figure 13. Mean void fraction at \(U_{LS}\) of 0.12 m s\(^{-1}\) (corrected for entrainment).

<table>
<thead>
<tr>
<th>(U_{LS}) (m s(^{-1}))</th>
<th>PDF</th>
<th>Flow pattern (Costigan and Whalley, 1997)</th>
<th>Flow pattern observed</th>
</tr>
</thead>
<tbody>
<tr>
<td>0.1</td>
<td>Slug</td>
<td>Slug</td>
<td>Slug</td>
</tr>
<tr>
<td>0.36</td>
<td>Churn</td>
<td>Slung</td>
<td>Churn</td>
</tr>
<tr>
<td>4.4</td>
<td>Churn</td>
<td>Churn</td>
<td>Churn</td>
</tr>
<tr>
<td>10.7</td>
<td>Annular</td>
<td>Annular</td>
<td>Annular</td>
</tr>
</tbody>
</table>

Table 2. Flow pattern identification from PDFs and visual observations.
behaviour using phase distribution in bubble flow. In essence, it has been shown through experimentation that there is an increased wall peak distribution when the dimensionless bubble size is large (as with smaller tubes). This accelerates the rate of bubble coalescence and increases the chances of creating large bubble at the wall before dispersion to the core. If a sufficient number of large bubbles are generated in the wall region some of them can reach the core without breakup and form the Taylor bubbles associated with slug flow. It appears therefore that larger dimensionless bubble sizes are more favourable towards slug flow formation and explains the reduction in the critical voidage for bubbles of this magnitude. Assuming the critical void fraction to be 0.05, the relationship of Taitel et al. (1980) gives an improved prediction for the transition to slug flow for our data as shown in Figure 16.

The changes in the slope of structure velocity obtained from cross correlation delineate the slug, churn and annular flow patterns very well at 1.1 bara based on comparisons with pictures from the high speed video camera. Visual observations also show that for the churn/annular flow transition, the theoretical relationship of Barnea (1986) is less reliable at the conditions examined.

At a liquid superficial velocity of 0.1 m s\(^{-1}\) and gas superficial velocity of 6.1 m s\(^{-1}\), the transition of Barnea (1986) wrongly predicts the existence of annular flow (Figure 16). The slug-to-churn flow transition by Jayanti and Hewitt (1993) which gives flooding as the mechanism for the transition is in good agreement with our transition using structure velocity.

The effect of pressure on flow pattern transitions is illustrated in Figure 17 using structure velocity. The data at 2 bara shows the same characteristic change in structure velocity as the 1.1 bara data (Figure 8) and can therefore be used to delineate the flow pattern transitions at 2 bara. The slug/churn flow transition moves to lower gas superficial velocities as the pressure is decreased from 2 to 1.1 bara. This results in a larger slug flow and smaller churn flow region at the higher pressure. The location of the churn/annular flow transition is less dependent on system pressure.

Our use of structure velocity for flow pattern discrimination is advantageous due to its objectivity. In addition, the reasonable success of the aforementioned method has shown that changes in the characteristics of flow structures are associated with flow pattern transitions and may be used for further study of flow pattern transition mechanisms.

**Void Fraction Prediction**

The mean void fraction values were compared to the flow pattern specific method described earlier (Figure 18). A statistical analysis of its performance is presented in Table 3. The terms, \(F\) and \(S\), which are the correction factor and the transformed standard deviation respectively, were shown to be

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**Figure 14.** Flow pattern transitions at 1.1 bara.

**Figure 15.** Critical void fraction for various bubble sizes.

**Figure 16.** Improved flow pattern transitions at 1.1 bara.

**Figure 17.** Effect of pressure on the experimental slug/churn and churn/annular flow transitions.
useful analytical tools by Govan (1988). \( F \) is defined as the average factor by which the calculated value must be multiplied to give the experimental value, \( 1/\exp(\mu) \); and \( S \) is given by \( \exp(\sigma) - 1 \).

\[
\sigma = \sqrt{\frac{1}{n} \sum_{i=1}^{n} (\varepsilon(i) - \mu)^2} \quad (26)
\]

\[
\varepsilon(i) = \log\left( \frac{P(i)}{E(i)} \right) \quad (27)
\]

\[
\mu = \frac{1}{n} \sum_{i=1}^{n} \varepsilon(i) \quad (28)
\]

\( P(i) \) is the predicted or calculated void fraction and \( E(i) \) is the experimental void fraction.

The slug flow model is applied to data identified as slug and churn flow and the annular model to experiments identified as cases of churn and annular flow. Flow pattern identification is performed using the methods we found to be most reliable. These methods are the modification to Taitel et al. (1980) for the bubble/slug transition and the use of structure velocity for the slug/churn and churn/annular flow transitions.

The void fraction predictions are marginally better when the slug flow model is employed for churn flow than with the annular flow model (Table 3). This can be explained by the fact that the huge waves in churn flow behave more like liquid slugs while annular flow with smaller drop and waves is significantly different. Overall, the flow pattern specific method tends to underpredict the data especially for the slug flow experiments. We attribute this to the fact that the slug flow model employs a correlation for the slug velocity only (Nicklin et al., 1982). The motion of other structures such as huge waves which we observed in fully developed slug flow is not accounted for (Figure 9). It is likely that the velocities of the huge waves differ significantly from those of the liquid slugs. The model could therefore be improved by using a robust method which takes the velocities of the various structures in the flow into consideration.

### Disturbance Wave Velocity

Azzopardi (1997) in his extensive review of drops in annular flow indicates that in spite of the importance of waves, there has been limited amount of theoretical work in comparison with the experimental aspect. Of the correlations discussed by Azzopardi (1997) for the film wave velocity in annular flows, the work by Pearce (1979) based on simplified equations of motion for the film and vapour core is tested against our data. Pearce (1979) correlated the film wave velocity using

\[
u_w = \frac{K u_\theta + u_{gt} \sqrt{\rho_g/\rho_l}}{K + \sqrt{\rho_g/\rho_l}}
\]

(29)

When tested against data from a horizontal tube of 31.8 mm diameter and using \( K = 0.8 \), the calculated wave velocity showed good agreement with the measured wave velocity obtained by cross correlation. The time-averaged film velocity \( \bar{u}_f \) is determined from

\[
\bar{u}_f = \frac{D \times m_f}{4 \times \rho_f \times d}
\]

(30)

The liquid film flow rate in annular flow, \( m_f \) is calculated using the method suggested by Hewitt and Govan (1990). Here, the expression below containing the entrainment and deposition correlations is integrated:

\[
\frac{d m_f}{d z} = \frac{4}{d} (m_d - m_e)
\]

(31)

When applying the relationship of Pearce to our data, the film thickness is computed by using the experimental void fraction (assuming the liquid film is symmetric around the wall). The best fit after optimization is obtained with \( K = 0.51 \) (Figure 19). Several data points have been ignored due to very noisy cross correlation signals. The wave velocity correlation is applied to other data obtained from different conditions.
pipe diameters. Figures 20 and 21 highlight the effect of pipe diameter on $K$. The 10 and 32 mm data are from Willets (1987) and Wolf (1995) respectively, and the 25 mm and 42 mm data are sourced from Schadel (1988). The value of $K$ is about 0.9 for pipes with diameters in the range of 25–42 mm. However, $K$ decreases to 0.61 and 0.51 as the diameter is reduced to 10 mm and 5 mm, respectively.

To explain the decrease of $K$ with pipe diameter, an understanding of the film thickness structure in vertical annular flow and the physical definition of $K$ is essential. The film thickness wave in vertical annular flow is described in Figure 22. $K$ and the mean film thickness are given by equations (32) and (33).

$$K = \frac{\bar{a}}{\sqrt{\bar{a_p}} \, a_f} \tag{32}$$

$$\bar{a} = a_f (1 - I) + a_p j \frac{a_f}{a_p} \tag{33}$$

where $j$ is the factor that accounts for the non-uniformity of the disturbance wave, $I$, the intermittency is given by $w/\lambda$.

Visualization studies by Hewitt and Lovegrove (1969) in a 32 mm diameter tube show that disturbance waves appear as coherent bands. However, Azzopardi et al. (1983) observed the waves to be circumferentially localized in a large tube having a diameter of 125 mm. Martin and Azzopardi (1985) have indicated that the breakdown in coherence is systematic with pipe diameter. This increased coherence in smaller pipes means that fewer imperfections exist on the liquid film circumference and translates to the increased growth of the disturbance wave peaks, $a_p$, given that a smaller amount of sites are available for atomization. From the work of Schadel (1988), the mean film thickness and intermittency for similar gas and liquid flow rates are found to be independent of pipe diameter. This implies that the increase in the disturbance wave peak, $a_p$, must correspond to a decrease in the wave trough or base film, $a_f$. Assuming that the wavelength is at least twice as large as the disturbance wave width and the volume of liquid contained in the disturbance wave is smaller than that which forms the rest of the film, the reduction in $a_f$ will be small in comparison to the increase in $a_p$. From the definition of $K$ in equation (32), the decrease of $K$ with pipe diameter is therefore reasonable. Support for our findings can be found in the autospectral density measurements of Martin and Azzopardi.
CONCLUSIONS

Conductance probes have been successfully used to acquire valuable time-varying void fraction data and flow pattern information in a 5 mm diameter pipe. The use of high speed video and PDF plots have enabled the study of flow pattern transitions. A novel approach has been developed for determining slug-to-churn and churn-to-annular flow transitions in vertical two phase flow based on changes in the structure velocity. The PDF method of Costigan and Whalley (1997) fails to delineate the exact location of the slug/churn transition accurately. The Jayanti and Hewitt (1982) slug/churn flow transition model is reliable at the conditions studied. The theoretical correlation of Barnea (1986) performs remarkably better than the empirical one of Barnea et al. (1983) for churn/annular flow transition. It is shown that better predictions of the bubble-to-slug flow transition in small passages is obtained by modifying the relationship of Taitel et al. (1980) to account for the variation of the critical void fraction with bubble size.

A flow pattern specific method modelled on slug and annular flow gives good predictions of void fraction. The slug flow model performs better than the annular flow model for the prediction of void fraction in churn flow. We believe that huge waves in churn flow being more similar in behaviour to liquid slugs than they are to the drops and waves in annular flow provides an explanation.

The relationship for the velocities of disturbance waves on the wall film in annular flow by Pearce (1979) is applied to both our data and other experiments performed in pipes with diameters ranging from 5 mm to 42 mm. The correlating constant, K increases sharply with pipe diameter before attaining a steady value of 0.9.

NOMENCLATURE

\( \bar{a} \) \hspace{1cm} \text{mean film thickness, m} \\
\( Bo \) \hspace{1cm} \text{Bond number \textit{[defined in equation (1)]}} \\
\( C \) \hspace{1cm} \text{droplet concentration in the gas core, kg m}^{-3} \\
\( D \) \hspace{1cm} \text{tube diameter, m} \\
\( E \) \hspace{1cm} \text{entrainment fraction} \\
\( f_f \) \hspace{1cm} \text{interfacial friction factor} \\
\( f_w \) \hspace{1cm} \text{wall friction factor} \\
\( F \) \hspace{1cm} \text{correction factor} \\
\( j \) \hspace{1cm} \text{non-uniformity factor \textit{[in equation (32)]}} \\
\( l \) \hspace{1cm} \text{intermittency} \\
\( k \) \hspace{1cm} \text{mass transfer coefficient, m s}^{-1} \\
\( K \) \hspace{1cm} \text{correlating parameter for wave velocity} \\
\( m \) \hspace{1cm} \text{mass transfer rate or flux, kg m}^{-2}s^{-1} \\
\( n \) \hspace{1cm} \text{power of density ratio} \\
\( N_0 \) \hspace{1cm} \text{dimensionless inverse viscosity \textit{[defined in equation (10)]}} \\
\( P \) \hspace{1cm} \text{pressure, bar} \\
\( r \) \hspace{1cm} \text{function of } N_0 \text{ \textit{[in equation (16)]}} \\
\( S \) \hspace{1cm} \text{transformed standard deviation} \\
\( u, U \) \hspace{1cm} \text{cross-sectional area-averaged velocity, m s}^{-1} \\
\( \bar{u} \) \hspace{1cm} \text{time-averaged film velocity, m s}^{-1} \\
\( \bar{u}_o \) \hspace{1cm} \text{rise velocity of a Taylor bubble in stagnant fluid} \\
\( z \) \hspace{1cm} \text{channel length, m} \\

Greek symbols

\( \varepsilon \) \hspace{1cm} \text{void fraction} \\
\( \sigma \) \hspace{1cm} \text{ratio of interfacial force to surface tension} \\
\( \rho \) \hspace{1cm} \text{density, kg m}^{-3} \\
\( \sigma \) \hspace{1cm} \text{surface tension, N m}^{-1} \\

Subscripts

\( d \) \hspace{1cm} \text{deposition} \\
\( e \) \hspace{1cm} \text{entrainment} \\
\( fs \) \hspace{1cm} \text{superficial liquid film} \\
\( g \) \hspace{1cm} \text{gas phase} \\
\( gs \) \hspace{1cm} \text{superficial gas phase} \\
\( l \) \hspace{1cm} \text{liquid phase} \\
\( f \) \hspace{1cm} \text{liquid film} \\
\( ls \) \hspace{1cm} \text{superficial liquid phase} \\
\( p \) \hspace{1cm} \text{peak} \\
\( s \) \hspace{1cm} \text{structure} \\
\( t \) \hspace{1cm} \text{trough} \\
\( w \) \hspace{1cm} \text{wave} \\

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