VALIDATION OF THE USE OF AIR/WATER IN SIMULATING BUBBLY STEAM/WATER FLOWS

BY
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ABSTRACT

Many studies with two-phase flows assume similar hydrodynamic characteristics for air/water and steam/water mixtures, usually because of the simplicity of air/water systems compared to the experimental difficulties associated with steam/water mixtures. However it is questionable whether this is an equivalent substitution. Recent work at Surrey University (Unis) has shown that mechanical agitation of boiling water requires dramatically more power than for air/water mixtures. Similar discrepancies have also been reported in studies of pressure drop, transport phenomena in flow boiling and interactions between air/steam and water flows in turbulent two-phase flow by several other investigators. Such distinctive behaviour has been regarded as resulting from phase changes the different physical properties of the two systems or different hydrodynamic characteristics when flow passes an obstruction. Despite these differences, no fundamental work on this subject appears to have been reported.

The aim of the present study is to put this controversy into perspective and also to provide effective guidance for design. Accordingly, this work involves both experimental and theoretical approaches. This begins with a literature survey which highlights the effective parameters which may be involved. An experimental test rig has also been constructed for both upward vertical and horizontal flows. A comprehensive set of experiments has been done with steam/water and air/water flows at ambient and elevated temperatures. The experiments at elevated temperature are carried out to discern how two-phase mixtures behave in the intermediate range from ambient temperature to boiling point. The results highlight a significant effect of temperature on the void fraction with respect to inlet gas flow rate, especially at higher liquid flow rates. The effect of temperature is more dominant nearer to boiling point, particularly when two-phase flow passes through a venturi. A modified volumetric flow ratio is proposed which includes the influence of vapour pressure and hydrostatic pressure on the void fraction at higher temperatures. The steam/water experimental data are also presented with a phenomenological comparison with the air/water results.

A new one-dimensional theoretical model is presented along with the numerical results from CFD, aiming to predict void fraction and pressure gradient along the pipe in a
vertical tube. The model is explicit and simple in form but is realistic, with reasonable accuracy compared with the experimental results. It also establishes the point in the tube at which the boiling starts. The characteristics of the model compared with those of air/water are found to be substantially different, particularly nearer to the bottom of the tube.
ACKNOWLEDGEMENTS

I must open by sincerely thanking my supervisors, Professor John M. Smith and Professor Hans Müller-Steinhagen.

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## NOMENCLATURE

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**GREEK LETTERS**

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eff  effective

g  gas

G  gas

Ga  gas-atmospheric

i  phase

l  liquid

L  liquid

La  liquid-atmospheric

m  mixture

O  boundary condition at a specified point along the tube

s  steam

wall  wall

x  differentiation with respect to the distance from top of the tube

ABBREVIATIONS

AWS  Air/Water/Steam

GD  Gamma Densitometer

LASER  Light Amplification by Stimulated Emission of Radiation

LDV  Laser Doppler Velocimetry

RPD  Relative Power Demand
LIST OF FIGURES AND TABLES

Figure 2.1  Flow patterns in vertical pipes
Figure 2.2  Flow patterns in horizontal pipes
Figure 2.3  The assumed differences between air and steam bubbles (Wadckar, 1997)
Figure 2.4a Relative power demand as a function of impeller speed for a Rushton turbine of 180 mm diameter in the steam/water mixture (Smith and Katsanevakis, 1993)
Figure 2.4b Relative power demand as a function of impeller speed for a Rushton turbine of 176 mm diameter in the air/water mixture (Smith and Katsanevakis, 1993)
Figure 2.5  Comparison of film thickness in air/water and steam/water flows at different axial position (Lineham, 1968)
Figure 2.6  Recordings of liquid film thickness signals in steam/water and air/water flows (Nakanishi et al., 1993)
Figure 2.7  Mean spatial wave separation. Comparison of steam/water and air/water (Nakanishi et al., 1993)
Figure 2.8  Total damping as a function of void fraction (Axisa et al., 1988)
Figure 2.9  Illustration of bubble behaviour due to lift force in upward vertical flow
Figure 2.10 Possible phase distribution pattern (Serizawa et al., 1975)
Figure 3.1  Schematic diagram of the AWS test rig
Figure 3.2  AWS experimental rig
Figure 3.3  Traverse apparatus of the gamma densitometer
Figure 3.4  3 different kinds of gas spargers used in this study
Figure 3.5  heater used in vertical section
Figure 3.6  Schematic diagram of venturi sections
Figure 3.7  Venturi used in this study
Figure 3.8  Gamma ray transmitted through the pipe
Figure 3.9  Schematic diagram of the gamma densitometer detectors
Figure 3.10 Fringe movement in a frequency-shifted system
Figure 3.11 Polycarbonate box for reduction of refraction
Figure 4.1  Effect of inlet gas flow rate on void fraction at different liquid flow rate
Figure 4.2  Effect of liquid flow rates on the variation of void fraction in vertical flows
Figure 4.3  Effect of inlet volumetric flow ratio on the void fraction
Figure 4.4  Effect of change of gas and liquid flow rates on void fraction at two constant inlet volumetric flow ratios
Figure 4.5  Effect of temperature on the void fraction in vertical direction at low liquid flow rate
Figure 4.6  Effect of temperature on the void fraction in vertical direction at high liquid flow rate
Figure 4.7  Variation of void fraction as function of temperature at constant volumetric flow ratio
Figure 4.8  Variation of void fraction as function of temperature at constant volumetric flow ratio
Figure 4.9  Variability of the operational parameters during the experiment of which the data are shown in Fig. 4.8
Figure 4.10 Effect of liquid flow rate on the void fraction at different temperature
Figure 4.11 Comparison of the void fraction at different axial positions as a function of temperature
Figure 4.12 Spatial evolution of the flow structure at different gas flow rates and at room temperature
Figure 4.13 Spatial evolution of the flow structure at different temperatures
Figure 4.14 Schematic diagram of bubble shape and size at ambient and high temperatures
Figure 4.15 Mean, minimum and maximum bubble size as a function of temperature
Figure 4.16 Effect of temperature on the spatial flow structure at constant void fraction
Figure 4.17 Changes in bubble size in boiling water
Figure 4.18 Local liquid velocity of liquid single phase as a function of r/R ratio
Figure 4.19 Effect of gas flow rate on the local liquid velocity
Figure 4.20 Effect of temperature on the local liquid velocity at a given void fraction
Figure 4.21 Void fraction as a function of gas flow rate at ambient temperature in horizontal flow
Figure 4.22  Effect of liquid flow rate on void fraction at different inlet gas flow rates
Figure 4.23  Effect of liquid flow rate on void fraction as a function of experiment number at different temperature
Figure 4.24  Effect of liquid flow rate on void fraction at different temperature
Figure 4.25  Effect of temperature on the variability of the void fraction
Figure 4.26  Effect of liquid flow rate on void fraction at constant gas flow rate at ambient temperature
Figure 4.27  Effect of liquid flow rate on the void fraction through venturi
Figure 4.28  Variation of void fraction through venturi at various gas flow rates
Figure 4.29  Variation of void fraction through venturi at different temperature
Figure 4.30  Variation of void fraction as a function of heat load in steam/water in vertical flow
Figure 4.31  Effect of heat load on the void fraction at different total flow rates
Figure 4.32  Influence of heat load on variation of void fraction with respect to the downstream distance
Figure 4.33  Nucleation curve for steam/water in vertical tube
Figure 4.34  Variation of void fraction with respect to heat load at different liquid flow rate
Figure 4.35  Comparison of air/water and steam/water two-phase flows pass through a venturi at a given liquid flow rate
Figure 5.1  Schematic diagram of the theoretical model, x (direction), a (atmospheric) and O (boundary)
Figure 5.2  Effect of liquid velocity on the variation of void fraction in a heated tube
Figure 5.3  Effect of heat load on the variation of void fraction in a heated tube
Figure 5.4  Variation of void fraction with respect to downstream distance measured at different heat loads
Figure 5.5  Comparison between experimental data and theory
Figure 5.6  Comparison of different models developed for air/water and steam/water in heated and unheated tubes
Figure 6.1  Variation of void fraction of air/water bubbly flow at different lateral sections
Figure 6.2  Comparison between numerical and experimental results
Figure 6.3  Void fraction distribution for air/water and steam/water bubbly flows
Figure 6.4  Comparison of void fraction distributions for air/water and steam water bubbly flows through a venturi at the throat and outlet points

LIST OF TABLES

Table 3.1 Specifications of the LDV system used in this study
Table 4.1 Physical properties of air/water at different temperature
Table 4.2 Changes in minimum, average and maximum bubble size corresponding to the data shown in Fig. 4.15
Table 4.3 The statistical void fraction results for different temperatures
Flowchart 3.1 Experimental procedure of the present study
TABLE OF CONTENTS

ABSTRACT .................................................................................................................... i
ACKNOWLEDGEMENTS .......................................................................................... iii
NOMENCLATURE ...................................................................................................... iv
LIST OF FIGURES AND TABLES ........................................................................... viii

CHAPTER 1     INTRODUCTION

1.1 The importance of two-phase systems in general ............................................... 1
1.2 Statement of the problem .................................................................................... 3
  1.2.1 Bubble dynamics in gas/liquid and vapour/liquid systems ......................... 3
  1.2.2 Aims and objectives ...................................................................................... 4
1.3 Scope of present work ....................................................................................... 5

CHAPTER 2     LITERATURE REVIEW

2.1 Flow regimes in two-phase flow ........................................................................... 7
2.2 Differences between air/water and steam/water systems ..................................... 10
2.3 Bubbly flow ........................................................................................................ 19
  2.3.1 Definition and importance .......................................................................... 19
  2.3.2 Vertical and horizontal bubbly flow .............................................................. 20
  2.3.3 Effect of bubble size on the hydrodynamics of bubbly flow ....................... 20
2.4 Dynamics of air and steam bubbles .................................................................... 22
2.5 Void fraction in bubbly flow .............................................................................. 23
  2.5.1 Void fraction distribution at different cross-sections for vertical and horizontal
       flows .................................................................................................................. 23
  2.5.2 Effect of elevated temperature on the void fraction .................................... 25
2.6 Geometrical effects ............................................................................................ 27
  2.6.1 Entrance effects .......................................................................................... 27
  2.6.2 Effect of roughness on developing vertical flow .......................................... 28
  2.6.3 Effect of flow obstructions ......................................................................... 28
2.7 Summary and conclusions ................................................................................. 29
CHAPTER 3 EXPERIMENTAL SET UP AND PROCEDURE
3.1 Experimental set-up ................................................................. 30
  3.1.1 Description of the ASW rig .............................................. 30
  3.1.2 Equipment for air/water experiments ............................... 33
  3.1.3 Equipment for steam/water experiments ............................ 33
  3.1.4 Description of the venturis ............................................. 34
  3.1.5 Gamma densitometer ...................................................... 36
  3.1.6 Laser Doppler Velocimeter (LDV) .................................... 38
  3.1.7 High speed video camera and image analysis .................. 41
  3.2 Experimental procedure .................................................... 42
  3.3 Data recording ................................................................. 42
  3.4 Error analysis ................................................................. 42

CHAPTER 4 RESULTS AND DISCUSSION
4.1 Void fraction in upward vertical air/water flow ....................... 45
  4.1.1 Void fraction at ambient temperature ............................... 45
    4.1.1.1 Effect of bubble size on the void fraction in vertical flow ........ 47
  4.1.2 Effect of elevated temperature on the void fraction .......... 49
  4.1.3 Void fraction at different axial positions ......................... 54
  4.1.4 Effect of temperature on the development of flow structure .. 55
  4.1.5 Bubble dynamics .......................................................... 61
  4.1.6 Laser Doppler Velocimeter measurement ......................... 63
  4.2 Void fraction in horizontal flow ......................................... 65
    4.2.1 Variability of the void fraction measurements ................. 68
  4.3 Effect of constrictions on the hydrodynamics of air/water ....... 69
  4.4 Steam/water system .......................................................... 73
  4.5 Conclusions ................................................................. 79

CHAPTER 5 THEORETICAL STUDY
5.1 Changes in void fraction in vertical saturated vapour/liquid flows in a heated tube ................................................................. 81
  5.1.1 The nine variables and nine equations ............................. 83

xiii
5.1.2 Deriving an expression for the local void fraction in terms of the local pressure ........................................................................................................... 89

5.1.3 Changes in pressure of saturated vapour/liquid upward flows in vertically heated pipes ........................................................................................................... 95

5.2 Changes in void fraction due to changes in pressure for saturated vapour/liquid flows in an unheated tube ........................................................................ 102

5.2.1 Validation of the model with comparison of experimental data .......... 103

5.3 Comparison between air/water and steam/water flows ......................... 105

5.4 Summary and conclusions .............................................................................. 106

CHAPTER 6 NUMERICAL STUDY

6.1 Previous works ................................................................................................. 108

6.2 Mathematical model .......................................................................................... 109

6.2.1 Governing equations ......................................................................................... 109

6.2.2 Interfacial transfer terms ................................................................................... 110

6.3 Numerical method ............................................................................................. 112

6.4 Results and discussion ..................................................................................... 113

6.4.1 Air/water and steam/water flows in equal diameter pipes........................... 113

6.4.2 Flow in venturi pipes ....................................................................................... 118

CHAPTER 6 CONCLUSIONS AND FUTURE WORK

7.1 Conclusions ......................................................................................................... 118

7.2 Future work ......................................................................................................... 121

REFERENCES ........................................................................................................ 124

APPENDIX A

APPENDIX B

APPENDIX C

APPENDIX D

APPENDIX E
CHAPTER 1

INTRODUCTION
INTRODUCTION

1.1 The Importance of Two-Phase Systems in General

Two-phase flow involving a mixture of gas or vapour and liquid or gas/liquid and solid is very common in industrial operations. It can be said that two-phase flows make up about one-half of all industrial, biological and environmental flow conditions (Wallis, 1996). The application of two-phase flow began in the 17th century, before the industrial revolution, when an air-lift system was used to compress the air for iron and brass smelting processes. Today, applications of two-phase flow are found in many areas of science. Intensive study of two-phase flows was initiated for nuclear engineering and oil and gas transportation and has later been extended to mass and heat transfer in separation processes and reactions in mixing tanks. From the hydrodynamic point of view, accurate prediction of two-phase flow behaviour is indispensable for the design of heat exchangers and the safety devices required for hazardous systems. The widespread application of two-phase flow has led to intensive studies in this field. Gouse (1966) showed that the number of publications in this field has approximately doubled every five years.

In spite of the importance of two-phase flows in industrial processes, performance predictions in this field are often reported to be in error by 40% (Chisholm, 1983 and Whalley, 1996). In a keynote address, Hewitt (1996) suggested that despite the achievements in the past decades, still lots need to be done in characterising two-phase flows. The difficulties associated with investigations in this field can be attributed to two main factors. The first point is that two phase flows are very complicated in their own right and the presence of the two deformable phases can impede an accurate measurement or prediction of relevant parameters. Accordingly, it can be said that the major errors in the calculation of two-phase flow are systematic. The following reasons lie behind this:

1) there are many variables in two-phase gas/liquid systems, including:
   1-i) physical properties of both phases;
   1-ii) independent operational variables, i.e. phase flow rates, temperature and pressure:
The Importance of Two-Phase Systems in General

1-iii) the orientation and inclination of the pipe, wall roughness and the nature of the phase distribution;
2) the deformable nature of the interfacial area between the phases prevents the accurate prediction of transport phenomena between the phases;
3) In many cases, the relationship between these variables is poorly understood.

The second source of error in describing two-phase flow might be caused by ignoring the differences between vapour/liquid and gas/liquid mixtures. Many designs for vapour and liquid flows are based on data from gas/liquid systems at ambient temperature. This may be a reflection of the intensive study of air/water in past years. It may be useful to consider some examples in which the gas/liquid analogy has been used in simulating vapour/liquid systems.

Hashizume and Ogiwara (1987) compared results in a huge data bank, including air/water and steam/water data, with a specific pressure drop correlation based on air/water flow. It was shown that there is generally good agreement between the experimental data and the predicted values for horizontal pipes. A similar comparison has been made in correlating steam/water void fraction by using air/water correlations (Chisholm, 1983). Hewitt and Roberts (1969) also emphasised that the flow pattern map fits reasonably well for both steam/water and air/water systems over a limited range of parameters, in particular for small tubes. Furthermore, Reimann and John (1978) showed that in recognising flow pattern in two-phase flows, the void fraction profiles for horizontal air/water and steam/water flows agree fairly well at similar values of the superficial velocities.

In steam generation and nuclear plants, as pointed out by Awwad et al. (1995), the majority of investigations have been performed using air/water. Moreover, the application of air/water based correlations has been extended to many areas of boiling, including pool and flow boiling. Air/water has frequently been used in such work because of its simplicity, despite the important differences in the interaction between bubbles and liquid which often involve heat and mass transfer. It has sometimes been stated that in the absence of phase change similar physical laws can be applied to gas/liquid and vapour/liquid systems (Chisholm, 1983).
The following conclusions can be drawn from the above:

1) in many investigations, the differences between air/water and steam/water have not been taken into consideration, implying a belief that the two systems would have similar dynamic and fluid mechanical behaviour;

2) it is important to note also that air/water has sometimes been used because of the inherent difficulties of using vapour/liquid in a hazardous system, e.g. in flow boiling when there may be problems with undamped vibration phenomena.

Despite the similarities assumed by many investigators, some work clearly shows differences between steam/water and air/water which cannot be ignored. These differences arise principally from:

1) the different physical properties of the two systems;

2) phase changes;

3) differences in the response of the respective two-phase flows when passing a constriction.

It can be concluded that a better understanding of the behaviour of the two distinctly different systems is essential for process design. A literature survey concentrating on this aspect will be presented in more detail in the next chapter.

1.2 Statement of the Problem

This study is intended to investigate under which circumstances the similarities between the two systems can or cannot be assumed. Any differences between air/water and steam/water flows are most likely to be evident in bubbly flow. Bubbly flow has the advantage of simplicity over other flow patterns since it is possible to characterise e.g. the bubble size, in ways that make it attractive. The choice of bubbly flow for investigation is made despite the accepted importance of the other flow regimes like churn and annular flow.

1.2.1 Bubble Dynamics in Gas/Liquid and Vapour/Liquid Systems

It seems from the above that a useful first step in dealing with the differences between gas/liquid and vapour/liquid flows would be to determine and describe bubble behaviour
in the two systems. Therefore, it is instructive to consider some distinct features of each system:

1) air bubbles, in contrast to those of steam, are essentially insoluble in the water phase, particularly at the ambient temperatures and pressures used in most experiments;
2) the resistance to mass and heat transfer between the bubbles and the liquid will be greater with gas than with vapour;
3) the dynamic response of vapour bubble to any change in temperature or pressure is different from that of air bubbles (stated by Smith and Millington, 1995);
4) the steam bubbles will be more affected than air bubbles by any obstruction to flow in the pipe.

These differences between gas/liquid and vapour/liquid systems should guide both the design of experiments and the development of theoretical models.

1.2.2 Aims and Objectives

The purpose of the current study is to put the differences between air/water and steam/water systems into perspective with experimental and theoretical studies.

The experimental work has validated a numerical simulation that provides a basis for design. Accordingly, the following were the objectives of the experimental study:

1) to establish values for the characteristic steady state void fractions of air/water and steam/water mixtures at corresponding phase flow rates in vertical and horizontal cocurrent flows;
2) to determine differences in the pressure field for developing flow conditions as steam/water or air/water mixtures of similar void fraction flow through a constriction;
3) to observe the differences in the bubble spatial shape in air/water and steam/water flows along the pipe;
4) to establish the effects of flow obstructions on the flow structure of the two systems.

The complementary theoretical and analytical programme has had the following aims:
1) to correlate the information in the available experimental data, where possible, to characterise the effect of system parameters;

2) to develop theoretical models in order to provide a quick comparison between the two systems;

3) To develop models that highlight the differences between the dynamics of relatively insoluble gas bubbles and those in which condensation or vaporisation will lead to bubble collapse and growth.

1.3 Scope of Present Work

This thesis uses the following structure to cover these points:

Chapter 2 presents a literature survey of the present research topic. This includes some general definitions, published work on the differences between air/water and steam/water and finally a survey of the parameters which are expected to be relevant to the project's aims. It will begin with a description of the flow pattern concept in two-phase flow. The second part of chapter 2 concentrates on the differences between air/water and steam/water. This part of the literature survey establishes the possible parameters which may have an influence on this behaviour. The final part of chapter 2 is devoted to the description of parameters covered in the experimental work.

The experimental facility is described in chapter 3 including a discussion of the test rig and the gamma densitometer, together with an assessment of the error associated with the experimental results. Relevant experimental results and their parametric trends are shown and discussed in chapter 4. The first part of this chapter covers the variation of the void fraction with different operational parameters in the vertical direction. It follows a discussion of the effect of temperature on the void fraction emphasising on the temperatures near the boiling point. The influence of constrictions, such as a venturi, will also be demonstrated to see how the flow behaves as it passes the venturi. The steam/water experimental data will finally be discussed by a phenomenological comparison with the air/water results.

Chapters 5 and 6 deal with the theoretical and numerical study. Simple theoretical models based on the scale simplification of the continuity equation and the first law of thermodynamics are derived in chapter 5. The theoretical results are then compared with
the experimental data described in chapter 4. Alongside the above theoretical models, a commercial computational fluid dynamics package (CFX-4.1) has been used to numerically evaluate the differences between air/water and steam/water. The thesis ends with the conclusions based on the results obtained in this study, and some recommendations for future work.
CHAPTER 2

LITERATURE REVIEW
LITERATURE REVIEW

This chapter presents a literature review, aiming to present parameters which may have influence on the differences between air/water and steam/water systems. It begins with a general overview of two-phase flow, considering simplifications such as flow pattern, which give a clearer insight than would attempts for detailed calculations. Subdividing two-phase flow into defined flow patterns allows better descriptions of specific flow regimes (e.g. bubbly flow, slug flow, etc) to be developed. This chapter then gives a discussion of research work that has reported differences between air/water and steam/water systems. This is followed by a discussion of the parameters, such as void fraction, bubble size, heat and mass transfer etc, which may help to put these differences into perspective. This discussion is also indispensable as the following chapters are based on these parameters.

2.1 Flow Regimes in Two-phase Flow

Several different two-phase regimes are observed when gas and liquid pass cocurrently through a pipe. Flow patterns are of great importance in developing models to determine void fraction, pressure drop, heat and mass transfer. The flow pattern developed depends on the pipe orientation. It is also a well-established subject which can be found in the technical multiphase flow books (Butterworth and Hewitt, 1977; Hetsroni, 1982 and Whalley, 1996), even though many different names are used for their classification. The following is a brief discussion of different flow patterns in vertical and horizontal pipes, in order to be consistent throughout this study. The common regimes for upflow in vertical pipes are shown in Fig. 2.1 and described below:

1) Bubbly flow; in this flow pattern, bubbles of approximately uniform shape move along the pipe. The mean velocity of the dispersed gas phase is slightly faster than that of the continuous liquid phase.

2) Slug flow, in which the bubbles coalesce due to higher gas velocity, progressively the bubble diameter approaches that of the tube in the form of large bullet-shaped bubbles. The liquid phase contains a dispersion of smaller bubbles in the wake of the bullet-shaped bubbles.
3) Churn flow, a chaotic movement dominated by large bubbles containing liquid drops as well as a dispersion of small bubbles in the liquid phase. This occurs at a very high gas velocity, leading to breakdown of the slug bubbles. This flow pattern is highly unstable, and the liquid phase may move up and down the pipe.

4) Annular flow, in which a wavy liquid film adheres to the pipe wall while the rest of the liquid is carried along the tube core as droplets. This pattern develops at very high gas flow rates.

![Flow patterns in vertical pipes](image)

**Figure 2.1** Flow patterns in vertical pipes.

The common flow patterns which can occur in a horizontal pipe are depicted in Fig. 2.2 and described below:

1) Bubbly flow, in which segregated bubbles move towards the top of the pipe because of the buoyancy force. It is also known that at a given gas flow rate, the bubbles become smaller and more spherical as liquid flow rate increases.

2) Plug flow, in which big bubbles are produced as a result of bubble coalescence. These elongated bubbles are separated by lengths of liquid which may contain smaller bubbles. It is also instructive to mention that the pipe inclination has a substantial effect on transition between sub-flow patterns. Hewitt (1982) referring to Barnea et al. (1980) showed that only 7° variation in the position of a horizontal
pipe may lead to the stratified flow (a sub-flow pattern between the bubbly and plug flows) to intermittent flow (either of the wavy or slug flows) transition.

![Flow direction](image)

**Figure 2.2** Flow patterns in horizontal pipes.

3) Wavy flow, at higher gas flow rates, when big bubbles break and move the stratified liquid film along the pipe. Waves are formed on the gas and liquid interface and the gas wave velocity is much higher than the liquid phase (Butterworth and Hewitt, 1977). In the extreme condition in which the waves enlarge profoundly until adhere to the upper surface of the pipe, which is then called Slug flow regime.

4) Annular flow, in which liquid drops are carried along in the high velocity gas core while a liquid film forms around the internal circumference of the pipe.

Despite these descriptions of the flow regimes, there is some confusion on the best way to define the transitions between the individual patterns. This is because flow pattern identification may be based on different methods:

1) visual observation;
2) measurements based on the different pressure fluctuating in each regime;
3) Study of void fraction changes during the transition from one regime to another.
As will be seen later, the most reliable method of establishing the flow regime uses gamma ray beam attenuation, which can be used for measuring the void fraction.

### 2.2 Differences Between Air/Water and Steam/Water Systems

The differences between air/water and steam/water systems will be discussed in the following section in order to highlight the circumstances under which distinctive behaviour may occur and what parameters affect it.

**Heat transfer coefficient in flow boiling**

The design of heat exchangers and reboilers involves hydrodynamic and transport phenomena in one-component two-phase flows known as flow boiling systems. Reliable values for such parameters as the heat transfer coefficient are crucial. Although flow boiling is usually a one-component two-phase flow, most simulations have used air/water. Wadekar (1997) has shown that Chen's correlation (1966) which is based on steam/water results, underpredicts the heat transfer coefficients for air/water data at low quality and overpredicts them at higher quality. This disagreement is related to the differences in heat transfer between gas/liquid and vapour/liquid systems. It is stated that during heat transfer in gas/liquid flows, heat needs to be transferred to the gas phase itself, whereas in vapour/liquid heat transfer process stops at the interface and further vapour phase heat transfer is not required. The additional gas phase heat transfer resistance tends to reduce the two-phase heat transfer coefficient. This reasoning leads to the conclusion that air/water data can be used to simulate steam/water flow, providing that the sensible heating requirement of the gas phase for air/water mixtures is taken into account. Although good agreement was attained with the author's own experimental data, no theoretical background is available to support this hypothesis. Fig. 2.3 illustrates the most important aspects of this hypothesis.
Mechanically agitated two-phase reactors

Agitated vessels are very commonly used in industrial applications, with and without reaction. In some cases considerable amounts of vapour are generated in mechanically agitated vessels. Some examples are:

- exothermic reactions in which solvent boil-off controls the temperature;
- organic syntheses carried out under reflux.

For the design of such systems some variables still need to be understood. The Relative Power Demand (RPD), which is the ratio of the power drawn by the impeller in two-phase condition to that needed to drive the same impeller at a given speed when the liquid is neither boiling nor gassed is given by:

\[
RPD = \frac{P_n}{P_v} = f(S, g, v, v_i)
\]

[2.1]

For agitated vapour/liquid systems, it was previously believed that both boiling and gas sparged systems were generally similar with respect to the evolution of ventilated cavities behind the impeller (Breber, 1986; Smith and Smit, 1988 and Smith and Verbeek, 1988). In these early investigations it was assumed that the effect of vapour on the impeller would correspond roughly to that of sparged gas at a given volumetric rate. In contrast, Smith and Katsanevakis (1993) and Smith and Millington (1996) have reported that there are remarkable differences between gassed and boiling systems.

Fig. 2.4 illustrates these differences as a function of RPD versus stirrer speed. Fig. 2.4a illustrates some of the experiments showing that RPD is essentially independent of the volumetric boiling rate in contrast to the situation in a gas/liquid system, illustrated by the smoothed curves of Fig. 2.4b, when it is a strong function of gas flow rate. It has
also been observed that in a boiling system, an impeller operates at a higher RPD than in a gas sparged system at similar volumetric flow rates. The differences between the two systems may be considered to arise from one or more of the following possibilities:

1) there may be differences in the impeller drag coefficient between aerated gas and boiling systems;
2) the actual pressure in a boiling cavity will be very close to the vapour pressure of the liquid around it and differences of only a tiny fraction of a degree of the temperature will cause significant changes in evaporation or condensation of the vapour;
3) the thermal resistance in a boiling system is negligible compared to that in a gassed system due to the weak mass and thermal boundary resistance;
4) a gas bubble which is not soluble is of relatively constant size, with little change due to pressure fluctuation or mass transfer. Conversely, a small change in local pressure or temperature leads to a remarkable change in vapour bubble size with almost instantaneous expansion or collapse.

Modelling the interaction between the two phases in stratified turbulent flow

An understanding of the interactions at either gas/liquid or vapour/liquid interface in cocurrent stratified flow is of considerable value in the engineering design of film
boiling and cocurrent streams in oil and gas pipelines. Lineham (1968) considered steam/water and air/water stratified two-phase flows in an attempt to discern the characteristics of the film thickness and interfacial shear stress (between gas/vapour and liquid layers). The experiments were undertaken in the same geometry of a horizontal rectangular conduit with inner dimension of 12.7 cm (width) by 1.43 cm (height). The dispersed phases (steam and air) were supplied via a boiler and compressor, respectively. The main conclusions of this study are:

- For steam/water flow, where the film is subcooled with respect to the temperature of saturated vapour, interfacial shear stress increases with the condensation rate. The condensation rate is proportional to the difference between the steam temperature and the film subcooling. In contrast, for air/water flow, the interfacial shear stress is found to be satisfactorily represented as a linear function of the film Reynolds number only. In other words, this difference can be expressed as:

  \[
  \text{Interfacial shear stress} \sim f(\text{Re}_{\text{film}}) \quad \text{For Air/Water}
  \]

  \[
  \text{Interfacial shear stress} \sim f(\text{Re}_{\text{film}} \text{ and Condensation rate}) \quad \text{For Steam/Water}
  \]

- A comparison of steam/water and air/water flows shows that for similar conditions, the mean film thickness in steam/water can be less than half that in the air/water case. This is further elaborated in Fig. 2.5.

In this figure Re_x and Re_y are Reynolds numbers based on the superficial phase velocity (approximately 0.05 m/s for the liquid phase and 10 m/s for the gas/vapour phase). As it can be seen the film thickness changes along the conduit for air/water system which is attributed to the effect of entrance effect (within 40 cm). Contrariwise, the smaller film thickness in steam/water case is related to the mass and heat transfer between two phases. This variation may also be a result of viscosity differences (0.9 mPa.s at 25 °C and 0.2 mPa.s at 100 °C), which are probably important since both liquid films are in laminar flow.
Differences Between Air/Water and Steam/Water Systems

Disturbance waves in flow boiling

Other examples of significant differences between gas/liquid and vapour/liquid hydrodynamics are found in the annular flows. The annular flow regime occurs in many heat transfer and multi-phase applications. As is well known, in most cases of two-phase annular flow, large surface waves travel on the annular liquid film adhering to the channel wall. These waves named as “disturbance waves”, have significant effects on the characteristics of the annular flow. These effects include interfacial shear stress (between gas and liquid phases) and liquid entrainment in the core (in extreme cases, up to 80% or 90% of the liquid may be transported as drops). These topics remain subjects of intense investigation by a number of researchers (Whalley, 1987). In order to find the effect of boiling on the behaviour of the disturbance waves Nakanishi et al. (1993) set up two vertical test rigs for air/water and steam/water systems. They found that the wave profile in boiling flow differs significantly from that in adiabatic air/water two-phase flow. Fig. 2.6 illustrates the typical example of the film thickness signal fluctuation (Y-axis) as function of the time. It can be seen that in steam/water case the wave profile has a fairly steep front side and a long-tailed rear side, which represents a highly dented wave surface. Contrariwise, in air/water system, the waves appeared to be spike-like, which corresponds to fairly smooth wave surfaces compared to the

![Graph showing film thickness vs. axial distance from the injector](image-url)

**Figure 2.5** Comparison of film thickness in air/water and steam/water flows at different axial position (Lineham, 1968).
steam/water case. This is seen as one of the possible reasons, which characterises the differences between air/water and steam/water systems.

Fig. 2.7 represents the variation of the mean spatial wave separation as a function of quality for air/water and steam/water flows. The mean spatial wave separation is given by the product of the wave velocity divided by the wave frequency (Whalley, 1996). It can be seen from Fig. 2.7 that in the steam/water system, the wave separation takes a constant value of approximately (0.3 - 0.4m). On the contrary, in the air/water system, the value begins to increase dramatically at a certain quality. This difference is considered to be closely related to the non-equilibrium effects arising from phase change in steam/water system, rather than the liquid distribution between the liquid phase film and the entrainment. The effect of viscosity which is different in both cases (as stated before) may also be relevant. These effects agree with the findings of Lineham (1968), see above.

![Figure 2.6 Recordings of liquid film thickness signals in steam/water and air/water flows (Nakanishi et al., 1993).](image)
Two-phase flow damping in U-tubes

The U-bend tubes of steam generators may be damaged by Flow Induced Vibration (FIV), especially at higher values of void fraction. The FIV is directly proportional to the void fraction, the force induced by the flow and in inverse relation with the pipe inclination and total mass flux. An attempt was made by Axisa et al. (1988) to verify the damping phenomena in different systems like steam/water and air/water. Fig. 2.8 demonstrates the effect of void fraction on damping. It can again be observed from this figure that there are significant differences between the two systems, with the damping more pronounced in the air/water case. Axisa et al. (1988) also found that for the air/water system at ambient temperature, damping is more than 3 times that for steam/water at 210 °C. The greater damping in air/water as compared to steam/water might be explained by considering the effect of surface tension, and its importance for the structure of two-phase flow. Another possibility may lie in the rate at which energy is dissipated by the vibrating tubes. This dissipation was found to be significantly less in steam/water than in air/water at higher void fractions. Therefore it seems that both surface tension and the rate of the dissipation of energy are keys to the clarification of this distinctive characteristic. The effect of local pressure fluctuation is also relevant as the FIV is influenced by the force induced by the two-phase flow. No further discussion
was made, but again further studies have been suggested to investigate the phenomena behind these differences.

![Figure 2.8 Total damping as a function of void fraction (Axisa et al., 1988).](image)

**Pressure drop in two-phase flow**

The prediction of pressure drop in two-phase flow systems is an essential component in the design of a variety of equipment used by process industries. The specifications of pipe length and diameter for oil transportation and heat exchangers, and the design of safety devices in power plants are examples for this. Therefore one of the original aims in the investigation of two-phase flow is to obtain reliable models for estimation of the pressure change. However the similar contradiction appears to be important between the air/water and steam/water systems in prediction of pressure drop in the pipes. Du kler et al. (1964) were among the first researchers who illustrated the differences between air/water and steam/water by comparing the experimental data with the pressure drop correlations. They reported that the Bankoff (1960) and Yagi (1954) pressure drop correlations, which are based on horizontal steam/water pressure drop experimental results show weak accuracy for horizontal air/water two-phase flow.

In addition, Freeston (1980) carried out a set of experiments, endeavouring to characterise pressure drop in one-component steam/water and two-component air/water flows. The aim was to design a safe and efficient pipeline for carrying the one-
component two-phase flow from the geothermal wells, in countries like New Zealand, to
the point where it is utilised (for the purpose of electric power generation). The
air/water experiments were undertaken in a horizontal pipe with industrial size (8.4 cm)
in the annular flow regime. He showed that steam/water pressure drop correlations
overpredict the air/water experimental data, particularly at higher liquid mass flow rate.
It was suggested that further study is necessary before detailed conclusions could be
drawn. However, the difference between steam/water and air/water was tentatively
attributed to the difference between surface roughness of the pipes used in the
experiments. As it will be demonstrated later, the surface roughness has some effects on
the behaviour of the two-phase flow. However as most experiments in this study are
carried out in stratified flow, the conclusions which are given by Lineham (1968) and
Nakanishi et al. (1993) are more relevant in describing this distinctive behaviour
between the two system.

More recently, Tribbe (1998) has evaluated a comprehensive experimental data set of
pressure gradients in horizontal two-phase flows with the available predictive models
(either of correlation or theoretical) in the literature. The pressure drop experimental
data are contained in two large data banks known as the TVT-Dukler and the Stanford
Multiphase Flow Database (SMFD). The TVT-Dukler data bank contains the
experimental data from the University of Houston collected by Professor A.E. Dukler’s
research team and the data from the Institut für Thermische Verfahrenstechnik (TVT) of
the University of Karlsruhe (Germany). The SMFD data bank mainly consists of the
University of Calgary’s Multiphase Database. These data banks comprise of 15000 data
of which 5505 were in the bubbly flow regime. It is shown that the air/water
experimental data are predicted relatively well with the models, which supports the fact
that air and water are used as a model two-phase system and the results are commonly
used in model development. However the accuracy of the models deteriorates when
they are compared with steam/water experimental results. It is also concluded, without
any discussion, that the error between the experimental data and the predictive results is
higher in the bubbly flow regime than in the other flow regimes.

From the above discussion, it is apparent that there are significant differences of opinion
on the use of air/water to simulate steam/water systems. While some researchers
emphasise the similarities of the systems in their individual studies, others have reported opposite results. It seems that any calculation in a specified gas/liquid flow regime cannot simply be applied to the same flow regime in steam/water system and vice versa. Perhaps the most significant distinctive behaviour is revealed in cases, when the two systems encounter obstructions (Axisa et al., 1988; Smith and Katsanevakis, 1993 and Smith and Millington, 1996). It can be concluded that analysis of air/water and steam/water systems still lags behind the theoretical basis of other general fields of flow theory. However, it appears that despite these differences no fundamental study on this subject has previously been carried out.

The first step towards dealing with the differing behaviour of air/water and steam/water systems is to study a bubbly flow regime, for the following reasons:

- Bubbly flow structure can be more easily characterised than that of other flow regimes. Similarly, there is some expectation that it should be possible to measure bubble size and interfacial area more reliably than would be the case for other flow patterns.
- Flow regimes other than bubbly flow are highly unstable and oscillatory. The experimental uncertainties in bubbly flow are therefore much more tractable than those of other regimes.

### 2.3 Bubbly Flow

#### 2.3.1 Definition and Importance

The bubbly two-phase flow regime is characterised by the presence of bubbles with a maximum size much smaller than the diameter of the containing pipe. This flow pattern is widely relevant in the process industries, especially in applications where a large interfacial area is needed for mass and or heat transfer between the phases. However even in adiabatic systems with constant gas and liquid flow rates, the structure of a bubbly flow is complicated. This is because the complex interactions between the two phases, due to the influence of parameters such as bubble shape and its distribution, pressure, and velocity variations. For a quantitative description of gas/liquid flow, it is important to understand the physical phenomena in this flow regime.
2.3.2 Vertical and Horizontal Bubbly Flow

In order to obtain a better understanding of bubbly flow structure, it is necessary to define the phenomena and associated forces involved in flows through both horizontal and vertical pipes. As mentioned earlier, in bubbly flow, bubble sizes are relatively uniform. However, the bubble distribution is different in horizontal and vertical pipes. In vertical upflow, the important force is a horizontal lift force which encourages bubble movement towards the wall, as illustrated in Fig. 2.9. In this figure, $U_b$ is the bubble velocity, $U_l$ the local liquid velocity and $U_r$ the rotation velocity which is the difference between $U_l$ and $U_b$. It is to be expected that bubble distribution can be affected by the velocity gradient in the neighbourhood of each bubble as depicted in this figure. It should be emphasised that the lift force only applies for small spherical bubbles with a diameter of less than 3-4 mm (Ohba et al., 1976).

![Figure 2.9 Illustration of bubble behaviour due to lift force in upward vertical flow.](image)

In horizontal pipes, buoyancy, which pushes the bubbles towards the top of the tube, is the dominant force. Kocamustafaogullari et al. (1994) reported that one of the main differences between horizontal and vertical upflow is that there is a significant positive relative velocity between bubbles and liquid in vertical upflow, while segregation leads to a small negative relative velocity in horizontal flow.

2.3.3 Effect of Bubble Size on the Hydrodynamics of Bubbly Flow

In attempting to characterise bubbly flow, one of the main problems is to determine the effect of bubble size on hydrodynamics, mass and heat transfer. This is crucial for
insight into the structure of this flow regime. It is also directly related to the transport phenomena between the phases. Bubble size in bubbly two-phase flow is strongly tied to the following parameters:

- the physical properties of the two phases;
- mass and heat transfer which may lead to evaporation and condensation;
- operational parameters, e.g. gas and liquid flow rates, temperature;
- geometrical parameters, e.g. bubble generation method and downstream distance.

As will be seen later, the hydrodynamics of bubbly flow are affected by any variation in bubble size resulting from changes in the physical and operational parameters. In order to show the effect of bubble size on the flow structure, Liu (1993) pointed out that the same void fraction can be obtained from either a large number of small bubbles or a small number of large bubbles flowing upward direction in a vertical pipe. However, the flow structures in the two cases are entirely different, with distinct interfacial areas and turbulence dissipation levels. This study also showed that the larger the bubbles generated at the inlet, the more rapidly is a slug flow regime attained in vertical upward flows. This implies that bubble size is an important parameter affecting the establishment of flow patterns.

Nakoryakov et al. (1996) showed that the momentum exchange between liquid and gas phases increases with increasing bubble size. This effect is considered to be closely related to the interfacial area and the intensified level of turbulence, which is considerable at higher gas flow rates. On the other hand, Kocamustafaogullari et al. (1994) observed that in horizontal pipes the effect of liquid velocity on bubble size is much greater than that of gas flow rate. It reflects a higher level of turbulent energy dissipation at higher liquid flow rates. From the perspective of geometrical effects, Takamasa (1989) has reported that wall roughness promotes bubble coalescence, leading to an increase in bubble diameter in sub-flow patterns up to slug flow in vertical tubes. Of the phenomena which may also have influence on the bubble size and subsequently on the hydrodynamics of the two-phase flow are heat and mass transfer, leading to different time response of bubbles to any imposed temperature or concentration.
2.4 Dynamics of Air and Steam Bubbles

The dynamics of bubbles may characterise the differences between the air/water and steam/water systems. Smith (1995) showed that there is a distinctive response to any change in temperature and pressure in the two systems. He considered the response of a 2 mm saturated air bubble at ambient temperature to an imposed temperature. The internal circulation inside the bubble is neglected and approximately considered to be stagnant. The mean internal temperature of a stagnant sphere approaches to within 1% of a step change in its surface temperature when the Fourier Number \((\alpha t/d^2)\) is higher than 0.13. For air the thermal diffusivity \((\alpha_i)\) is about 2*10^{-5} m^2s^{-1} which implies that for a bubble of 2 mm diameter the response time is in the order of 25 ms.

The volume change associated with the condensation of a steam bubble is more complex. However a simple approach can be considered which is only applicable for small bubbles (<3 mm) with no internal circulation, as well as ignoring mass transfer effects. A change in local pressure of 0.01 bar in boiling water corresponds to a difference in boiling point of about 0.25 K. As far as the internal temperature distribution is concerned this would change as rapidly in steam as in air except that vaporisation or condensation, that will lead to volume changes, requires the addition or removal of latent heat. Assuming as a direct contact condensation film heat transfer coefficient does not depend on the bubble size the following differential heat and mass balance is written for a bubble of initial radius \(R_i\) which changes by \(dr\) over a time internal \(dt\):

\[
\int (4\pi r^2)dr \quad \text{(changes in the volume of bubble)} \quad [2.2]
\]

\[
\int (4\pi r^2 \rho_s)dr \quad \text{(mass condensed)} \quad [2.3]
\]

\[
\int (4\pi r^2 \rho_s \Delta h_v)dr \quad \text{(latent heat to be removed)} \quad [2.4]
\]

\[
\int (4\pi r^2 h_c \Delta T)dt \quad \text{(heat transferred)} \quad [2.5]
\]

This leads to:

\[
t = \int_{R_i}^{r=0} \left(4\pi r^2 \rho_s \Delta h_v \right)dr = \frac{\rho_s \Delta h_v R_i}{h_c \Delta T} \quad \text{(time for total collapse)} \quad [2.6]
\]

Assuming that \(\Delta h_v\) (latent heat) is 2.2*10^6 J kg^{-1}, \(h_c\) (internal heat transfer coefficient) is about 10^5 Wm^{-2}K^{-1}, (Jeje et al., 1990), \(\rho_s\) (density of saturated steam) is 0.6 kg.m^{-3} and a
ΔT is 0.25 K (temperature difference between inside and outside the bubble), then the
time of total collapse for a bubble of initial diameter 2 mm would 52 ms.

2.5 Void Fraction in Bubbly Flow

Knowledge of the internal flow structure is needed to advance the investigation of two-
phase flow. There is a major problem in the estimation of the effective local density of
the mixture when trying to determine how the pressure changes along a pipe. This
requires information about the ratio of the cross-section occupied by gas, which is called
void fraction. This is a crucial parameter for hydrodynamic and thermal calculations,
for the characterisation of the flow regime and for defining the onset of changes in flow
pattern. An accurate local void fraction profile is also necessary to determine
momentum, heat and mass transfer between the phases.

2.5.1 Void Fraction Distribution at Different Cross-sections for
Vertical and Horizontal Flows

Void fraction can vary dramatically in both radial and axial directions. As mentioned
above, in vertical pipes the main force acting on the void fraction is the lateral force.
This force in the radial direction is the result of the velocity gradient. There is a second
effect involved in the void fraction profile in that geometric factors prevent bubbles
from getting closer to the wall than their own radius. The wall effect keeps bubbles
away from the pipe wall in the opposite direction to the lift force which is directed
towards the wall. The result of the balance between these effects is to create a local
maximum of void fraction either near the wall or in the pipe centre. The experimental
results reported by Serizawa et al. (1975) with a pipe diameter of 25 mm for fully
developed flow show that at a fixed axial position, four major void distribution patterns
exist (wall peak, intermediate peak, transition and core peak) as shown in Fig. 2.10.

This figure illustrates that as the flow regime changes from bubbly to slug flow, the void
fraction profile alters from peaking near the wall (or saddle shape) to the core peak in
the centre of pipe. These observations have been repeatedly confirmed by many other
investigations (Kobayasi et al., 1970; Liu, 1993; Nakoryakov et al., 1981; Ohba et al.,
1976 and Zun, 1980). The results of these investigations show that in the bubbly flow regime, the void fraction in a given cross-section but at different radial positions may vary by as much as 50%. It is apparent that an explanation for this characteristic is crucial for the development of satisfactory theoretical approaches.

The mechanism of the saddle shape void fraction profile that maintains most of the bubbles at a certain distance close to the wall can be explained as follows:

1) Takamasa (1988) considered that there are two reasons for the peculiar void fraction profile near the wall. In bubbly flow, rolling vortices can be produced near the wall as a result of liquid turbulence and wall friction. These vortices concentrate the small bubbles near the wall. The second effect is that of the lift force. As stated above, the radial gradient in the liquid velocity profile causes the bubbles to rotate. This rotation with the velocity difference between the bubble and the surrounding liquid results in the transverse lift force which keeps the bubbles near the wall.

2) Kobayasi et al. (1970) observed that the bubble velocity near the wall is less than that in the central region, causing bubbles apparently to accumulate near the wall.

3) Liu and Bankoff (1993) suggested that the migration of the bubbles towards the wall can be explained by the effect of viscosity. It is apparent that owing to a strong velocity gradient near the wall, the viscous effect may be important and
subsequently causes a larger drag force on the bubbles near the wall. This encourages the bubbles to rotate near the wall. Other bubbles which are close by might tend to coalesce and produce the higher void fraction near the wall. This reason seems unlikely in a highly turbulent flow.

Liu (1993) observed that the radial void fraction profile is a strong function of bubble size and the phase velocities. The location of maximum void fraction has been reported to be at a distance of between 0.5 and 0.7 \( d_b \) from the wall (Kobayasi et al., 1970) and also at between 0.6 and 1.5 \( d_b \) (Ohba et al., 1976).

Much less attention has been paid to bubbly flow in horizontal pipes than in vertical upflow. As discussed above, in horizontal bubbly flow, bubbles tend to migrate to the upper wall under the dominating influence of buoyancy force. Kocamustafaogullari and Wang (1991) observed that there is a distinct peak of the void fraction near the top wall at about \( r/R = 0.8-0.9 \). A similar observation was reported by Heringe and Davis (1978). The results of both investigations lead to the conclusion that the associated bubble coalescence is the main reason for the asymmetric void fraction profile. This implies that the physical properties of the two phases, particularly surface tension, will influence the void fraction profile.

2.5.2 Effect of Elevated Temperature on the Void Fraction

The temperature of water is obviously important when the distinctive behaviour of air/water and steam/water mixtures is to be considered. It is relevant therefore to look at both gas and vapour dynamics in liquids near the boiling point. It is obvious that the physical properties of the phases are affected by temperature. A survey of the literature reveals that most investigations of gas/liquid two-phase flow simulation and regime recognition have not considered temperature effects, whereas some commercial two-phase flow systems often operate at elevated temperatures, particularly near to or at the liquid boiling point (Tribbe, 1998).

Relevant conditions arise in many tubular reactors in which exothermic reactions are carried out in a two-phase regime; perhaps thermal cracking provides the most spectacular example. A different application involving a similar physical situation is
used in the heat exchangers in which fouling resistance is reduced by increasing turbulence near the heat transfer surface by injecting inert gas into the hot liquid feed. Most investigations of inert gas injection have nevertheless been carried out at ambient temperature (Kuru and Panchal, 1997). It has generally been believed that the effect of temperature could be ignored because of the much higher liquid flow rates than those in bubble columns, together with the immiscibility of gas and liquid. Little or no fundamental work has been published in this respect.

It is believed that one key phenomenon related to the variation of void fraction is bubble coalescence or collapse. When two air bubbles coalesce, they must overcome the resistance to drainage of the liquid film between them. In the case of pure water this resistance is caused by the intermolecular forces of water (hydrogen bonds) which keep water molecules bound together. Jamialahmadi and Müller-Steinhagen (1993) stated that if the intermolecular forces are relatively weak, then the chance of bubble coalescence should increase. It is shown that in the absence of vibration and agitation, temperature has a considerable effect on the rate of bubble coalescence. Several investigators have looked at the effect of elevated temperature in bubble columns. There are large discrepancies between the results of various investigations. Grover et al. (1986) conducted a set of experiments in a pipe with an inside diameter of 10 cm and air and water as the working fluids. They observed that with an increase in temperature the gas holdup decreases substantially below 50 °C. Above this point, however, the effect of temperature is only marginal. They also reported that an increase in temperature leads to a decrease in the gas velocity at which the flow regime changes from bubbly to slug flow. Saxena et al. (1992) confirmed the above results in their experiments in a pipe with 30.5 cm. They observed that as the temperature goes up, the bubble size increases. This effect is attributed to the influence of vapour pressure near the boiling point. Contrariwise, Renjun et al. (1988) showed that, in a pipe with inner diameter of 10 cm, the effect of temperature on the gas holdup can be divided into two stages. At first, below about 75 °C gas holdup increases slowly with an increase of the temperature. Above that temperature gas holdup increases remarkably as the temperature rises. This effect is said to be closely related to the bubble size and vapour pressure of the liquid. They also incorrectly stated that bubble size is smaller at the elevated temperatures because of the increased higher vapour pressure. The present study aims to characterise
the effect of temperature on the void fraction at elevated temperature, particularly near to the boiling point. It is expected that the experiment at elevated temperature will help us to understand how the flow structure changes as the air/water system approaches a steam/water system, and provide a better insight into the different behaviour between air/water and steam/water systems. The effect of parameters, which control the effect of temperature will be demonstrated in chapter 4.

2.6 Geometrical Effects

Knowledge of geometrical effects, is needed:
1) for the design of two-phase experimental test rig;
2) to identify conditions with maximum possible difference between air/water and steam/water.

Literature relevant to these topics is briefly discussed in the following.

2.6.1 Entrance Effects

Some of the most open questions of two-phase flow arise in any discussion of entrance effects. The definition of a fully developed region in two-phase flow is one in which the time averaged void and velocity distribution does not change with length. Reported values of the ratio of the downstream length to the pipe diameter to reach to fully developed stage are in the range from $L/D=6.0$ to 700. The entry length to attain the fully developed flow in a single phase is a function of the pipe diameter, the fluid velocity, the physical properties of the fluid and the wall roughness. However two-phase flows are more complex. Takamasa (1989) pointed out that expansion of the gas phase as a result of the pressure change along the pipe causes density changes of the components with an associated continuous change in flow behaviour. He also stated that for air/water flow a $L/D$ ratio of more than 150 is required to reach a dynamic equilibrium and a fully developed condition. On the other hand, Liu (1993) stated that it is possible to reach the fully developed condition earlier at higher liquid flow rates. Takamasa (1989) also showed that in bubbly flow the effect of bubble size is significant. Fully developed conditions in one-component two-phase flows such as in steam/water, could be more complicated due to condensation or evaporation phase changes.
2.6.2 Effect of Roughness on Developing Vertical Flow

Despite the fact that most commercial pipes are rough, the majority of experimental work has been done in smooth tubes. Takamasa (1988, 1989) showed, for flow patterns up to slug flow in upward vertical pipes, the following effects as the wall roughness increases:

1) the void fraction profile changes from saddle shape (with a maximum near the wall) to the core profile (with the maximum in the centre);
2) bubble diameter increases;
3) the effect of wall roughness reduces as the distance from the inlet increases. This was explained in terms of the change in void fraction profile from saddle to core profile.

These phenomena may be explained as follows: The friction factor increases with wall roughness, leading to the formation of large vortices in the centre of the pipe. The movement of bubbles maintained by the lift force near the wall is disturbed and the void fraction profile changes from a saddle shape to a core profile. On the other hand, Spedding and Spence (1993) reported that except in horizontal bubbly flow, the effect of wall roughness is negligible. From this it appears that wall roughness behaves as a resistance to the flow and that this may cause some effect on the bubbly flow. However, this unlikely to have the same effect on bubble size and void fraction in regimes such as slug and upper flow regimes. In such regimes, the bubbles are much bigger and dominate the effect of vortices which are generated by the surface roughness.

2.6.3 Effect of Flow Obstructions

The investigation of the pressure drop for two-phase flow through constrictions, is relevant to most applications. Obstructions decrease the area available for flow thus increasing the velocity, in particular that of the gas. As a result, a constriction or an obstruction in the flow can induce a transition between flow regimes at lower liquid and gas superficial velocities than would otherwise be the case. (Salcudean et al., 1983).

Kuo and Wallis (1988) observed that in a vertical venturi, bubble velocities first sharply increase and then decrease beyond the throat point. It was also shown that as bubbles reach the throat, they become distorted from spherical to ellipsoidal shape. This can be
explained by the fact that the difference between gas and liquid velocities increases as the throat is approached with the consequence of higher values of the Weber number. This leads to the observed change in bubble shape at the throat. Both the alteration of bubble size and velocity influence the local pressure changes as the flow passes through constrictions.

2.7 Summary and conclusions

This chapter has reviewed literature concerned with flow processes in which any differences between flows of dispersed gases and vapours might be evident. Some discrepancy arises when two-phase mixtures come across an obstruction such as an impeller in mixing tanks. Perhaps the clearest effects have been reported for annular flow, which is probably a result of different drag coefficients in the systems compared. The validation of an air/water system in simulating steam/water mixtures might be most appropriate for bubbly flow due to the simplicity of the flow structure and experimental reliability compared to the other flow regimes. The effects on bubbly flows of the complex bubble dynamics associated with either evaporating or condensing systems do not appear to have been investigated. This confirms the suitability of this area for future work.
CHAPTER 3

EXPERIMENTAL SET UP AND PROCEDURE
EXPERIMENTAL SET UP AND PROCEDURE

The aim of the present study is to obtain reliable measurements of the effect of the parameters that control the differences between air/water and steam/water flows. The present chapter describes the experimental facilities that have been used, including the ASW (Air-Steam-Water) test rig; gamma densitometer; the LDV (Laser Doppler Velocimeter) system and the high speed video camera.

3.1 Experimental Set-up

3.1.1 Description of the ASW Rig

A closed flow loop (open to the atmosphere) has been designed and constructed to investigate the void fraction and pressure drop for air/water and steam/water flows through either a normal pipe or a venturi. The experimental test rig is illustrated schematically in Fig. 3.1.

![Figure 3.1 Schematic diagram of the ASW test rig.](image)

The loop has 1.5 m vertical [V] and 2 m horizontal [H] measuring sections. It is appreciated that these are inadequate for the establishment of fully developed two-phase
flows, however the restricted available space was the main reason in constructing the loop with the above lengths. This limitation is less significant as it is thought that the entry length may be one of the more relevant differences between the two types of flows. The pipe is Pyrex glass with 24.2 mm inside diameter. Flow visualisation, velocity measurement (using LDV system), and high speed video photography are possible in the glass tube. A gamma densitometer [G] can be mounted on either measuring section to measure the local void fraction. The low pressure circulating loop contains a holding tank [A] at the top right hand side, in which the bulk water temperature is regulated by an immersed electric heater and cooling coil. The loop has a circulating pump [P] and a magnetic liquid flow meter (Brooks Instrument, Model 8722) [E] with a range of 0-40 lit/min. A gas mass flow meter [I] is also used within 0 to 12 lit/min (Brooks Instrument, Model 5850s). The details of the error in measurement are subsequently explained in this chapter. The whole test rig is protected against pressure surges caused by the water pump or other nearby equipment. A photo of the experimental test rig is shown in Fig. 3.2.

Figure 3.2 AWS experimental rig.
Both vertical and horizontal test sections have thermocouples (J and K types) to measure the entering flow temperature. The gamma densitometer [G] can be traversed along the test sections. The movement allows the void fraction to be measured at points along each test section separated by only 5 mm. This precise positioning allows the void fraction dynamics of each system to be accurately assessed. The traverse apparatus is shown in Fig. 3.3. For the steam/water work bubbles are generated by a secondary heater either located at the entry to the vertical section [B] or at the start of the horizontal section, as appropriate. The following sections describe the equipment used in air/water and steam/water experiments.

Figure 3.3 Traverse apparatus of the gamma densitometer.
3.1.2 Equipment for Air/Water Experiments

Air is introduced to the test section through a gas sparger. In this study three kinds of gas spargers are used, depending on whether the nature of the experiment requires different bubble sizes or bubble concentrations. Fig. 3.4 shows three different spargers used in this work. Two are concentric stainless steel tubes terminated by cylindrical porous plugs, which provide good gas distribution. The third gas sparger is simply a single nozzle. This is used when low concentration of bubbles is of interest, particularly at elevated temperatures. The sparged air bubbles are in the 2 to 5 mm size range, the size depending on the liquid flow rate. The mixture of air and water passes through the test sections where local void fraction and pressure can be measured. The air/water mixture is separated in the supply tank, with the water being re-circulated.

![Figure 3.4 3 different kinds of gas spargers used in this study.](image)

3.1.3 Equipment for Steam/Water Experiments

All these experiments are naturally carried out at the boiling point. For vertical flow measurements, the circulating water can be superheated by, or steam is generated on the external surface of, a separate 6.0 kW heater in the bottom left hand side tank [B] as shown in Fig. 3.5. In the experiments with horizontal flow, vapour bubbles are generated on the surface of a coaxial rod heater, at the start of the experimental section with a heat input up to 1.6 kW. Since steam bubbles are very sensitive to temperature variation, it is convenient to maintain the bulk temperature of the water at the boiling point with the main heater in the supply tank [A]. The whole test rig, excepting for those areas which needed to be accessed for the experiments, is insulated in order to minimise heat losses.
3.1.4 Description of the Venturis

Venturi profiles have been introduced into the test sections. Two different polycarbonate inserts have been made. These fit closely into the glass tubes and provide the necessary changing pipe cross sections in which the changes in local pressure and void fraction can be measured. Polycarbonate absorbs little gamma radiation and is unaffected by the operating temperatures used in this study. Figs. 3.6 and 3.7 show the schematic diagram and a picture of the venturi inserts with the geometric specifications. Both are 120 mm in length and 10 mm in diameter at the throat with the outer diameter closely matching the pipe inside diameter. The first venturi, which was designed for measuring pressure profiles, has pressure tappings at four points along its length. Four circumferential grooves have been machined, beginning from the throat, separated 20 mm from each other. As the aim is to measure the average pressure over the cross-section, four radial holes of 1.5 mm diameter have been drilled from each channel groove into the central free cross-sectional area. The pressure channel grooves have been isolated from each other with "O" rings that eliminate the danger of any leakage.
flow. Four water filled capillary tubes pass downstream along the glass tube wall inside accurate grooves in the outer surface of the venturi to transmit the pressures for measurement.

The dimensions of the second venturi are similar, but since it is used only to measure the void fraction variation along the venturi, it is made without the pressure tappings and grooves. Void fraction is measured at six points along the venturi as shown in Fig. 3.6. Before doing any experiment, the gamma densitometer was calibrated at specified points along the venturi. The whole unit is tightly sealed.

Figure 3.6 Schematic diagram of venturi sections.

Figure 3.7 Venturi used in this study.
3.1.5 Gamma Densitometer

Measurement of void fraction in two-phase flow is important but challenging. A number of different methods are available. The most common method uses quick-closing valves to trap the two-phase mixture, which is then allowed to separate. This method not only has the disadvantage that the flow has to be stopped or diverted but also has poor accuracy and repeatability. A conductance probe technique is also widely used; it gives precise phase fraction measurements. This can be done either by a point conductivity probe averaged over time or the average conductivity between two large electrodes. Nevertheless the first method is mainly used, in order to maximise the intrusion free passage for the flow into the pipe. The method depends on the difference between the conductivity of the liquid and gas phases, hence the accuracy is poor for two phases of similar conductivity. However, as mentioned above, the probes may obstruct the two-phase flow, particularly at high temperatures, where the air bubbles are partially saturated and tend to collapse or coalescence. The gamma absorption technique avoids these disadvantages and can give accurate results. Moreover this intrusion free method provides absolute measurements with an integration time of less than 0.1 s. This also provides local instantaneous values of the void fraction where a rapid change in the flow characteristics is inevitable such as steam/water two-phase flow at high flow rates. The basic principle of such measurements is simple. The attenuation law of a mono-energetic narrow gamma ray beam in a homogenous medium is defined by the following equation:

\[ I = I_0 e^{-\mu x} = I_0 e^{-\gamma x} \]  

[3.1]

where \( I \) is the transmitted intensity of the gamma ray, \( I_0 \) is the initial incident intensity, \( \mu \) the mass absorption coefficient, \( x \) the thickness of the medium traversed, \( \rho \) the density of the medium through which the beam is passing and \( \gamma (= \mu \rho) \) the linear absorption coefficient of the medium. By considering Fig. 3.8 when there are two phases flowing together, with a wall of total thickness \( x_{\text{wall}} \), the resulting gamma ray intensity transmitted is, according to Eq. [3.1],

\[ I = I_0 e^{-(\gamma_{\text{wall}}x_{\text{wall}} + \gamma_L x_L - \gamma_G x_G)} \]

[3.2]
where the subscripts wall, G and L denote pipe wall, gas phase, and liquid phase, respectively. Since the wall thickness at a given position is fixed, the fractional attenuation of the gamma ray passing through the wall is also constant and equal to:

\[ I_o' = I_o e^{(-\gamma_{wall} x_{wall})} \]  

[3.3]

Subsequently, Eq. [3.2] can be re-written as:

\[ I = I_o e^{(-\gamma_G x_G - \gamma_L x_L)} \]  

[3.4]

It follows from Fig. 3.8 that:

\[ H = x_G + x_L \]  

[3.5]

where H is the total distance between the conduit walls at the measurement cross-section. By substituting Eq. [3.5] into [3.4] it follows that:

\[ \varepsilon_G = \frac{x_G}{H} = \frac{\ln\left(\frac{I_0}{I}\right) / H - \gamma_L}{(\gamma_G - \gamma_L)} \]  

[3.6]

where \( \varepsilon_G \) is the local gas void fraction.

Figure 3.8 Gamma ray transmitted through the pipe.

In this study, an Americium-214 gamma source densitometer is employed for measuring the void fraction. The system has been designed to be sensitive to very small changes in the void fractions in the range from 0 to 2% within a fluid. The slit collimator supplied with the system provides a transverse beam 25 mm wide and 5 mm thick. The signals
for measuring average void fraction are generated by a linear array of 100 detectors, as shown in Fig. 3.9. More design specifications of the gamma densitometer can be found in Appendix A.

![Gamma densitometer diagram]

Figure 3.9 Schematic diagram of the gamma densitometer detectors.

3.1.6 Laser Doppler Velocimeter (LDV)

LDV measurements provide instantaneous information about the fluid flow. The advantages of Laser velocimetry include:

- no flow calibration is required;
- there is no probe which may physically disturb the flow;
- senses only the velocity and is virtually independent of temperature and density of the fluid;
- provides precise measurements of the velocity and turbulence intensity of the flow.

The components of the laser velocimetry system include the laser source, light collecting optics, a photo-detector to convert the light signal into electrical signals, a signal processor to convert frequency to voltage, and a data processor.

The mechanism of LDV measurement is schematically depicted in Fig. 3.10. In the simplest arrangement two intersecting beams originating from a common source produce an interference fringe pattern. A particle traversing this pattern will scatter light with an intensity that depends on its position, so that its velocity can be determined
by measuring this scattering frequency. A velocity measurement can only be made when fine particles scatter light in all directions while going through the volume defined by the crossing beams.

It is appropriate to discuss the reliability of LDV measurement in two phase flow, even though the above advantages have made the LDV system one of the most reliable methods for measuring liquid velocity in single phase flow, more care should be taken where if need for gas/liquid two phase flows. Some of the fine particles mentioned above may become attached to the external surface of gas bubbles, producing noisy signals and subsequently interference in the output. There are significant advantages in applying LDV system for bubbly flow where the bubbles are relatively small (Hassan et al., 1998 and Wilson, 1982). It is advisable, however, not to use the LDV system above a void fraction of 0.15.

In the dual-beam LDV system, the frequency of one of the laser beams is changed slightly (“shifted”) while the other beam is not. This causes the interference pattern in the measurement volume to move in a direction from the shifted beam (with a higher frequency) towards the unshifted beam (at the lower frequency), at a frequency equal to the shift (as illustrated in Fig. 3.10). Thus, a particle passing through this measurement point, and moving in a direction against the fringes, generates a signal with a frequency that is equal to the frequency shift plus its own Doppler frequency. On the other hand, the signal from a particle passing through the measurement point but moving in the same direction at the fringes, has a frequency that is equal to the difference of the frequency shift and its Doppler frequency. This technique allows resolution of the ambiguity in the direction of movement that would otherwise arise with particles moving with small velocities. The frequency shift used in the available LDV system is 40 MHz.

A signal processor extracts frequency information from analogue signals that are mixed with noise due to chaotic movement of particles in the measurement point. The source of these signals is proportional to the particle velocity.
In this study a Helium-Neon laser source was used in the LDV system that measures the velocity and turbulence intensity of the liquid phase. The detailed specifications of the system are given in table 3.1. A precision digitised traversing mechanism with a positional accuracy in the order of fractions of millimetre facilitates the collection of data at different cross-sectional positions. A number of preliminarily experiments were undertaken, prior to the final liquid velocity measurements, to ensure that the focal point is positioned in the right place with respect to the focal length (102.0 mm). The light scattering particles used were aluminium oxide with a range of 0.1 – 0.3 μm.

Table 3.1 Specifications of the LDV system used in this study.

<table>
<thead>
<tr>
<th>Type</th>
<th>He-Ne, 30 mW</th>
</tr>
</thead>
<tbody>
<tr>
<td>Wave length</td>
<td>632.8 nm</td>
</tr>
<tr>
<td>Beam spacing</td>
<td>15.0 mm</td>
</tr>
<tr>
<td>Focal length</td>
<td>102.0 mm</td>
</tr>
<tr>
<td>Fringe spacing</td>
<td>4.3147 μm</td>
</tr>
</tbody>
</table>

Figure 3.10 Fringe movement in a frequency-shifted system.
3.1.7 High Speed Video Camera and Image Analysis

Visualisation is considered a key role in the phenomenological description of two-phase flow systems. This is rather important at the higher temperatures to discern spatial evolution of the flow structure in air/water mixtures where bubble size and distribution changes along the pipe. Video with its facility for instant control of image quality and processing provides an excellent medium of this. In this study a high speed video camera (Kodak Ektapro HS Motion Analyser Model 4540) is used. This camera can acquire at frame rates up to 40500 frames per second. The full frame resolution is 256 x 256 pixels. The images can be downloaded to VHS or digitally to an IEEE GPIB board on the image analysis system.

During the experiments the video images were taken at two positions defined by the downstream length ratios, L/D = 17 and 45, respectively to observe the spatial evolution of flow structure in vertical direction. The test section was covered with a water-filled rectangular polycarbonate box, as shown in Fig. 3.11, that minimised the effects of refraction on the recorded images. The video records were analysed using the image analysis package (Optimas version 5.22) to establish the bubble size distribution. Optimas includes the process of extracting measures, formatting and analysing data.

![Diagram of Polycarbonate box and High speed video camera](Figure 3.11 Polycarbonate box for reduction of refraction.)
3.2 Experimental Procedure

The measurements were carried out by varying liquid flow rate, gas flow rate, temperature, and the initiating local heat load (for steam/water flow at boiling point). The liquid flow rates ranged from 0 to 40 lit/min (0-1.5 m/s through the test section) and the gas flow rates covered a range from 0 to 12 lit/min (superficial velocities from zero up to 0.76 m/s). As this study is limited to bubbly flow, most of the experiments have been carried out at higher liquid flow rates. At each liquid flow rate, the gas flow rate was increased to the point that the bubbly flow regime could not be maintained. The temperature of the continuous phase was maintained at a selected value by regulating the amount of heating load or cooling in the main supply tank [A]. To avoid subcooling of the steam/water system, the measurements were started using the highest value of the local heat supply [B] to initiate boiling.

As mentioned before, different techniques and test rig geometries have been used in this study, aiming to discern any difference between air/water and steam/water systems. These include measurement of void fraction, local phase velocity (LDV), and high speed video photography. In addition to these a variety of flow geometries, including flow through the straight vertical and horizontal pipe and also through venturis. Flowchart 3.1 illustrates the lay-out of the experimental facility.

3.3 Data Recording

An analogue to digital data conversion board with 8 channels is interfaced with an IBM-486 compatible digital computer for data collection. This acquisition system records the gamma ray attenuation, liquid and gas flow rates and the temperatures at different positions of the gamma densitometer using Thuring-Löffel software. The power supplied to the heaters for the steam/water system is controlled by an auto-transformer and recorded from the control panel.

3.4 Error Analysis

The accuracy of the liquid flow meter is within 1% of full scale and for the gas flow meter is within 0.5% (measurement) and 0.1% (full scale). In this study the most
important variable is the void fraction, which is determined as follows: 100 detectors are available to measure the average void fraction at a given axial position as shown in Fig. 3.9. The results of these detectors are integrated by the software to provide a value for the instantaneous mean void fraction over the cross section in question. To achieve
reliable results of void fraction for each data point at different times, several matrices of data were collected to calculate the time averaged void fraction. To check the accuracy of the void fraction measurements, and to provide an independent calibration, data were collected for the empty and full pipe. These results showed that the mean average error was 1.3%. A confirmatory experiment has been carried out to evaluate the accuracy of the averaging of the void fraction by using the gamma probe as follows. An 8 mm diameter glass rod was inserted at different cross-sectional positions and the attenuation measured. These point to point results were found to be within a variation of 3%, which is regarded as satisfactorily consistent. The gamma probe has also been calibrated at various locations in the venturi, as shown in Fig. 3.6, to allow measurements of the two-phase flow through the constriction. The wall thickness of the venturi at each relevant point was measured since this has to be implemented in the calibration program. The gamma probe was calibrated by measurements at the required locations with the venturi both empty and full of liquid, exactly as was done for the straight pipe. These void fraction results were within an average error of 1.5%. A one minute sampling time was used for all data; this brought the average deviations from the known steady state values within 1.0%.
CHAPTER 4

RESULTS AND DISCUSSION
RESULTS AND DISCUSSION

This chapter presents the experimental results for air/water and steam/water flows which provide the basis for the further discussion and analysis. It starts with air/water in upward vertical direction, showing the results at ambient and elevated temperature. This is accompanied by a study of spatial development of the flow and liquid velocity measurement along the pipe. The horizontal air/water results come afterwards in comparison with the vertical results. The next part of this chapter is concerned with the study of void fraction through a vertical venturi. This chapter ends with the presentation of steam/water with a phenomenological comparison with the air/water experimental data.

4.1 Void Fraction in Upward Vertical Air/Water Flow

4.1.1 Void Fraction at Ambient Temperature

The variation of void fraction with respect to the inlet gas flow rate in vertical upflow at room temperature is illustrated in Fig. 4.1. The figure shows data for various fixed water flow rates taken at a point thirty-five diameters from the inlet air sparger. As might be expected, the results confirm that the void fraction rises sharply as liquid flow rate decreases. Changes in the slope for liquid flow rates below 21.5 lit/min ($u_l = 0.8 \text{ ms}^{-1}$) correspond to perhaps the change of the flow pattern from bubbly to slug flow, while at the higher liquid flow rates this transition was not reached.

Fig 4.2 shows the influence of liquid flow rate on the void fraction at several constant inlet gas flow rates. It is obvious that because of the exponential shape of the variation of void fraction with liquid flow rate, it is obviously impossible to reach the two limits, i.e. void fractions of 0 and 1, for the lowest and highest liquid flow rates, especially at higher gas flow rates. The results are consistent with the data reported by Serizawa et al. (1975). They showed that in bubbly flow, the bubble size decreases as the liquid velocity increases, so less space is available for the gas phase leading to lower value of void fraction at a given gas flow rate.
Figure 4.1 Effect of inlet gas flow rate on void fraction at different liquid flow rate.

Figure 4.2 Effect of liquid flow rates on the variation of void fraction in vertical flows.
For the processing of the experimental results at ambient temperature, it is more meaningful to plot the data as a function of the inlet volumetric flow ratio, which is defined as:

$$\beta = \frac{Q_G}{Q_G + Q_L}$$  \[4.1\]

In this equation $Q_G$ and $Q_L$ are the inlet gas and liquid volume flow rates, respectively. From now on, the term ($\beta$) is defined as the inlet volumetric flow ratio at ambient temperature. However at higher temperatures, when vapour pressure is significant and the air bubbles are partially saturated with the local vapour pressure, a modified form is needed which will be defined later in this chapter. Fig 4.3 illustrates the effect of the inlet volumetric flow ratio on void fraction at ambient temperature. It can be seen from this figure in the range $0.02<\beta<0.3$ with the data spread around a straight line, $\alpha<0.35$, which is the bubbly flow regime.

4.1.1.1 Effect of Bubble Size on the Void Fraction in Vertical Flow

Many attempts have been made within the past decades to develop correlations for void fraction. In most of them, the proposed correlation is independent of bubble size (Bankoff, 1960; Zuber and Findlay, 1965). However, Sekoghechi et al. (1987) and Liu
(1993) have reported that there is a strong dependence of the phase distribution and bubble frequency on bubble size in concentrated bubbly flow. In Fig. 4.4 the effect of constant inlet volumetric flow ratio on the void fraction is plotted for various liquid and gas flow rates. The data in this figure have been taken from Fig. 4.3. The inlet volumetric flow ratio is also constant, however at different values of liquid and gas flow rates. It is obvious from the figure that despite the constant value of the inlet flow ratio $\beta$, the void fraction changes dramatically at higher liquid and gas flow rates. It is sufficient to note that the visual observations of the flow at constant $\beta$ (i.e. increasing both $Q_G$ and $Q_L$) show that the diameter of bubbles reduces while the bubble frequency increases when flow rates increase. Smaller bubbles will rise less quickly, slip velocity is reduced and so the void fraction can be expected to rise. It appears that the influence of bubble size on void fraction should be taken into account when correlating void fraction in bubbly flow, however further study is necessary before firm conclusions can be drawn on this point.

Figure 4.4 Effect of change of gas and liquid flow rates on void fraction at two constant inlet volumetric flow ratios.
4.1.2 Effect of Elevated Temperature on the Void Fraction

It has already been said that the various investigations on the effect of temperature on void fraction do not agree. Most of these earlier studies have been done in bubble columns, so a complementary set of experiments has been carried out for flowing air/water mixtures. The present work includes data that cover a range of increasing partial vapour pressure as the system approaches its boiling point. The vapour pressure shows its effect in the form of bubble growth at higher temperatures. Apart from this, the second effect at higher temperature is bubble coalescence. Because of its importance at higher temperatures it is appropriate to discuss bubble coalescence at this stage. The coalescence of a pair of bubbles occurs essentially in two stages: (i) the draining of the intervening film of the continuous phase liquid down to a critical thickness (ii) the rupture of the remaining film. An increased temperature leads to a lowering of surface tension and viscosity together with a dramatic increase in vapour pressure. Any effect leading to evaporation of the liquid film between the bubbles will lead to the rapid growth and therefore bubble coalescence, particularly near to the boiling point.

![Graph](image)

**Figure 4.5** Effect of temperature on the void fraction in vertical direction at low liquid flow rate.

The effect of inlet volumetric flow ratio ($\beta$) on the void fraction at different temperatures is depicted in Figs. 4.5 and 4.6 for low and high liquid flow rates, respectively. It is obvious that rising temperature causes void fraction to increase.
especially between 90 and 95 °C. It should be pointed out that changes in void fraction at higher liquid flow rates with a given gas flow rate are considerably higher than at lower liquid flow rates, particularly at higher temperatures. A comparison between Figs. 4.5 and 4.6 shows the significant effect of temperature at any flow rate.

Figure 4.6 Effect of temperature on the void fraction in vertical direction at high liquid flow rate.

Figure 4.7 Variation of void fraction as function of temperature at constant volumetric flow ratio.
Fig. 4.7 illustrates the variation of void fraction as a function of temperature at approximately constant inlet volumetric flow ratio for 6.65 lit/min liquid flow rate. At first, the void fraction increases but only marginally until a substantial rise at about 60 °C occurs. It can also be seen that the variation of void fraction is more pronounced near the boiling point. The significant change in void fraction with temperature may result in a change in the flow regime. A comparison of the variation of void fraction at temperatures of 24 °C and 95 °C reveals that there is a difference between the dependence of void fraction on the inlet volumetric flow ratio at ambient and elevated temperatures. Of the parameters, which may influence this phenomenon, one of the most important is the liquid vapour pressure. Table 4.1 gives the physical properties of water and air at different temperatures, which shows the significant variation of vapour pressure, especially near to the boiling point.

**Table 4.1**  
Physical properties of air/water at different temperature.

<table>
<thead>
<tr>
<th>Temperature (°C)</th>
<th>$\rho_G$ (kgm$^{-3}$)</th>
<th>$10^2 \mu_G$ (mPa.s)</th>
<th>$\rho_L$ (kgm$^{-3}$)</th>
<th>$\mu_L$ (mPa.s)</th>
<th>$p_v$ (10$^5$ Pa)</th>
<th>$\sigma$ (Nm$^{-1}$)</th>
</tr>
</thead>
<tbody>
<tr>
<td>25</td>
<td>1.185</td>
<td>1.833</td>
<td>998.23</td>
<td>0.894</td>
<td>0.0313</td>
<td>79.96</td>
</tr>
<tr>
<td>95</td>
<td>0.962</td>
<td>2.163</td>
<td>960.96</td>
<td>0.295</td>
<td>0.8813</td>
<td>60.51</td>
</tr>
</tbody>
</table>

The above results have been presented as a function of inlet volumetric flow ratio. Nevertheless a modified value of $\beta$ should be used since the injected bubbles will increase in volume as they quickly saturate with water vapour at the operating temperature, (a 3 mm bubble reaches 95% saturation within about 1 second of exposure to hot water). Therefore, it is logical and convenient to define a modified volumetric flow ratio as:

$$\beta^* = \frac{Q_G}{Q_G + Q_L}$$  \[4.2\]

in which:

$$i = \frac{(p_a + \rho gh)}{(p_a + \rho gh) - p_v}$$  \[4.3\]

Equation \[4.2\] shows that the modified volumetric flow ratio changes in response to vapour pressure and local hydrostatic pressure. In this equation $p_a$ is atmospheric
pressure, $p$, vapour pressure, $g$ the gravitational acceleration, $h$ the local submergence below the free surface of the measurement point and $\rho$ the liquid density. Substitution of equation [4.3] into [4.2] yields:

$$\beta^* = \frac{i}{\left(\frac{1}{\beta} + (i - 1)\right)}$$  \[4.4\]

It is apparent that at ambient temperature ($i \approx 1$) the modified volumetric flow ratio ($\beta^*$) reduces to the ordinary $\beta$. The influence of temperature on the modified inlet gas flow rate required to achieve a given void fraction is shown in Fig. 4.8. This figure highlights that the inlet gas flow rate required to achieve a void fraction of 0.15, decreases from 1.52 lit/min to 0.4 lit/min for a temperature change from 20 °C to 90 °C, while the modified gas flow rate, including the effect of vapour pressure at the operating temperature is almost constant. This figure also indicates that at higher temperature, especially near the boiling point, the ordinary volumetric flow ratio definition does not reflect the actual volume of the gas flow rate at different temperatures.

![Figure 4.8](image)

Figure 4.8 Variation of inlet gas flow rate as function of temperature at constant void fraction.
The experimental data shown in Fig. 4.8 are measured with the lowest possible fluctuation as illustrated in Fig 4.9. Experiments at higher temperature show that a small change in operational parameters may substantially change the void fraction at elevated temperatures, particularly near to the boiling point. Of these parameters are gas flow rate, temperature at a given axial position of the tube. Therefore it is important to obtain the data at lowest fluctuation. Fig. 4.9 corresponds to the data which are shown in Fig 4.8.

![Variability of the operational parameters during the experiment of which the data are shown in Fig. 4.8.](image)

**Figure 4.9** Variability of the operational parameters during the experiment of which the data are shown in Fig. 4.8.

Malayeri et al. (1998) studied the effect of liquid flow rate on the void fraction and experimentally showed that in vertical flow at ambient temperature with a constant inlet gas flow rate the void fraction decreases as the liquid flow rate goes up. Fig. 4.10 illustrates the effect of temperature on the void fraction as a function of liquid flow rate at constant inlet gas flow rate. Void fraction varies remarkably at ambient temperature, while at 95 °C variation of void fraction with liquid flow rate is small as the void fraction is about 0.75 and hence in the churn flow regime. It therefore can be concluded that the liquid flow rate influences the void fraction at low temperatures. However, at high temperatures the contribution of water vapour is the main parameter which changes the initial gas flow rate from 8 lit/min (at 15 °C) to 45 lit/min (at 95 °C). The bubble
growth/coalescence occurring at higher operating temperatures can minimise the effect of liquid flow rate and change the flow regime.

Figure 4.10 Effect of liquid flow rate on the void fraction at different temperature.

4.1.3 Void Fraction at Different Axial Positions

A set of experiments has been undertaken to investigate the importance of vapour pressure on the void fraction and also to discern how quickly air bubbles will be saturated with the hot water at different axial positions or in other words, the bubble dynamics. Great care has been taken to generate bubbles with diameters as small as possible at very low void fraction ($\varepsilon=0.02$) and at high liquid flow rate to avoid bubble coalescence which may occur at lower liquid and higher gas flow rates. Accordingly a single metal tube with 1.2 mm inside diameter replaced the former porous gas sparger. The generated bubbles were within the range of 2.0 - 3.0 mm. More experimental data were collected within a range of 90 °C to 98 °C, due to the importance of vapour pressure near to the boiling point.

Fig. 4.11 represents the variation of void fraction as a function of temperature at two different axial positions. It can be seen that void fraction changes slightly up to 60 °C and increases monotonically thereafter. The results are consistent with those shown in
Fig. 4.7. When the comparison of the void fraction is made at different axial position, there is a slightly lower void fraction at the point L/D = 45 at lower temperature. Contrariwise, there is a higher void fraction at the point L/D = 45 at higher temperatures, particularly above 90 °C. These results imply that i) as expected, bubbles contain more vapour at higher temperatures and ii) the significant change of void fraction at L/D = 45 compared to the L/D = 17, is because of a further increase in bubble size as a result of approaching saturation as well as lower pressure level at L/D = 45. This figure also demonstrates that the volume fraction of the gas phase in the test section can be changed 7 times with respect to its initial volume as the temperature changes from 17 °C to 98 °C.

![Void fraction vs Temperature](image)

Figure 4.11 Comparison of the void fraction at different axial positions as a function of temperature.

4.1.4 Effect of Temperature on the Development of Flow Structure

Although the primary intention of this work was to consider the characteristics of void fraction, consideration of the spatial evolution of the flow structure at different temperatures is also of interest. Grover et al. (1986) reported that in bubble columns, increasing temperature leads to a decrease in the gas velocity at which the flow regime changes from bubbly to slug flow. This can only be a result of bubble growth and/or coalescence. Saxena et al. (1992) observed that as the temperature goes up, the bubble
size increases. This effect is attributed to the influence of vapour pressure near the boiling point. On the contrary, Renjun et al. (1988) found that bubble size is smaller at elevated temperatures. It can be concluded that with fixed flow conditions, the effect of temperature is the key parameter describing bubble size and shape and consequently the spatial evolution of the flow structure. The results which were recorded using the high speed video camera are presented in this section.

<table>
<thead>
<tr>
<th>T=17 °C, Q=6.0 lit/min and L/D = 45</th>
</tr>
</thead>
<tbody>
<tr>
<td>Q_g=0.6 lit/min</td>
</tr>
</tbody>
</table>

\[
\text{Fig. 4.12 Spatial evolution of the flow structure at different gas flow rates and at room temperature.}
\]

Fig. 4.12 shows the recorded images of flow structure at two different positions with respect to the variation of gas flow rate at ambient temperature. It is obvious from this figure that i) not only bubble size, but the tendency of the bubbles to cluster increases as the gas flow rate rises, ii) the change of bubble shape from spherical to ellipsoidal shape and subsequently distorted shape, which is one of the most important parameters in the process of bubble coalescence, becomes more and more distinct as either gas flow rate increases or liquid flow rate decreases, iii) smaller bubbles as well as bubbles which are flowing downwards in the liquid film around the large bubbles enter into the wake of
the large bubbles as a result of a predominant updraft force and, v) big bubbles tend to move in the centre of the pipe. Tomiyama et al. (1998) and Zun et al. (1993) showed that the presence of shear-induced turbulence near the wake of large bubbles is the main cause of the clustering of the smaller bubbles. It should also be noted that the cluster of bubbles at L/D = 17 is likely to develop into a Taylor bubble at L/D = 45, as a result of the coalescence of many of the bubbles in the cluster. The coalescence of bubbles and formation of a larger cluster or a Taylor bubble are generally seen at different gas flow rates, but typically take place at lower L/D as the gas flow rate rises. Fig. 4.12 also implies that in spite of the almost constant void fraction at low temperature (Fig. 4.11) the structure of the flow is entirely different at the different axial positions. More photographs of the bubble shape can be found in Appendix B.

<table>
<thead>
<tr>
<th>Qg = 0.6 lit/min, Ql = 35 lit/min and L/D = 45</th>
</tr>
</thead>
<tbody>
<tr>
<td>T = 17 °C</td>
</tr>
</tbody>
</table>

Figure 4.13 Spatial evolution of the flow structure at different temperatures.

Fig. 4.13 shows the effect of temperature on the void fraction at a given liquid flow rate and different axial positions. The highest possible liquid flow rate has been taken to minimise the effect of initial bubble coalescence at the entrance. From 17 °C to 60 °C
the variation of bubble size is small, but it becomes more and more distinct between 90 °C and 95 °C. A comparison of the bubble shape at different positions also shows that the bubbles become saturated at higher position in flow direction, as indicated the by the increase in size. This distinctive behaviour is more dominant at 95 °C. On the contrary, the variation of bubble size is much smaller at lower temperature, due to the limited effect of vapour pressure. The interesting result is that the bubble clustering seen in Fig. 4.12 was not observed at higher temperatures. This can be related to the partial saturation of air bubbles with water vapour. Since the air bubbles have been saturated at higher temperatures, the resistance of the interface between the phases becomes weaker while this resistance is stronger for pure air bubbles at lower temperatures. More evidence for the spatial development of the bubbles can be found in Appendix B.

![Figure 4.14 Schematic diagram of bubble shape and size at ambient and high temperatures.](image_url)

The bubble shape evolution at different axial positions is schematically illustrated in Fig. 4.14 at ambient and elevated temperatures. There is a small increase in bubble size
the higher axial position and a change in bubble structure from spherical to ellipsoidal shape at ambient temperature. Bubbles also tend to move as a cluster of distorted bubbles. While at high temperature the air bubbles are partially saturated and as a result of this bubble size changes rapidly as well as its shape. It is also observed that bubble clustering declines and instead coalescence of big ellipsoidal bubbles in a cluster with low population of bubbles. These observations at higher temperature, particularly near to the boiling point, are comparatively similar to the evolution of steam bubbles in steam/water flow.

Fig. 4.15 shows the quantitative variation of mean area-based bubble diameter with temperature at different axial positions corresponding to the experimental runs, which were shown in Fig. 4.13. This figure also contains the minimum and maximum range of bubble sizes as well as the schematic bubble shape at a specified temperature. The biggest bubbles are significantly larger in the range from 90 °C to 95 °C than at lower temperatures. The size range is also wider and so depends on the operating temperature. The significance of the saturation process becomes more evident when different axial positions are compared.

![Figure 4.15 Mean, minimum and maximum bubble size as a function of temperature.](image)
Two factors influence the growth of air bubbles in an upward flow of hot water: the fast but not instantaneous saturation of the gas phase with water vapour and the probability of coalescence within clusters of small bubbles. Observation suggests that bubble clusters are more stable at lower temperatures, so that larger bubbles occur naturally in hotter conditions. In considering the data of Fig. 4.11 we observe at 95 °C that the void fraction increases from 13% to 16% in moving from L/D = 17 to 45. This increase in volume may well reflect the final stages of saturation of relatively large bubbles. However, the photographs of Fig. 4.13 suggest that the individual bubble volumes may have increased as much as fourfold over this same distance. It seems very likely that saturation has had less influence on the size of these large bubbles than enhanced coalescence at high temperature. Certainly the changes at lower temperature, also shown in Fig. 4.13, are not as spectacular.

In addition to the above observations in which the liquid and inlet gas flow rates were kept constant, a set of experiments has also been undertaken, to establish the variability of bubble size at constant void fraction. Void fraction was maintained at a specified value by adjusting the flow rate of inlet gas at the appropriate temperature. Fig. 4.16 shows pictures of bubbles for various temperatures and a given constant void fraction.

![Figure 4.16](image-url)
(more results are shown in Appendix C). It is evident that the bubble shape and size are remarkably different at the various temperatures. At low temperatures the bubbles tend to move as a cluster and are significantly larger, while this is less so at elevated temperatures. The same results were also obtained as the experiment was repeated with different liquid and gas flow rates. These observations are potentially important for the processes in stirred vessels where parameters such as relative power demand (RPD) is dependent on the shape and size of bubbles at elevated temperatures.

4.1.5 Bubble Dynamics

It is evident from Fig. 4.13 that the small bubbles introduced into the pipe saturate and grow rapidly but not instantaneously. Therefore it is rather important to determine how quickly air bubbles saturate with the vapour water. This can be done by considering a simple analysis in which a single bubble with initially 2 mm diameter is considered to be in saturated water. The internal circulation is ignored, which is valid for small bubbles. As a result of this the mass transfer is controlled by molecular diffusion. It is also assumed that the internal pressure in the bubble does not change as it expands and that the water is kept at its boiling point so that the driving force for mass transfer is given by the difference between the local vapour pressure around the bubble and the partial pressure inside the bubble. Further to these assumptions, the resistance to heat transfer on the liquid phase is assumed to be marginal. With these assumptions, the internal Sherwood number can be considered to be of a value of 6.6 for diffusion into a stagnant sphere. The variation of bubble size, based on these assumptions, as a function of time is illustrated in Fig. 4.17. It can be seen that a bubble with initial size of 1.5 mm is increased to 5 mm within 1 second. However the rate of change in bubble size decreases as the initial size increases (for instance for a bubble with the initial size of 3.2 mm). The results are comparatively consistent with the results shown in Figs. 4.13 and 4.15. Table 4.2 presents the quantitative results for the data which are shown in Fig. 4.15. The homogenous velocity between the two phases is assumed to be the superficial gas velocity (0.022 m/s) is much lower than the liquid velocity (1.26 m/s). This assumption seems to be questionable for flow with big bubbles, however this will be discussed in chapter 5 in more detail. Considering the two axial positions along the pipe, it can be calculated that it takes 0.31 see for the air bubbles to travel between these
points. The change in average bubble size varies from 3% at ambient temperature to 95% at 95 °C.

**Table 4.2** Changes in minimum, average and maximum bubble size corresponding to the data shown in Fig. 4.15.

<table>
<thead>
<tr>
<th>Temperature (°C)</th>
<th>Changes in average size compared to the initial size (%)</th>
<th>Minimum and maximum bubbles size (mm)</th>
</tr>
</thead>
<tbody>
<tr>
<td>14</td>
<td>3</td>
<td>1.7-5.5</td>
</tr>
<tr>
<td>60</td>
<td>16</td>
<td>1.9-7</td>
</tr>
<tr>
<td>90</td>
<td>66</td>
<td>2-12</td>
</tr>
<tr>
<td>95</td>
<td>95</td>
<td>2-23</td>
</tr>
</tbody>
</table>

**Figure 4.17** Changes in bubble size in boiling water.
4.1.6 Laser Doppler Velocimeter Measurement

Figs. 4.12 to 4.16 illustrated the characteristics of flow structure at ambient and elevated temperatures in which the flow structure is shown to be different even at a given void fraction and different temperatures (Fig. 4.16). Further to this, it might be of interest to consider LDV measurement in characterising the flow structure of the two-phase in more details.

The mechanism and reliability of LDV measurement in two-phase flow was briefly described in chapter 3. A preliminary experiment was carried to discern the optimum amount of fine particles that has to be used to avoid neither of interfering the output signals (excess particles), nor lack of necessary signals for data processing (less particles). There is no method of calculating the optimal dose of particle in advance, therefore this has to be done experimentally.

The experiments start with local velocity measurement for the liquid single phase, prior to two-phase flow measurement. This is, in fact, an experiment to confirm the reliability of the experimental results in relation to the superficial liquid velocity \(Q_l/A\). Fig. 4.18 demonstrates the variation of local liquid velocity for liquid single phase as a function of \(r/R\) ratio, in which \(r\) is distance from the centre of the tube divided by \(R\),

![Figure 4.18](image-url)
the pipe radius. As expected, there is a low liquid velocity near to the wall with a sharp increase near to the centre of the tube. A comparison of the superficial liquid velocity with the average experimental results at different positions at a specified cross section of the tube shows an error of 1.1 percent. All data have been collected 1 m above the gas sparger (L/D=40).

Figure 4.19 Effect of gas flow rate on the local liquid velocity.

The effect of gas flow rate on the local liquid velocity and comparison of single phase and two-phase flow are shown in Fig 4.19. It is evident that injection of gas into the tube, leads to a similar trend as that obtained for the single phase. Moreover, an increase in gas flow rate promotes the local liquid velocity, predominantly near the centre of the tube. However increase in the local liquid velocity is less than 8%, for changes in gas flow rate from 0 to 1 lit/min. Fig 4.20 represents the effect of temperature on the local liquid velocity where the inlet liquid flow rate and void fraction are kept constant, while the inlet gas flow rate and temperature change. Again changes in local liquid velocity are not significant near the wall, nevertheless the increase in local liquid velocity becomes dominant nearer the centre or at higher temperatures close to the boiling point. The results are confirmed by repeating the experiment for other values of constant void fraction.
4.2 Void Fraction in Horizontal Flow

The dependence of void fraction in horizontal flow on the gas flow rate for several liquid flow rates is shown in Fig. 4.21. With a fixed gas flow rate it can be seen that void fraction decreases as liquid flow rate increases. The fluctuation of void fraction with respect to the inlet gas flow rate is significant at a lower liquid flow rate owing to the generation of big bubbles which leads to increasing segregation of the phases. The experimental data are also more linear at higher liquid flow rates. Fig. 4.22 shows the dependence of void fraction on liquid flow rate. The change in void fraction is more linear in horizontal flow than in vertical flow. This is probably as a result of the larger effects of buoyancy forces in horizontal flow which tend to propel bubbles towards the top of the tube, leading to faster bubble agglomeration than in vertical upflow.

Considering the effect of temperature on the void fraction, the observations have confirmed that this effect is more pronounced at higher liquid flow rate and smaller gas flow rate. A typical illustration of this is Fig. 4.23. It can be seen that at a liquid flow rate of 22.3 lit/min ($u_l = 0.8$ m/s), the void fraction increases about sixfold between 15
°C and 95 °C, while this variation would be tenfold at 34.3 lit/min ($u_L = 1.24$ m/s). At 15 °C the flow regime corresponds to bubbly flow, for which it has earlier been pointed out that void fraction decreases as liquid flow rate rises. At 95 °C the effect of gas flow rate is considerably greater than that of the liquid flow rate because of the influence of the high vapour pressure; as a result of this the flow regime is changed to a wavy/stratified flow.

![Figure 4.21 Void fraction as a function of gas flow rate at ambient temperature in horizontal flow.](image)

An alternative illustration of this effect is as follows. It was shown earlier that in vertical flows at ambient temperature with a constant inlet gas flow rate the void fraction decreases as the liquid flow rate goes up (Figs. 4.2 and 4.10). Fig. 4.24 shows the effect of liquid flow rate on the void fraction in horizontal flow at constant inlet gas flow rate. Void fraction varies strongly at ambient temperature, while at 95 °C variation of void fraction with liquid flow rate is negligible as the void fraction is about 0.78, (in the wavy flow regime).

This can be related to the fact that the most important parameter at low temperatures is the liquid flow rate. However, at high temperatures the contribution of water vapour is the parameter which changes the initial gas flow rate from 8 lit/min (at 15 °C) to 45 lit/min (at 95 °C). Therefore, bubble growth and coalescence associated with higher
operating temperatures can minimise the effect of liquid flow rate and change the flow regime.

**Figure 4.22** Effect of liquid flow rate on void fraction at different inlet gas flow rates.

**Figure 4.23** Effect of liquid flow rate on void fraction as a function of experiment number at different temperature.
4.2.1 Variability of the Void Fraction Measurements

The transition from a specified flow regime to another can be identified with a gamma densitometer. In order to do this, the measurements of the instantaneous void fraction taken at different times have to be reliable. In bubbly flow because of the approximate uniformity of bubble size, void fraction cannot be independently varied, but it changes substantially as the flow pattern changes to slug or wavy flow when large bubbles develop across the pipe section. The results of a sequence of experiments at 0.01 sec of sampling time, fixed inlet air and water rates ($\beta=0.067$) and different temperature are shown in Fig. 4.25. This figure is also intended to demonstrate the effect of growing bubble size on the reproducibility of the void fraction even in a specified flow regime such as bubbly flow. The details are also summarised in table 4.3.

![Figure 4.24 Effect of liquid flow rate on void fraction at different temperature.](image)

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Temperature (°C)</th>
<th>Number of data</th>
<th>$\alpha$ (average)</th>
<th>$\pm\alpha$</th>
</tr>
</thead>
<tbody>
<tr>
<td>□</td>
<td>18</td>
<td>14</td>
<td>0.040</td>
<td>0.006</td>
</tr>
<tr>
<td>Δ</td>
<td>45</td>
<td>12</td>
<td>0.057</td>
<td>0.005</td>
</tr>
</tbody>
</table>
Figure 4.25 Effect of temperature on the variability of the void fraction measurement.

It is apparent that there is a significant influence of temperature, which increases the void fraction from 0.04 to 0.18. Moreover there is a remarkable gap when the temperature is raised from 75 °C to 90 °C. The reproducibility of the measured void fraction also deteriorates as the temperature goes up. This variability is closely related to bubble growth/coalescence and the generation of bigger and unstable bubbles at higher temperature, giving rise to a change in flow regime.

4.3 Effect of Constrictions on the Hydrodynamics of Air/Water

Constricting devices such as venturis are widely used in flow rate and pressure drop measurements. Despite their importance, sufficient experimental results are not available in the open literature (Lin, 1994). As far as the aims of this study are concerned, it has also been said before that the differences between air/water and
steam/water flows could be most evident in the changing flow past obstructions or through constrictions. A venturi nozzle described in chapter 3 can be mounted in either the vertical or horizontal test section. The basic principle underlying the flow behaviour through venturi nozzles are an increase in the velocity of the flow, as well as a decrease in the pressure of the flow under consideration. The local pressure changes will also cause phase change in the steam/water system, so called flashing phenomenon.

![Figure 4.26](image)

**Figure 4.26** Effect of liquid flow rate on void fraction at constant gas flow rate at ambient temperature.

Fig. 4.26 shows how the local void fraction changes as the mixture flows through a vertical venturi. There is a maximum voidage at the throat in stagnant conditions, with a dramatic change to a minimum when there is an upward liquid flow. In order to explain the results some consideration of slip velocity between the phases is essential. In a slow downflow the void fraction at the venturi throat will always be higher than that in the approaching tube. With a liquid downward velocity slowly increasing from rest, stationary bubbles will eventually accumulate at the throat. The opposite result will be true for upflow, as bubbles will always tend to rise relative to the liquid phase. At the throat of the venturi bubbles, which are usually of between one and three mm in
diameter, tend to coalesce because of significant wall effects in a throat of 10 mm diameter. As a result of this coalescence, the relative velocity ratio between the phases \( (u_r) \) gradually becomes more significant, leading to a minimum void fraction at the throat. Bubble coalescence at the throat point is inevitable due to the wall effects. Moissis and Radovcich (1964) observed a change in flow regime from bubbly to slug flow through a venturi tube with 5 cm diameter at the inlet and 2.23 cm at the throat. They also found that the effect of slip velocity is more dominant than that of pressure drop at the throat, leading to a decrease in void fraction (Thang and Davis, 1979).

![Figure 4.27](image)

**Figure 4.27** Effect of liquid flow rate on the void fraction through venturi.

Fig. 4.27 illustrates the effect changes in liquid flow rate on the behaviour of the void fraction through a venturi. The trend of having of a minimum point is repeated for all curves. This figure also leads us to conclude that the proportional variation of void fraction is greater at higher liquid flow rates. Fig. 4.28 shows the impact of the different gas flow rates as flow passes through the venturi. The relative changes in void fraction are almost the same at the various gas flow rates.
The effect of temperature on the variation of void fraction for a fixed inlet flow ratio in the vertical venturi is depicted in Fig. 4.29. In this figure, void fraction rises at constant inlet gas and liquid flow rates as the temperature increases. These observations are very similar to those in a straight pipe. It is interesting that the minimum point at higher temperature is considerably deeper than that at lower temperature. One explanation for this might lie in a change in flow regime because of bubble growth and coalescence at higher temperatures. Visual observation has confirmed that, alongside the vapour pressure effect on the gas phase volume, bubble size is a strong function of temperature. Bubbles tend to become larger and perhaps coalesce together (particularly in the tubes with small diameters due to the wall effect) as temperature increases and consequently bubbly flow can change to the slug flow at the venturi throat. The variation of void fraction through the venturi may again provisionally be attributed to the change in the relative velocity between the phases and bubble coalescence owing to the wall effect at the throat. It is also interesting to note that increase and decrease at the outlet of the venturi become less and changes to an asymptotic line to a certain value of the void fraction.
fraction. This can be explained by consideration of rapid expansion of bubble, because depressurisation of the flow in the outlet, particularly near to the boiling point.

![Graph showing variation of void fraction through venturi at different temperature.](image)

**Figure 4.29** Variation of void fraction through venturi at different temperature.

### 4.4 Steam/Water System

Steam/water experiments have been conducted in both horizontal and vertical flows to provide quantitative comparisons with the air/water data. Steam/water experiments are obviously more difficult than those in an air/water system with the instability of steam bubbles causing difficulties in obtaining reliable data. The first step in dealing with the comparison between the two systems is to measure void fraction. The circulating water is maintained at boiling point by a controlled immersion heater in the supply tank. Steam bubbles are generated on the surface of secondary local heater at the entrance of either of the measuring sections. The role of the secondary heater is to provide a degree of superheat for the liquid which enters to the measuring section. Therefore most steam bubbles are generated as a result of a decrease in pressure at the point of which water reaches its boiling point, particularly near the top of the tube. It is also observed that only at lower liquid flow rates, vapour bubbles are produced on the surface of the
secondary heater and this does not occur at higher liquid flow rates. All data have been taken at steady state conditions keeping temperature and flow constant. The liquid flow rate was measured upstream of the test section. It is also more convenient to refer to the liquid flow rate as the total flow rate, since a proportion of this liquid will be converted into steam and provides the vapour component of the two-phase flow.

Fig. 4.30 shows how the void fraction changes as a function of the local heat load. It can be seen that there is considerable hysteresis between increasing and decreasing the local heat supply, leading to different values for the measured void fraction at the same value of the heating rate. Such boiling hysteresis is a common feature of increasing and decreasing heat loads due to different rates of bubble nucleation and generation on the heating surfaces as the heat load is increasing or decreasing. Accordingly, all the present experiments have been carried out, going from a high to low local heat loading. It has been found experimentally that the system is more stable and that this procedure also prevented subcooling of the bulk liquid below saturation temperature.

![Figure 4.30 Variation of void fraction as a function of heat load in steam/water in vertical flow.](image)

The effect of local heat load on the void fraction is illustrated in Fig. 4.31 for different total flow rates in vertical flow. Void fraction is higher at a lower total flow rate at constant local heat supply. It was also observed that bubble coalescence in the steam/water system takes place within a very short distance from the heater with a
subsequent rapid bubble collapse as a result of the high rate of heat and mass transfer between the phases during condensation, particularly at higher total mass flow rate. It is also evident from this figure that the nucleation of steam bubbles happens at higher heat loads as a result of higher degree of superheating. In other words, even in well-isolated tubes to be at saturated temperature does not simply mean that nucleation will happen as soon as the pressure is low enough. Therefore in vertical flows, the generation of vapour bubbles is a strong function of the degree of superheat. In air/water flows on the other hand, bubble growth and coalescence occur at much greater distances from the gas sparger and there is of course no subsequent bubble collapse as a result of condensation (note that the secondary heater has not been used in air/water experiment at higher temperatures).

![Graph showing the effect of heat load on the void fraction at different total flow rates.](image)

**Figure 4.31** Effect of heat load on the void fraction at different total flow rates.

The variation of void fraction in steam/water flow, as a function of the downstream distance, L/D, is illustrated in Fig. 4.32 which relates to different heat loads at a given total flow rates in vertical flow. It can be seen from this figure that void fraction increases as downstream length increases. This trend may be explained by the considerable vapour bubble generation near the top of the tube where the pressure continuously drops. On the other hand, steam bubbles are generated nearer to the
bottom of the tube as higher heat loads. As mentioned before this effect is consistent with the reduction of liquid flow rates as a result of change in local pressure at different liquid flow rates. Vapour bubble generation near the bottom of the tube is more probable at lower liquid flow rates. As heat load increases the point at which the nucleation might happen is much closer to the bottom of the tube. Fig. 4.33 illustrates the nucleation curve at a given flow rate and different heat loads corresponding to the data which are shown in Fig. 4.32. It is apparent that at lower heating rates, nucleation of vapour bubble occurs at higher values of L/D. The results shown in this section will then be compared with the theoretical model in the next chapter (section 5.2) based on the assumption that vapour bubbles are produced as a result of degree of superheat which they have at the bottom of the tube and pressure drop along the pipe. Another parameter which can intensify the generation of vapour bubbles would be the supply of additional heat to this superheated flow causing more vaporisation along the tube. A theoretical model which includes the effect of such a heated wall will be presented in section 5.1.

Fig. 4.34 shows the variation of void fraction in horizontal section at a given point, downstream the pipe. A similar trend was found to that in vertical flow. The void fraction is expected to approach unity as the regime changes to annular flow regime.

Figure 4.32 Influence of heat load on variation of void fraction with respect to the downstream distance.
Figure 4.33 Nucleation curve for steam/water in vertical tube.

Figure 4.34 Variation of void fraction with respect to heat load at different liquid flow rate.
The air/water and steam/water data will quantitatively be compared in chapter 5 together with data about the local pressure and temperature along the tube. This cannot be done at this stage, because the calculation of quality needs information about the local temperature and pressure along the tube. However as promised before, a phenomenological comparison can be made between the two systems as they pass through a venturi. Fig. 4.35 represents a comparison of the two systems at the same flow rate. Air/water results correspond to the data shown in Fig. 4.29 at ambient and elevated temperatures. It is evident that the characteristics of flow are utterly distinct in the two systems. As stated before, air/water data exhibit a minimum point at the throat point, while an abrupt increase in the void fraction can be seen at the throat. The drastic increase in the void fraction for steam/water flow is related to the effect of flow flashing, in which considerable amount of vapour volume can be produced as a result of decrease in pressure. Therefore it can be concluded that the distinctive behaviour of the two systems is more evident where the flow passes an obstruction such as a venturi throat.

![Figure 4.35](image-url)  
Figure 4.35 Comparison of air/water and steam/water two-phase flows pass through a venturi at a given liquid flow rate.
4.5 Conclusions

The results presented in this chapter highlight some important features of the effects of temperature on void fraction, evolution of the flow structure, and the distinctive behaviour of air/water and steam/water systems, which are summarised as follows:

- The effect on the void fraction of increasing vapour pressure at elevated temperatures cannot be ignored even at high liquid flow rates. The experimental data show substantial influence of temperature, especially above 60 °C. This effect becomes dominant near the boiling point. The modified volumetric flow ratio (equation [4.4]) adequately includes the effect of vapour pressure on the void fraction at higher flow rates. The results also reflect a change in flow regime as temperature rises due to a sharp increase of bubble size with temperature.

- The recorded images show that the chance of bubble clustering becomes unlikely at higher temperature. This is probably due to weak boundary layer of the partially saturated air bubbles leading to rapid bubble coalescence (Figs. 4.12 and 4.13). It is also shown experimentally that the change in flow regime is caused by bubble coalescence at lower temperature, however it would be affected by growth and subsequent coalescence, as the air bubbles saturate at elevated temperatures. This phenomenon is more significant at greater downstream distances.

- The most distinctive characteristic of air/water and steam/water was observed at the throat of a venturi, in which there is an abrupt increase of the void fraction as a result of depressurisation. The magnitude of this depends on the degree of superheat in the liquid.
CHAPTER 5

THEORETICAL STUDY
THEORETICAL STUDY

This chapter deals with steam/water one-component two-phase flow and presents a new one-dimensional theoretical model, aiming to predict average void fraction and pressure gradient along a pipe in a vertical heated tube. The following is closely based on the model which is developed by Lisseter et al. (1999a), which predicts void fraction and pressure drop of saturated water in a vertical unheated tube. These models are called simplified models as they are based on the fundamentals of two-phase flows of steam/water systems in vertical tubes, simplifying the terms which are negligible in compare to the other terms. In the field of two-phase flows, such models are needed as they are not only simple in form but are also realistic. Simplified models are also useful when sophisticated computing facilities are not readily available.

The simplification will be done in a rigorous manner based on the asymptotic methods. Asymptotic methods based on scale analysis will be used to determine which terms in the derived equations are small and may therefore be neglected. The validity of the resulting set of equations will then be tested by using them to obtain a simple expression for the void fraction and pressure gradient in steady steam/water bubbly flow.

This chapter starts by developing a set of equations for prediction of void fraction along a heated vertical tube for saturated vapour/liquid flows. There is an additional change in void fraction in the steam/water case due to evaporation as the pressure decreases along the pipe as well as generation of the steam bubbles due to the heat load from the wall. This follows with a brief discussion of the prediction of void fraction in an unheated tube, but with assumptions that the water is superheated at the bottom of the tube. The theoretical results will then be compared with the steam/water experimental data. Therefore the common assumption in both cases of heated and unheated tube is that the water is superheated at the bottom of the tube, but the difference is the additional heat from the wall to the flow. This chapter ends with a comparison of the results in air/water and steam/water for heated and unheated tube.
5.1 Changes in Void Fraction in Vertical Saturated Vapour/Liquid Flows in a Heated Tube

Steam/water bubbly flow in vertical channels is encountered in many industrial applications, particularly in power plants (Houghton, 1961 and Zeitoun et al., 1994). This section aims to predict the variation of void fraction due to evaporation as a result of combined effect of pressure change and heating the tube. It is also intended to compare the results with a model, in which the void fraction will be predicted for an unheated tube.

The model considers saturated vapour/liquid bubbly flows passing up an heated vertical tube. In such cases, as the saturated liquid passes along the tube the pressure falls and hence so does the boiling point. However, boiling would already occur much closer to the bottom of the tube because of the heat load. These two effects together, cause liquid to turn into vapour and so the local void fraction increases nearer to the top of the tube, as does the velocity of the liquid and vapour mixture. The local void fraction may also change due to changes in gas density caused by changes in local pressure and temperature. However the contribution of this term is small as the changes in gas density are only marginal compared to the terms caused by the local pressure and heating the tube. In this section, an expression has been derived showing how the void fraction depends on the heat load and the local pressure. However this cannot explicitly be solved because in two-phase flows the pressure is in turn dependent on the void fraction profile downstream. Thus an expression is posed for the pressure in vertical flows and iterate between this and the expression for the void fraction. An explicit expression has finally been derived for the void fraction and the pressure as a function of distance along the tube.

The precise situation that we consider is as follows. It is assumed that a flow of saturated water enters the bottom of a vertical heated tube as shown in Fig. 5.1. The pressure is above atmospheric, and the temperature of the liquid is initially above 100 °C. but below the boiling point that would correspond to the local pressure. As the water travels along the tube the pressure falls and so does the boiling temperature. This phenomenon in which liquid turns into the vapour will certainly be intensified by the
effect of heat load. As the liquid and vapour mixture moves along the tube both temperature and pressure fall further as more liquid evaporates.

It is also assumed that at all points, once boiling has been initiated the generation of vapour maintains the liquid temperature precisely at the boiling point that corresponding to the local pressure, i.e. the transfer of enthalpy to latent heat of vaporisation maintains the matching of liquid and vapour phase temperatures. Changes in liquid and gas densities are considered to be negligible. We assume that the liquid and gas mixture is homogenous, which is valid up until the bubbly flow breaks down into churn/annular flow (Lisseter et al., 1999b; Zeitoun et al., 1994).

When modelling this situation it would not be useful to start by predicting when boiling would occur and taking the void fraction to be zero up until that point. This is because we do not know the location at the first boiling point because that pressure is dependent on the void fraction distribution downstream, which is unknown at the start of the calculation. This problem could be avoided by iterating and integrating values of void fraction and pressure up and down the tube. However, this has not been done because of the aim to compare the predictions of the void fraction with the air/water models, in which the solution was with knowledge of the void fraction and pressure at the end of the tube. Thus, this model defines void fraction and pressure at the end of the tube and
The Nine Variables and Nine Equations

integrates back down the tube to determine the void and pressure profiles along the tube together with the point at which boiling commences.

The first part of this chapter sets out the nine variables and governing equations that make up the new simplified models, which will be solved in the rest of this chapter. The solution procedure starts in section 5.1.2, when eight of the equations will be used to derive an expression for the local void fraction in terms of the local pressure. It follows with the solution of the non-linear, non-homogenous differential equation, which is derived from the above equations. The change in the density of the dispersed phase (steam) is neglected as it is found (steam tables) that the change is only marginal. This equation expresses the local void fraction in terms of the pressure profile downstream. Section 5.1.3 considers the pressure expression for vertical flow and iterates between it and the expression for the local void fraction in terms of the pressure to obtain explicit expressions for the void fraction and pressure in terms of the distance from the end of the tube.

5.1.1 The Nine Variables and Nine Equations

This section presents the variables and the equations for both cases of heated and unheated tube, which follows closely that presented by Lisseter et al. (1999a) and Seward (1989). The nine local variables that will appear in the succeeding equations throughout this chapter are as follows. The liquid and gas volume fractions are $\varepsilon_L$ and $\varepsilon_G$. The liquid and gas densities are $\rho_L$ and $\rho_G$. The liquid and gas velocities are $u_L$ and $u_G$. The pressure is $p$, the temperature is $T$ and evaporation rate is $\Gamma$ (in kgm$^{-1}$s$^{-1}$). Thus nine equations are needed to solve for these nine variables. However among these variables in the first section of this chapter the changes in gas and liquid densities have been ignored (see Appendix D where the Lisseter et al. (1999)'s work is presented in more detail).

The first equation is an approximate equation, expressing the pressure in terms of the void fraction. Suitable expressions for the pressure gradient in bubbly flow can be obtained from Seward (1989) by adding together the two-fluid momentum equations for the liquid and gas phases. The pipe diameter is $D_h$ and $\eta_L$ is the liquid viscosity. The
subscript ‘x’ represents differentiation with respect to the distance, x. It is measured from the downstream end of the tube, where the pressure has the value \( p_a \) as illustrated in Fig 5.1. Because the distance is taken as being in the opposite direction to the direction of flow, the pressure gradients are positive. In vertical flow:

\[
\frac{d}{dx} \left( \rho_L \varepsilon_L u_L^2 \right) + \varepsilon_L \frac{dp}{dx} = -\left( \frac{\Gamma}{\mu} \right) u_L + \rho_L \varepsilon_L g + \tau_i \tag{5.1}
\]

Where \( \Gamma \) is the rate of evaporation and \( \tau_i \) is the interfacial stress. This expression for the pressure gradient is based on two-fluid equation set, with evaporation and neglecting terms, which are small, such as the momentum of the gas, and the wall friction.

The second equation is merely:

\[
\varepsilon_G + \varepsilon_L = 1 \tag{5.2}
\]

The third equation relates the pressure to the temperature. It is more convenient to consider only the region of flow in which boiling is occurring. Thus the temperature is the boiling temperature that corresponds to the pressure. Atkins (1982) stated that the addition of energy to a system at its transition temperature is used in driving the transition rather than raising the temperature. That is, the system has only one degree of freedom, so as the water passes up the tube and its pressure decreases, the temperature of the water must track the boiling temperature.

At the end of the tube, the pressure is atmospheric, \( p_a \), which has the value of \( 1.0135 \times 10^5 \) Pa at the corresponding temperature, \( T_a \), of 100 °C. It is also evident from tables that when the pressure is \( 1.2082 \times 10^5 \) Pa then the temperature is 105 °C. Thus the linearised equation relating the pressure and temperature is:

\[
\frac{T - T_a}{p - p_a} = m = 2.568 \times 10^{-4}, \; \text{K} \; \text{Pa}^{-1} \tag{5.3}
\]

It should be pointed out that the most realistic relationships between temperature and pressure are exponential rather than linear. Nevertheless a linear relationship may be assumed because the changes in temperature over which vaporisation is complete are small, which helps the simplicity of the following derivation. Eq. [5.3] may also be written:

\[
(T - T_a) = (m p_a) \tilde{p} = 26.03 \; \tilde{p}, \; \text{K} \tag{5.4}
\]
where
\[ \tilde{p} = \frac{p - p_a}{p_a} \]

is the dimensionless pressure. Alternatively, it may be written:
\[ T = m(p + c_1) \quad \text{K} \quad [5.5a] \]
where
\[ c_1 = 2.88 \times 10^5 \quad \text{Pa} \quad [5.5b] \]

This result from steam tables is consistent with the Clapeyron equation. This equation gives that the relationship between the vapour pressure and the boiling temperature is such that changes in pressure of 0.1 atm result in changes of temperature of 2.5 K.

The fourth equation expresses how the liquid density changes with temperature and pressure. It can be found from steam tables that when the pressure is \( p_a \) and the temperature is \( T_a \), the liquid density is:
\[ \rho_{L_a} = 975.85 \text{ kg m}^{-3} \quad [5.6] \]

When the pressure is \( 1.208 \times 10^5 \text{ Pa} \) and the temperature is \( 105 \text{ °C} \) then \( \rho_L = 954.2 \text{ kg m}^{-3} \). Thus the linearised equation relating density and saturation pressure is:
\[ \left( \frac{\rho_L - \rho_{L_a}}{\tilde{p}} \right) = 1.876 \times 10^{-4} = m_2 \quad [5.7] \]

This may also be written:
\[ \rho_L = \rho_{L_a} \left[ 1 + \left( \frac{m_2 p_a}{\rho_{L_a}} \right) \tilde{p} \right] \quad [5.8a] \]
where
\[ \frac{m_2 p_a}{\rho_{L_a}} = 0.0198 = \delta_1 \quad [5.8b] \]

The fifth equation is also obtained from steam tables. It describes how the gas density varies with pressure. When the pressure is \( p_a \) and the temperature is \( T_a \), the gas phase density is:
\[ \rho_{G_a} = 0.5978 \text{ kg/m}^3 \quad [5.9] \]

When the pressure is \( 1.2082 \times 10^5 \text{ Pa} \) and the temperature is \( 105 \text{ °C} \) then \( \rho_G \) is 0.7045 kg/m\(^3\). Thus the linearised equation relating density and pressure is
\[
\frac{\rho_G - \rho_{Ga}}{p - p_a} = m_3 = 5.48 \times 10^{-6}
\]  
[5.10]

This may also be rewritten as:

\[
\rho_G = \rho_{Ga} \left[ 1 + \left( \frac{m_3 p_a}{\rho_{Ga}} \right) \right]
\]  
[5.11a]

in which

\[
\frac{m_3 p_a}{\rho_{Ga}} = 0.929 = \gamma
\]  
[5.11b]

Alternatively, the ideal gas equation could have been used to calculate the gas density. This equation is:

\[
pV = nRT
\]  
[5.12]

where \(V\) is the volume of the gas, \(n\) is the number of moles of the gas and \(R\) is the ideal gas constant. For a given bubble, if \(M\) is the mass of gas in the bubble, then

\[
V = \frac{M}{\rho_G}
\]  
[5.13]

Substituting this in [5.12] gives

\[
\frac{1}{T} \frac{p}{\rho_G} = \frac{nR}{M} = \text{constant}
\]  
[5.14]

Considering [5.5a] yields:

\[
\frac{1}{m_1 (p + c_1)} \frac{p}{\rho_a} = \text{constant}
\]  
[5.15]

so

\[
\frac{\rho_G}{\rho_{Ga}} = \frac{p (p_a + c_1)}{p_a (p + c_1)}
\]  
[5.16a]

Substituting the above values of \(\rho_{Ga}, p_a\) and \(c_1\) leads to the fact that when \(p\) is equal to \(1.2082 \times 10^5\) Pa then \(\rho_G\) has the value 0.6787 kg m\(^{-3}\), compared with the actual value of 0.7045 kg m\(^{-3}\). Thus we conclude that it is better to use [5.11a] for \(\rho_G\) than the ideal gas equation.

A more accurate equation of state than the ideal gas equation is the Van der Waals equation,

\[p = nRT(V-nb)-an^2/V^2\]  
[5.16b]
where \( a \) and \( b \) are appropriate constants (for water, \( a=0.554 \text{ m}^6\text{kgm}^{-1}\text{s}^{-2}\text{mol}^{-2} \) and \( b=3.05 \times 10^{-5} \text{ m}^3 \text{ mol}^{-1} \)). However, under the conditions of the current study, in which liquid and gas coexist, this equation exhibits loop iteration which must be removed by the Maxwell construction (Atkins, 1982). It is therefore simpler to use [5.11a] for \( \rho_G \).

The next equations are related to the rate of evaporation. The sixth equation is the equation for conservation of mass of gas. This is:

\[
\Gamma = (A \rho_G \rho_G u_G)_x \quad [5.17a]
\]

Where in this equation \( A \) is the cross-sectional area of the tube. The seventh equation is that of conservation of mass for the liquid phase. It is:

\[
-\Gamma = (A \rho_L \rho_L u_L)_x. \quad [5.17b]
\]

The eighth equation gives an expression for \( \Gamma \). To derive this, the first law of thermodynamics will be used which says that the energy of an isolated system is constant. We will consider the energy involved in the various processes going on as the liquid and gas mixture flows up the pipe, and then relate the energies by putting the sum equal to zero. It will be also assumed that the pipe is well lagged so that no heat is lost from the system.

It has already been noted that the liquid and gas system has only one 'degree of freedom' when it is boiling, so that as the flow rises up the pipe and the pressure decreases, the temperature of the liquid must also decrease. For a decrease in temperature \( \Delta T \), the internal energy of the liquid decreases by an amount \( C_v \Delta T \), where \( C_v \) is the specific heat (heat capacity) of the liquid, which has the value of \( 4.18 \times 10^3 \text{ Jkg}^{-1} \text{ K}^{-1} \). From steam tables, and hence [5.8b], it can be seen that the change in volume of water as it cools about the boiling point is small, and so the work done (resulting in a change in internal energy) is negligible.
Again assuming that the system has only one degree of freedom, we deduce that any energy liberated by the cooling of the liquid, must go into promoting a phase change. Steam tables give us the change in internal energy on evaporation, $U_{LG}$. This is dependent on the local pressure. At $1.0135 \times 10^5$ Pa, $U_{LG}$ is $2.0876 \times 10^6$ J kg$^{-1}$, while at $1.208 \times 10^5$ Pa, $U_{LG}$ is $2.0723 \times 10^6$ J kg$^{-1}$. The change is small, compared with the size of $U_{LG}$, so the average of these values is used, namely $2.08 \times 10^6$ J kg$^{-1}$, and assumed that $U_{LG}$ is constant over the range of pressures under consideration. Note that $U_{LG}$ can be calculated from the enthalpy of vaporisation ($\Delta h_v$, the latent heat) and the change in volume on evaporation, as in Aktins (1982) $U_{LG}$ is approximately 10% less than the latent heat because the gas has done work in expanding from the volume it took up as a liquid to that which it occupies at its vapour pressure.

The above values of $U_{LG}$ assume that the gas is liberated from the liquid, whereas in our case the gas forms local bubbles. Thus some energy goes into creating surfaces between the liquid and gas phases. The work done in doing this is given by the value of the surface tension (at 100 °C, 0.058 Nm$^{-1}$), multiplied by the interfacial area (Atkins, 1982). Assuming that all the bubbles created have a radius, $R$, of 0.003 m, and are spherical, then the surface area per bubble is $4\pi r^2$. When this is multiplied by the value of the surface tension, it can be found that the work done per bubble is $6.55 \times 10^{-6}$ J. The volume of a single bubble is $1.13 \times 10^{-7}$ m$^3$. Thus, since 1 mole of gas (0.018 kg) occupies $24.5 \times 10^{-3}$ m$^3$, the mass of one bubble must be $8.30 \times 10^{-8}$ kg. It can also be found that the work done per kilogram (by dividing the specific work done by the mass of the bubble) is 79 J kg$^{-1}$, which is much less than $U_{LG}$. In fact, after an initial occurrence of nucleation, the subsequent evaporation of water vapour will feed the existing bubbles, which will increase in size. When this happens the increase in surface area will be less than if new bubbles were created, so the above estimate of the work done is too large. Hence this component of energy use may be neglected.

Finally, it can be assumed that the density of the gas changes, as its temperature and pressure adjust to that of its surroundings. This could be modelled by the ideal gas law or equation [5.11]. The work done in adjusting this density may be shown to be negligible compared with that involved in evaporation. This assumption will be
confirmed later by the finding that the parameter for expansion of gas ($\gamma$) is much less than that for evaporation.

The first law of thermodynamics thus allows to equate the energy lost as the liquid cools ($C_v \Delta T$) with the energy ($U_{LG}$) used to evaporate the liquid at that pressure and temperature, other effects being negligible, as discussed above. The additional term is the rate of vapour generated due to the heat load, which is constant along the tube. A decrease in temperature along the tube in the negative ‘x’ direction causes a loss of liquid by evaporation, so:

\[- \Gamma = A \rho_L u_L \varepsilon_L \left( \frac{C_v}{U_{LG}} \right) T_x - \frac{\dot{Q} L}{U_{LG}}.\]

in which:

\[\dot{q} = \frac{\dot{Q}}{A}\]

\(\dot{Q}\) is total input heat load (W), A is the uniform cross sectional area of the pipe and L is the total length of the tube.

The last equation makes the assumption that the liquid and gas phases travel at the same velocity. This is known as the homogenous assumption, and is valid for bubbly flow. Lisseter et al. (1999b) have shown that when the homogenous assumption is made, then predictions of the void fraction in bubble flow differ from their true values by less than 10%. The hypothesis of homogeneity is also assumed and justified by other investigations (Houghton, 1961 and Zeitoun et al., 1994). The homogenous equation is:

\[u_L = u_G.\]

5.1.2 Deriving an Expression for the Local Void Fraction in Terms of the Local Pressure

In this section the variables will be determined that arise in the above equations to derive an expression for the local void fraction in terms of the local pressure. This may be started by eliminating $\Gamma$ between [5.17] and [5.18]:

\[- \Gamma = A \rho_L u_L \varepsilon_L \left( \frac{C_v}{U_{LG}} \right) T_x - \frac{\dot{Q} L}{U_{LG}}.\]
\[(\Delta \rho L \varepsilon_L u_L)_x = (\Delta \rho L \varepsilon_L u_L) \left( \frac{C_v}{U_{LG}} \right) T_x - \frac{q_L}{U_{LG}} \]  \[5.20\]

The above equation is a non-linear, non-homogenous ordinary differential equation. The non-linearity comes from the fact that temperature depends upon pressure and subsequently distance along the pipe. To make the above equation and its solution simple, consider that:

\[y = \rho_L \varepsilon_L u_L \]  \[5.21\]

From Eq. [5.20] we can get:

\[y \lambda = y \lambda \frac{dT}{dx} - N \]  \[5.22\]

where:

\[\lambda = \frac{C_v}{U_{LG}} \]  \[5.23\]

and

\[N = \frac{q_L}{U_{LG}} \]  \[5.24\]

Eq. [5.22] can be re-written to:

\[\left[ y e^{-\lambda T} \right]_x = -N e^{-\lambda T} \]  \[5.25\]

We then integrate and apply boundary conditions at points along the tube, starting from the top towards downstream the tube:

\[\left[ y e^{-\lambda T} \right]_0^x = -N \int_0^x e^{-\lambda T} dx \]  \[5.26\]

and subsequently:

\[y e^{-\lambda T} = y_0 e^{-\lambda T_0} - N \int_0^x e^{-\lambda T} dx \]  \[5.27\]

Then both sides can be divided by the term \((e^{-\lambda T})\), so the left hand side in the above equation is only a function of void fraction and liquid velocity:

\[y = y_0 e^{\lambda (T - T_0)} - N e^{\lambda T} \int_0^x e^{-\lambda T} dx \]  \[5.28\]

The integral in the right hand side cannot easily be solved, as it is unknown how the temperature changes with the pressure along the tube. Moreover, as far as the
applicability of the theory is concerned, the relationship between either of temperature or pressure with the distance is more convenient than that of pressure and temperature themselves. Eq. [5.3] can be used in order to find a relationship between temperature and pressure, which is:

\[ T = T_a + m(p - p_a) \]  

In which subscript “a” corresponds to the atmospheric conditions at 100 °C and “m” is small value. Subtracting \( T_0 \) from the above equation:

\[ (T - T_0) = (T_a - T_0) + m(p - p_a) \]  

Again, it should be pointed out that this is necessary, as there is no information on the variation of temperature with the distance in a heated vapour/liquid two-phase flow. Substituting [5.30] into [5.28] leads to:

\[ y = y_0 e^{[\lambda(T_a - T_0) + m(p - p_a)]} \left[ Ne^{-[\lambda(T_a + m(p - p_a))] \int_0^x} \right] e^{-[\lambda(T_a + m(p - p_a))] \int_0^x} \]  

All parameters inside the integral are constant except pressure that depends upon distance along the tube. This can be regarded as a second-degree polynomial. The assumption of second-order equation will be justified in the following section that shows the coefficient of the terms above \((x^2)\) are negligible:

\[ p - p_a = C_0 + C_1 x + C_2 x^2 \]  

where considering \( x=0 \) and \( p=p_0 \) lead to \( C_0=p_0-p_a \). It is still not possible to integrate the second term in right hand side of Eq. [5.31], as there is no exact solution for the exponential term \((\exp(C_1x+C_2x^2))\). This term can be expanded in order to overcome this:

\[ e^\lambda m(C_1 x + C_2 x^2) \int_0^x e^{-\lambda m(C_1 x + C_2 x^2)} \, dx = \]

\[ \left[ 1 + \lambda m C_1 x + \frac{\lambda^2 m^2}{2} \left( C_1 x + C_2 x^2 \right)^2 + \text{terms in } x^3 \right]. \]  

\[ \int_0^x \left[ 1 - \lambda m(C_1 x + C_2 x^2) + \text{terms in } x^2 \right] \, dx \]
Integrating this leads to:

\[
= \left(1 + \lambda m C_1 x + \text{terms in } x^2 \right) \left[1 - \frac{\lambda}{2} m C_1 x^2 + \text{terms in } x^3 \right]_0^x \tag{5.33b}
\]

Now the order of magnitude of the various terms should be considered in order to neglect the terms with very small values. We know that \(C_v\) is in order of \((10^3)\) and \(U_{LG}\) is in \((10^6)\), so the order of magnitude of \(\lambda\) (=\(C_v/U_{LG}\)) is \((10^3)\). On the other hand, the order of magnitude of the parameter “\(m\)” is \((10^4)\) (as calculated in [5.3]). This gives the order of magnitude of the product of “\(\lambda m\)” is \((10^7)\). Therefore, the terms in and above \((x^3)\) are neglected. The discussion of this matter will be elaborated in Appendix D. Therefore Eq. [5.33a] can be simplified as:

\[
= (1 + \lambda m C_1 x)(1 - \frac{\lambda}{2} m C_1 x) + \text{terms in } x^3 \tag{5.34}
\]

The first term in the right hand side of [5.31] can be rewritten similarly as:

\[
y_0 e^{(\lambda(T_a-T_0)+m\lambda(p-p_0))} = y_0 e^{(\lambda m(p_a-p_0)+\lambda m(C_o+C_1 x+C_2 x^2))} \tag{5.35}
\]

Considering again the relationship between temperature and pressure:

\[
\frac{T_a - T_0}{p_a - p_0} = m \tag{5.36}
\]

where if \(x=0\) and \(p=p_0\), we get \(C_0=p_0-p_a\). Substituting [5.36] into [5.35] yields:

\[
y_0 e^{(\lambda(T_a-T_0))} = y_0 \left(1 + \lambda m(C_1 x + C_2 x^2) + \frac{1}{2} (\lambda m C_1 x)^2 + \text{terms in } x^3 \right) \tag{5.37}
\]

Now, replacing [5.34] and [5.37] into [5.31] leads to:

\[
y = y_0 \left(1 + \lambda m(C_1 x + C_2 x^2) + \frac{1}{2} (\lambda m C_1 x)^2 \right) - N(1 + \lambda m C_1 x)\left(1 - \frac{1}{2} (\lambda m C_1 x) \right) \tag{5.38}
\]

where \(y = \rho_L u_L \varepsilon_L\). Eq. [5.38] can simply be re-written as:

\[
\rho_L u_L \varepsilon_L = \rho_{L0} u_{L0} \varepsilon_{L0} \theta - \omega \tag{5.39}
\]

in which:

\[
\theta = \left(1 + \lambda m(C_1 x + C_2 x^2) + \frac{1}{2} (\lambda m C_1 x)^2 \right) \tag{5.40}
\]

and

\[
\omega = N(1 + \lambda m C_1 x)\left(1 - \frac{1}{2} (\lambda m C_1 x) \right) = N\left(x + \frac{1}{2} \lambda m C_1 x^2 - \frac{1}{2} (\lambda m C_1 x)^2 x^3 \right) \tag{5.41}
\]

As discussed earlier, the third term on the right hand side of Eq. [5.41] can be eliminated as the term \((\lambda m)^2\) is in order of \((10^{-14})\).
The void fraction in Eq. [5.39] is a function of local liquid velocity, a parameter that we do not have. However, it can be replaced from another equation, the total mass flux of the two-phase flow, which is:

\[ G = \rho_L u_L \varepsilon_L + \rho_G u_G \varepsilon_G \]  

[5.42]

We assume that the liquid and gas mixture is homogeneous, which is valid in the bubbly flow regime (Houghton, 1961 and Zeitoun et al., 1994), so \( u_L = u_G \). Having considered this assumption and replacing [5.42] into [5.39] gives rise to:

\[
\frac{\rho_L \varepsilon_L G}{\rho_L \varepsilon_L + \rho_G \varepsilon_G} = \frac{\rho_L \varepsilon_L \theta \varepsilon_G}{\rho_L \varepsilon_L + \rho_G \varepsilon_G} \]

[5.43]

This can be rearranged as:

\[
\frac{\varepsilon_L G}{\varepsilon_L + \varepsilon_G} = \frac{\varepsilon_L \theta \varepsilon_G}{\varepsilon_L + \varepsilon_G} \]

[5.44]

where \( R = \frac{\rho_G}{\rho_L} \)

As is also shown in Appendix D, the changes in liquid and gas densities are negligible, so we can assume that:

\[ \rho_L = \rho_{L0} \quad \text{and} \quad \rho_G = \rho_{G0} \]

[5.45]

and as a result of this:

\[
\frac{\varepsilon_L}{\varepsilon_L + \varepsilon_G} = \varphi = \frac{\varepsilon_L \theta}{\varepsilon_L + \varepsilon_G} \]

[5.46]

All terms on the right hand side of the above equation are constant or solely a function of the distance along the tube, so it is convenient to rearrange Eq. [5.46] and find a relationship between the void fraction and parameter called \( \varphi \).

\[ 1 - \varepsilon_G = (1 - \varepsilon_G) \varphi + \varepsilon_G \varphi = \varphi - \varepsilon_G \varphi + \varepsilon_G \varphi \]

[5.47]

and finally:

\[ \varepsilon_G = \frac{1 - \varphi}{1 - \varphi(1 - R)} \]

[5.48]

in which:
\[
\varphi = \frac{\varepsilon_{L,0}}{\varepsilon_{L,0} + \Re G_0} \theta - \frac{\omega}{G} \quad [5.49]
\]

Subsequently it can be written, substituting \( \theta \) and \( \omega \) from Eqs. [5.40] and [5.41]:
\[
\varphi = \Phi \left( [1 + \lambda m C_1 x + \lambda m C_2 x^2] - \Psi \left( x + \frac{1}{2} \lambda m C_1 x^2 \right) \right) \quad [5.50]
\]

In this equation:
\[
\Phi = \frac{\varepsilon_{L,0}}{\varepsilon_{L,0} + \Re G_0} = \frac{1 - \varepsilon_{G,0}}{1 - \varepsilon_{G,0} (1 - R)} \quad [5.51a]
\]

and
\[
\Psi = \frac{N}{G} \quad [5.51b]
\]

At this point some dimensionless parameters are defined as:
\[
\delta = p_a \lambda m \quad [5.52a]
\]

and
\[
\tilde{C}_1 = \frac{C_1}{p_a} \quad [5.52b]
\]

\[
\tilde{C}_2 = \frac{C_2}{p_a} \quad [5.52c]
\]

Now Eq. [5.50] will be:
\[
\varphi = \Phi \delta \tilde{C}_1 x + \Phi \delta \tilde{C}_2 x^2 - \Psi x - \frac{1}{2} \Psi \delta \tilde{C}_1 x^2
\]

\[
= \Phi \left( \Phi \delta \tilde{C}_1 - \Psi \right) x + \left( \Phi \delta \tilde{C}_2 - \frac{1}{2} \Psi \delta \tilde{C}_1 \right) x^2 \quad [5.53]
\]

or for simplicity, it can be considered as a second-order polynomial function:
\[
\varphi = f_0 + f_1 x + f_2 x^2 \quad [5.54]
\]

where:
\[
f_0 = \Phi \quad [5.55a]
\]

\[
f_1 = \Phi \delta \tilde{C}_1 - \Psi \quad [5.55b]
\]

\[
f_2 = \Phi \delta \tilde{C}_2 - \frac{1}{2} \Psi \delta \tilde{C}_1 \quad [5.55c]
\]

Replacing Eq. [5.54] into [5.48] yields:
95 Changes in Pressure of Saturated Vapour/Liquid Upward Flows in Vertically Heated Pipes

\[
\varepsilon_G = \frac{1 - \left(f_0 + f_1 x + f_2 x^2\right)}{1 - \left(f_0 + f_1 x + f_2 x^2\right)(1 - R)} \tag{5.56}
\]

The above equation is a non-linear function that is not easy to use, particularly when expressing the pressure profile as a function of void fraction. It is therefore more convenient to convert it to polynomial function. Expanding the above, using a Maclaurin polynomial approximation gives rise to:

\[
\varepsilon_G = \frac{1 - \varphi}{1 - \varphi(1 - R)} = \sum_{n=0}^{\infty} \frac{1}{n!} \varepsilon_G^{(n)}(0)x^n \tag{5.57}
\]

In this equation the term \(\varepsilon_G^{(n)}\) is the “n” order differentiation of [5.56]. Therefore it follows that:

\[
\varepsilon_G^{(0)} \big|_{x=0} = \frac{1 - f_0}{1 - f_0(1 - R)} = \frac{\mathrm{Re}_G}{\mathrm{Re}_G + \mathrm{Re}_L} = \varepsilon_G^0 \tag{5.58a}
\]

\[
\varepsilon_G^{(1)} \big|_{x=0} = \frac{f_1 (\varepsilon_G^0 (1 - R) - 1)}{1 - f_0 (1 - R)} = -\frac{f_1}{R} \left[1 - \varepsilon_G^0 (1 - R)\right]^2 \tag{5.58b}
\]

\[
\varepsilon_G^{(2)} \big|_{x=0} = -\frac{f_2}{R} \left[1 - \varepsilon_G^0 (1 - R)\right]^2 - \frac{f_1^2}{R^2} \left[1 - \varepsilon_G^0 (1 - R)\right]^3 (1 - R) \tag{5.58c}
\]

The terms above \((x^2)\) are found to be small and in the order of \(O(10^{-3})\) which can be ignored. Substituting equation [5.58] into [5.57] and simplifying this yields:

\[
\varepsilon_G = \varepsilon_G^0 - \left\{\frac{f_1}{R} \left[1 - \varepsilon_G^0 (1 - R)\right]^2\right\}x
- \left\{\frac{f_2}{R} \left[1 - \varepsilon_G^0 (1 - R)\right]^2 - \frac{f_1^2}{R^2} \left[1 - \varepsilon_G^0 (1 - R)\right]^3\right\}(1 - R)x^2 + O(10^{-3}) \tag{5.59}
\]

This is a second-order polynomial equation, which gives an explicit expression of void fraction as a function of distance from top of the tube, physical properties of the phases and pressure along the tube. In this equation pressure is the only parameter, which needs to be calculated.

5.1.3 Changes in Pressure of Saturated Vapour/Liquid Upward Flows in Vertically Heated Pipes

The change in pressure along the tube is assumed to follow a second–order polynomial equation [5.32]. We develop an independent equation (liquid momentum equation) from what has been assumed before [5.1]. The next step is to equate the coefficients in
Eq. [5.32] with the coefficients in the new equation which will be derived in the following. This should be done in order to obtain the coefficients in [5.32] as a function of physical properties and the operational parameters such as heat load, distance along the tube, total mass flux and diameter of the pipe.

We start by considering the liquid momentum equation:

$$\frac{d}{dx} \left( \rho_L \varepsilon_L u_L^2 \right) + \varepsilon_L \frac{dp}{dx} = -\left( \frac{\Gamma}{A} \right) u_L + \rho_L \varepsilon_L g + \tau_i$$ \hspace{1cm} [5.60]

In this equation, \( \Gamma \) is the rate of evaporation, which is defined in [5.18] and \( \tau_i \) is the interfacial pressure gradient which can be assumed zero as \( u_i = u_g \) (Lisseter et al. 1999b; Houghton, 1961 and Zeitoun et al., 1994). We can substitute the expression \(-\left(\Gamma/A\right)\) from Eq. [5.18] into the above equation:

$$\frac{d}{dx} \left( \rho_L \varepsilon_L u_L^2 \right) + \varepsilon_L \frac{dp}{dx} = u_L \frac{d}{dx} \left( \rho_L \varepsilon_L u_L \right) + \rho_L \varepsilon_L g$$ \hspace{1cm} [5.61]

Some of the terms in this equation can be eliminated. We start with the differentiating from the right hand side:

$$\frac{d}{dx} \left( \rho_L \varepsilon_L u_L^2 \right) = \varepsilon_L u_L^2 \frac{d}{dx} (\rho_L) + \rho_L u_L^2 \frac{d}{dx} (\varepsilon_L) + 2\rho_L \varepsilon_L u_L \frac{d}{dx} (u_L)$$ \hspace{1cm} [5.62]

and if we do the same for the first term on the right hand side in Eq. [5.61], gives:

$$u_L \frac{d}{dx} \left( \rho_L \varepsilon_L u_L \right) = \varepsilon_L u_L^2 \frac{d}{dx} (\rho_L) + \rho_L u_L^2 \frac{d}{dx} (\varepsilon_L) + \rho_L \varepsilon_L u_L \frac{d}{dx} (u_L)$$ \hspace{1cm} [5.63]

By substituting [5.62] and [5.63] into [5.61]:

$$\rho_L u_L^2 \frac{d}{dx} (\varepsilon_L) + 2\rho_L \varepsilon_L u_L \frac{d}{dx} (u_L) + \varepsilon_L \frac{dp}{dx} = \varepsilon_L u_L^2 \frac{d}{dx} (\varepsilon_L) + \rho_L \varepsilon_L u_L \frac{d}{dx} (u_L) + \rho_L \varepsilon_L \frac{d}{dx} (u_L) + \rho_L \varepsilon_L g$$ \hspace{1cm} [5.64]

and this can be simplified to:

$$\rho_L \varepsilon_L u_L \frac{d}{dx} (u_L) + \varepsilon_L \frac{dp}{dx} = \rho_L \varepsilon_L g$$ \hspace{1cm} [5.65]

Dividing the above by \( \varepsilon_L \) leads to:

$$\rho_L u_L \frac{d}{dx} (u_L) + \frac{dp}{dx} = \rho_L g$$ \hspace{1cm} [5.66]
It is convenient to define a dimensionless pressure, because we wish to compare the results of this model with that dimensionless equation of pressure [5.32], [5.51] and [5.52]:

\[ p = \frac{p - p_0}{p_a} \]  

so

\[ \frac{dp}{dx} = p_a \frac{dp}{dx} \]  

Substituting the above into Eq. [5.66] yields:

\[ \rho_L u_L \frac{d u_L}{dx} + p_a \frac{dp}{dx} = \rho_L g \]  

This is a non-linear, non-homogenous first-degree differential equation which cannot be simply solved. To overcome this, we have to find a relationship between liquid superficial velocity and distance along the tube, which from Eq. [5.42] we have:

\[ u_L = \frac{G}{\rho_L \varepsilon_L + \rho_G \varepsilon_G} = \frac{G}{\rho_L \left(1 + \varepsilon_G (R - 1)\right)} \]  

In this equation \( \varepsilon_G \) can be replaced from [5.59]:

\[ \rho_L u_L = \frac{G}{1 + \varepsilon_G (R - 1)} = \frac{G}{1 + \left(d_0 + d_1 x + d_2 x^2\right)(R - 1)} = \frac{G}{k_0 + k_1 x + k_2 x^2} \]  

where coefficients \( d_0, d_1 \) and \( d_2 \) are defined in Eq. [5.58] and:

\[ k_0 = 1 + d_0 (R - 1) \]  

\[ k_1 = d_1 (R - 1) \]  

\[ k_2 = d_2 (R - 1) \]  

Eq. [5.71] can be rewritten as:

\[ u_L = \frac{g_1}{k_0 + k_1 x + k_2 x^2} \]  

where:

\[ g_1 = \frac{G}{\rho_L} \]  

By differentiating of Eq. [5.73] respect to "x":

\[ \frac{du_L}{dx} = -\frac{g_1 \left(k_1 + 2 k_2 x\right)}{\left(k_0 + k_1 x + k_2 x^2\right)^2} \]
Substituting [5.73] and [5.75] into [5.69] gives:

\[
\frac{G}{k_0 + k_1x + k_2x^2} \cdot \frac{g_1(k_1 + 2k_2x)}{(k_0 + k_1x + k_2x^2)^2} + p_a \frac{dp}{dx} = \rho_L g
\]  

[5.76]

or

\[
dp\sim = A \frac{(k_1 + 2k_2x)}{(k_0 + k_1x + k_2x^2)^3} + B
\]  

[5.77]

in which:

\[
A = G g_1 = \frac{G^2}{\rho_L p_a}
\]  

[5.78a]

\[
B = \frac{\rho_L g}{p_a}
\]  

[5.78b]

Integrating Eq. [5.77] for the boundary conditions of 0 (top of the tube) to x (any point along the tube):

\[
\int_0^x dp = \int_0^x \left( A \frac{(k_1 + 2k_2x)}{(k_0 + k_1x + k_2x^2)^3} + B \right) dx
\]  

[5.79]

\[
\sim p = \frac{A}{2} \frac{1}{(k_0 + k_1x + k_2x^2)^2} + \frac{A}{2} \frac{1}{k_0^2} + Bx
\]  

[5.80]

We use the Maclaurin polynomial approximation again to obtain a linear polynomial expression for the pressure along the tube. This has to be done because we wish to equate the coefficients of [5.32] with those of derived from [5.80].

\[
\sim p = -\frac{A}{2} \left[ \frac{1}{k_0} - \frac{2k_1}{k_0^2} \right] x + \left\{ \frac{4k_1^2}{k_0^4} - \frac{(2k_0k_2 + k_1^2)}{k_0^4} \right\} x^2 + \frac{A}{2} \frac{1}{k_0^2} + Bx
\]  

[5.81]

or

\[
\sim p = -\frac{2Ak_1}{k_0^3} x + B \left( \frac{A}{2} \left[ \frac{3k_1^2 - 2k_0k_2}{k_0^4} \right] \right) x^2
\]  

[5.82]

which can be rewritten as:

\[
\sim p = A_1x + A_2x^2
\]  

[5.83]

in which:
Changes in Pressure of Saturated Vapour/Liquid Upward Flows in Vertically Heated Pipes

\[ A_1 = \left[ \frac{2A k_1}{k_0^3} + B \right] \]  \hspace{1cm} [5.84]

\[ A_2 = -\left( \frac{A}{2} \right) \left[ \frac{3k_1^2 - 2k_0k_2}{k_0^4} \right] \]  \hspace{1cm} [5.85]

Meanwhile we knew from Eq. [5.32] that \( \tilde{p} = C_1 x + C_2 x^2 \). Equating [5.32] and [5.83] gives:

\[ \tilde{C}_1 = A_1 = \left[ \frac{2A k_1}{k_0^3} + B \right] \]  \hspace{1cm} [5.86a]

\[ \tilde{C}_2 = A_2 = -\left( \frac{A}{2} \right) \left[ \frac{3k_1^2 - 2k_0k_2}{k_0^4} \right] \]  \hspace{1cm} [5.86b]

where:

\[ k_0 = 1 + \varepsilon_{G0}(R - 1) \]  \hspace{1cm} [5.87a]

\[ k_1 = d_1(R - 1) = -\frac{f_1}{R} k_0^2(R - 1) \]  \hspace{1cm} [5.87b]

\[ k_2 = d_2(R - 1) = -\left[ \frac{f_2}{R} k_0^2(l - R) + \frac{f_1^2}{R^2} k_0^3 \right](R - 1) \]  \hspace{1cm} [5.87c]

\[ f_1 = \Phi \delta \tilde{C}_1 - \Psi \]  \hspace{1cm} [5.87d]

\[ f_2 = \Phi \delta \tilde{C}_2 - \frac{\Psi}{2} \delta \tilde{C}_1 \]  \hspace{1cm} [5.87e]

\[ \Phi = \frac{1 - \varepsilon_{G0}}{1 + \varepsilon_{G0}(R - 1)} \]  \hspace{1cm} [5.87f]

\[ \delta = \lambda m_{pa} = \frac{C_{v}}{U_{LG}} m_{pa} \]  \hspace{1cm} [5.87g]

\[ \Psi = \frac{N}{G} \text{ where } N = \frac{q}{L U_{LG}} \]  \hspace{1cm} [5.87h]

\[ A = G g_1 = \frac{G^2}{\rho_{L} p_{a}} \]  \hspace{1cm} [5.87i]

and

\[ B = \frac{\rho_{L} g}{p_{a}} \]  \hspace{1cm} [5.87j]
However it is more convenient to find an explicit expression for $\tilde{C}_1$ and $\tilde{C}_2$ in relation with the operational parameters such as heat load and physical properties of the system. To do this, we replace parameters in Eq. [5.86] by evidence as from Eq. [5.87], which leads to:

$$\tilde{C}_1 = A_1 = \frac{2Ak_1}{k_0^3} + B = -2A \frac{f_1}{R} \frac{k_0^2}{k_0^3} (R - 1) + B$$

and finally:

$$\tilde{C}_1 = \frac{2qLG(R - 1) + \rho_l^2 U_{LG} g R k_0}{2\Phi m \rho C_v \rho_s G^* (R - 1) + R k_0 \rho_l \rho_s U_{LG}}$$

The same procedure gives:

$$\tilde{C}_2 = \frac{\left(\frac{f_1}{R}\right)^2 (R - 1)^2 - \frac{1}{k_0} \frac{(R - 1)}{R} \Psi \delta \tilde{C}_1}{1 + \frac{A (R - 1) \Phi \delta}{k_0 \rho C_v \rho_s G^* (R - 1)}}$$

It is worthwhile noting that $\tilde{C}_1$ is independent of the heat load, which, however has an effect on the second coefficient ($\tilde{C}_2$).

In this study no experiments have been performed for a heated tube. However the above equations will be validated by a phenomenological comparison with the results of the following section. However the accuracy deteriorates, when the comparison is made at low liquid flow rates.

Fig. 5.2 shows the variation of the void fraction as a function of distance from top of the tube at different incoming liquid velocities for a pipe with 24.2 mm in diameter. It is apparent, as expected that an increase in liquid velocity leads to sharp decrease in the void fraction. The variation of void fraction with distance along the tube at different head load is illustrated in Fig. 5.3. It can be seen that the nucleation occurs nearer to the top of the tube as the heat load decreases. It should be pointed out that the above figures are based on the specified void fraction and pressure at the top of the tube. More
comparisons for the above equation will be made in the following section with the result of theory at which void fraction is calculated in an unheated tube and the air/water theory at ambient temperature.

Figure 5.2 Effect of liquid velocity on the variation of void fraction in a heated tube.

Figure 5.3 Effect of heat load on the variation of void fraction in a heated tube.
5.2 Changes in Void Fraction due to Changes in Pressure for Saturated Vapour/Liquid Flows in an Unheated Tube

This section briefly describes the model that is posed by Lisseter et al. (1999a) in which the void fraction is calculated as a function of distance along the unheated tubes. The solution procedure is based on the set of equations presented in section 5.1.1 from [5.1] to [5.20]. As far as the definition of problem is concerned, there are two differences between what will be described in this section and the model derived in the previous section. The first one is that the generation of steam bubbles is ignored due to heat load. Therefore the second term in the right hand side of [5.19] can be ignored. The second difference is that the changes in the vapour density are allowed, though as it is mentioned before these changes are negligible.

The derivation of void fraction begins by equating [5.17] and [5.18], in which the rate of evaporation (Γ) can be eliminated. This gives an expression for the liquid velocity at different points along the tube as a function of void fraction. The rate of evaporation is eliminated again between [5.18] and [5.19] which gives a homogenous differential equation between void fraction, local liquid velocity and temperature downstream the tube. The liquid velocity is then deleted between these two equations. However the above differential equation cannot be easily solved as it is not known here the temperature changes between different axial positions. Therefore a relationship between temperature and pressure [5.3] has to be included in this differential equation. The final equation is a relationship between the pressure gradient and the void fraction and has only the pressure as an unknown parameter.

The solution proceeds by developing an equation for the prediction of pressure in relation with the distance along the tube. This starts by consideration of equation [5.1] which is the momentum equation. In this section, changes in vapour density are allowed. The solution of a simplified form of Eq. [5.1] leads to a polynomial, which is a function of void fraction. The terms with small values compared to the others are deleted throughout the derivation of void fraction and pressure gradient. Void fraction and pressure profile can be calculated along the tube by iteration between these two sets of equations because each equation is depended upon the other one. The last part of this model is concerned with prediction of the point at which the liquid phase turns into the vapour phase. The full explanation of the
Validation of the Model by Comparison with Experimental Data

As mentioned in chapter 3, steam bubbles are generated by the secondary local heater at the entrance of the test section. All data have been taken at steady state conditions, keeping temperature and flow rate constant. The liquid flow rate was measured before the...
test section. As mentioned before, it is also more convenient to refer to the liquid flow rate as the total flow rate since it is a proportion of this liquid, which is converted into steam and provides the vapour component of the two-phase flow.

The variation of void fraction as a function of the downstream distance, $L/D_h$, is illustrated in Fig. 5.4 which relates to different local heat loads for a fixed total flow rate. It can be seen from this figure that void fraction goes up as downstream length increases. This trend can be attributed to the fact that steam bubbles are generated at higher downstream distance to reduce superheated liquid to its boiling temperature. At lower heat loads, the generation of saturated steam bubbles takes place at a significantly greater distance from the heater, while for a higher heat load, due to the higher rate of the vapour generated, it happens very close to the heater (for instance at 5.14 kW in this figure). The comparison between experimental data and theory is shown in Fig. 5.5 for two different flow rates. It is evident that the agreement between the experiment and theory is good, particularly at higher downstream distance. These results will be compared with the theoretical results of the effect of heat load in the following section.

![Graph showing variation of void fraction with respect to downstream distance measured at different heat loads.](image)

**Figure 5.4** Variation of void fraction with respect to downstream distance measured at different heat loads.
5.3 Comparison between Air/Water and Steam/Water

As promised before a comparison should be made between air/water and steam/water flows in heated and unheated tubes. We now recall the results described in the paper by Lisseter and Fowler (1992) in which the change in void fraction and pressure is analysed with respect to distance in air/water flows at ambient temperature. In such circumstances, there is no evaporation only expansion of gas due to change in pressure (which is almost negligible for the circumstances of this study). It is shown that the variation of void fraction with distance down along the pipe is:

$$\varepsilon_G = \varepsilon_{GO}(1 + S(1 - \varepsilon_{GO})\lambda)$$

where $S = (\rho_L g)/\rho_a$  \[5.93\]

$$\frac{p}{p_0} = \frac{\varepsilon_{GO}}{\varepsilon_G}$$

\[5.94\]

We now consider the situation in which there are two identical pipes, one containing the steam/water flow and the other the air/water flow. Both pipes are lagged so that there is no heat loss. The flows are adjusted in a way that at the ends of both pipes, the void fraction is $\varepsilon_{GO}$ and the pressure is $p_a$. Then equations [5.59], [5.92] and [5.93] tell us that in vertical upward flow, the void fractions before the end of the pipe will be different in the two different cases of steam/water and air/water flows. A comparison of the equation [5.93] for
air/water at ambient temperature, [5.59] for steam/water in a heated tube and steam/water in an unheated tube [5.92] are shown in Fig. 5.6. There is no change in the void fraction for the air/water two-phase flow downstream the tube. On the contrary, in steam/water system, the void fraction changes along the tubes, due to change to pressure drop. This is more pronounced in a heated tube as the effect of evaporation due to heating is much more important than that of pressure drop. Thus it can be said that steam/water flows cannot be modelled in the same way as air/water flows. In particular when modelling steam/water flows, evaporation must be taken into consideration.

![Figure 5.6 Comparison of different models developed for air/water and steam/water in heated and unheated tubes.](image)

### 5.4 Summary and Conclusions

In this chapter new explicit expressions have been derived for void fraction and pressure in evaporating bubbly flows in a heated tube. The solution is given by equation [5.59]. The void fraction in an unheated tube is also described briefly in Section 5.2. Under these conditions for liquid flows of approximately 1 ms⁻¹, the contribution of the acceleration of the liquid to the pressure drop is nearly as important as the contribution of hydrostatic head. The effects of gas acceleration are negligible. The void fraction expression is validated with experimental results and good agreement was obtained.
We may also note that in the equations the void fraction is presented in terms of evaporation and that the change in void fraction due to a change in pressure is compounded by expansion of the evaporated liquid. Evaporation is far more important than expansion in steam/water systems. The evaporation of the continuous phase (liquid) is found, as expected, to be substantially due to the heat load compared to the change in pressure downstream the flow in a heated tube.
CHAPTER 6

NUMERICAL STUDY
NUMERICAL STUDY

Chapter 5 presented the theoretical models in dealing with prediction of void fraction and pressure drop in the heated and unheated vertical pipes. This chapter deals with the calculation of void fraction in horizontal pipes in either of a straight pipe or a venturi. In the work described here, studies have been made using one of the CFD (Computational Fluid Dynamics) codes namely called CFX4.1. The work was initiated by Dr. Y. Yan as part of his research fellowship at Surrey University for vertical pipes and the further development reported here was a study of two-phase flow in horizontal pipes.

6.1 Previous Works

Much work has been done on the numerical modelling of two-phase bubbly flows. One-dimensional methods have been widely used for design calculations. Probably the homogenous equilibrium mixture model is the simplest among the models. Such models have been upgraded by prescribing velocity and void distribution profiles, and the relative velocity between the phases. Strictly, these require advance knowledge of the radial distributions, which information is not normally available. Therefore we consider multidimensional calculations with an appropriate two-fluid model. The conservation equations of such a model for two-phase flow have been rigorously developed by Ishii (1975) and updated by Lahey and Drew (1990). In the resultant model, both the Reynolds stresses and some new interfacial transfer forces were contained. Similar work with two-fluid models has also been carried out by Gaard and Brekke (1992), Simonin and Viollet (1989) and Lopez de Bertodano et al. (1994).

Two important issues arise in the modelling of two-phase flows. One is the effect of bubbles on the turbulent field and the other is the nature and magnitude of the interfacial forces. The turbulent field in the liquid can clearly affect the distribution of bubbles, where the bubbles in turn affect the liquid phase turbulence. The earlier work of Drew and Lahey (1982, 1987), in which the mixing length theory was applied to analyse phase distribution of bubbly flow, suggested that turbulence could be a dominant mechanism in lateral phase distribution. Since then the k-ε model has been used in turbulent two-phase flow (Lee et al., 1989, Simonin and Viollet, 1989). As far as interfacial forces are
concerned, the drag force on bubbles has been extensively investigated and many correlations for the drag coefficient of bubbles reported. Other forces for interfacial momentum transfer, such as the added mass force and the lift force, may be derived from the first principles of inviscid flow. Drew and Lahey (1987) and Clift et al. (1978) suggested that the added mass force should be significant for low concentrations of dispersed bubbles in continuous liquid. However, Gaard and Brekke (1992) showed that the added mass force is likely to be important only in flow situations where there are large differences between the liquid and gas acceleration. Their numerical results have shown that there are no significant differences between computations with and without added mass force terms.

It is noted that the constitutive relations published for phase distribution in bubbly two-phase flow are normally only concerned with flow without phase changes. In particular, interphase mass transfer has usually been ignored. This may be appropriate for an air/water system, which is at room temperature, but may not be for other vapour/liquid bubbly flows in which evaporation or condensation may occur. In vapour/liquid bubbly flows, like steam/water systems, interfacial heat and mass transfer are likely to be very significant. The effect of phase changes as a result of heat and mass transfer is shown to be important in chapter 4, where the flow passes through a venturi (Fig. 4.35).

6.2 Mathematical Model

6.2.1 Governing Equations

A three-dimensional two-fluid model has been developed for predicting gas/liquid and vapour/liquid turbulent two-phase horizontal bubbly flows. The two-phase approximation used in the present study assumes that each phase co-exists and flows in a horizontal tube and that each phase may have its own unique flow and temperature field. Conservation equations for mass, momentum and energy are solved for each phase to determine the respective flow fields. Interphase transport terms allow for the exchange of momentum and energy between the phases. The equations are as follows:

Continuity equations:

\[
\frac{\partial (\varepsilon_i \rho_i)}{\partial t} + \nabla \cdot (\varepsilon_i \rho_i \mathbf{u}_i) = m_{ij} - m_{ji} \quad (i = L, G, \text{ and } j \neq i)
\]  

[6.1]
Chapter 6

Momentum equations:

\[
\frac{\partial(\varepsilon_i \rho_i u_i)}{\partial t} = \varepsilon_i (\rho_i g - \nabla p) + \nabla \cdot (\varepsilon_i \mu_{\text{eff}} (\nabla u_i + (\nabla u_i)^T)) - \nabla \cdot (\varepsilon_i \rho_i u_i u_i) + M_i + (m_{ij} u_j - m_{ji} u_i) \tag{6.2}
\]

Energy equations:

\[
\frac{\partial(\varepsilon_i \rho_i H_i)}{\partial t} + \nabla \cdot (\varepsilon_i \rho_i u_i H_i) = \nabla \left( \frac{\varepsilon_i \lambda_i}{c_p} \nabla H_i \right) + \beta_i (T_j - T_i) + (m_{ij} H_j - m_{ji} H_i) \tag{6.3}
\]

In the above equations, the subscripts \(i\) indicates the phase \((i = L \text{ for liquid phase and } j = G \text{ for gas phase})\), \(t\) indicates time and \(u\) the velocity. The other symbols \(\rho\) represents the density, \(p\) the pressure, \(T\) the temperature, \(H\) the enthalpy, \(c_p\) the specific heat, \(\lambda\) the thermal conductivity, \(\mu_{\text{eff}}\) the effective viscosity, \(m_j\) is the mass flow rate per unit volume into phase \(i\) from phase \(j\), \(\beta\) the interface heat transfer term and \(M_i\) represents the interphase momentum transfer forces, \(\varepsilon_i (i = L \text{ and } G)\) represents the void fraction of liquid phase and gas phase, respectively, and we know that:

\[
\varepsilon_L = 1 - \varepsilon_G \tag{6.4}
\]

The two phases are supposed to share the same pressure field. Due to the dominance of the frictional resistance of the liquid flow in the pressure drop for a horizontal pipe, the pressure term in the momentum equation can be expressed as:

\[
p_L = p_G = \frac{2}{d} f_w \rho_L u_L^2, \quad (f_w = 0.079 \text{Re}^{0.25}_L = 0.079 \left( \frac{\rho_L \varepsilon_L u_L D_h}{\mu_L} \right)^{0.25}) \tag{6.5}
\]

where \(D_h\) is the diameter of the pipe and \(f_w\) is the wall friction factor of which the Blasius relation (Mercadier, 1981) is used.

6.2.2 Interfacial Transfer Terms

Mass Transfer

In the continuity equations, \(m_{ij} - m_{ji}\) is the net change in mass flow rate associated with phase change. For dispersed air/water flow at room temperature, the air bubbles may be considered relatively insoluble, thus, a process with isothermal mixing and no phase change may be assumed and hence \(m_{ij} - m_{ji} = 0\) applied. Nevertheless mass transfer in a steam/water bubbly flow cannot be ignored. This can be described in terms of a mass transfer coefficient, \(C_M\), as:
\[ m_{ij} - m_{ji} = -C_M \varepsilon_j \rho_i \quad (C_M > 0), \text{ for evaporation,} \]  
\[ m_{ij} - m_{ji} = C_M \varepsilon_j \rho_i \quad (C_M < 0), \text{ for condensation} \]  

**Interphase forces**

The interphase momentum transfer forces, \( M_i (i=L,G) \), consist of the drag term and the other forces, such as lift and added mass forces. The forces for different phases can normally satisfy the relation of \( M_G + M_L = 0 \) \[6.7\]

In the present study, only the drag forces are considered. This is partly because it has been reported that the effects of other forces on bubbly flow are negligible (Gaard and Brekke, 1992). Furthermore, the current model uses the CFX4.1 code, which can highlight the differences in the interphase heat and mass transfer for the two different bubbly flows. Thus, the forces is represented as:

\[ M_L = M_G = \frac{3}{4} C_D \frac{\rho_G}{d_b} \varepsilon_G |u_G - u_L|(u_G - u_L) \]  
\[6.8\]

In this equation, \( d_b \) is the bubble diameter, and \( C_D \) is the non-dimensional drag coefficient of the bubbles. For \( 1000 \leq \text{Re} \leq 1 \times 10^5 \), \( C_D=0.44 \) was used.

**Heat Transfer**

Heat transfer across a phase boundary is described in terms of an energy coefficient, \( \beta \), which is the amount of heat transferred per unit time per unit temperature difference. In the model, we assume the gas side temperature is uniform and equal to the interfacial temperature, so the energy coefficient, \( \beta \), is actually proportional to the external heat transfer coefficient \( h \). Thus, assuming a dispersion of uniform spherical bubbles, the rate of heat transfer \( Q \) per unit time across a interphase boundary of area \( A \), from phase \( i \) to phase \( j \), is:

\[ Q = \beta_i (T_j - T_i) = h_{ij} A (T_j - T_i) = \frac{6 \varepsilon_G}{d_b^2} \lambda_L \text{Nu} (T_j - T_i) \]  
\[6.9\]

The Nusselt number \( \text{Nu} \) has been given by Hughmark (1967) as:

\[ \text{Nu} = 2 + 0.27 \text{Re}^{0.62} \text{Pr}^{0.33} \quad (450 \leq \text{Re}) \text{ and } (0 \leq \text{Pr} \leq 250) \]  
\[6.10\]
6.2.3 The Turbulent Model

To model the turbulence in the continuous liquid phase, a low Reynolds number $k$-$\varepsilon$ model, which is well developed for single-phase flow (CFX manual, 1995), was employed. The molecular and turbulent diffusion of momentum is governed by an effective viscosity:

$$
\mu_{\text{eff}} = \mu_L + \mu_{\text{TL}} \quad \text{where} \quad \mu_{\text{TL}} = C_\mu \alpha_k \rho_L \frac{k_L^2}{\varepsilon_L} \quad \text{[6.11]}
$$

The transport equations for the turbulence kinetic energy $k_L$ and turbulence dissipation rate $\varepsilon_{DL}$ are:

$$
\frac{\partial (\rho_L k_L)}{\partial t} + \nabla \cdot (\rho_L \mathbf{u}_L k_L) - \nabla \left( \left( \mu_L + \frac{\mu_{\text{TL}}}{\sigma_k} \right) \nabla k_L \right) = P + G - \rho_L \varepsilon_{DL} - D \quad \text{[6.12]}
$$

and

$$
\frac{\partial (\rho_L \varepsilon_{DL})}{\partial t} + \nabla \cdot (\rho_L \mathbf{u}_L \varepsilon_{DL}) - \nabla \left( \left( \mu_L + \frac{\mu_{\text{TL}}}{\sigma_k} \right) \nabla \varepsilon_{DL} \right) = C_1 \varepsilon_{DL} \left( P + C_1 \max(G,0) \right) \quad \text{[6.13]}
$$

Where in these equations $P$ is the shear production and $G$ is the production due to body force. The details of the definition equation of $P$ and $G$ and another functions, such as $f_p$, $f_2$, $D$ and $E$, have been elaborated in detail by Launder and Sharma (1974). The constants in Eqs. 6.11-6.13 are specified as in the standard single-phase flow $k$-$\varepsilon$ model, namely, $C_1=1.44$, $C_2=1.92$, $C_3=0.0$ and, $C_\mu=0.09$, respectively. The turbulent Prandtl number for $k$, $\sigma_k=1.0$ and the turbulent Prandtl number for $\varepsilon$, $\sigma_\varepsilon=K^2/((C_2-C_1)(C_\mu)^{0.5})=1.27$, where the Von Karman constant, $K=0.419$.

6.3 Numerical Method

The equation system of the two-fluid model was discretised and solved on the controlled grid using the finite volume program, CFX-F3D version 4.1. Body-fitted co-ordinates are included to allow the treatment of arbitrary three-dimensional geometry. The effect of the number of grids and their sizes on the convergence of the solution has been considered prior to the runs at different phase velocities. Moreover, the maximum size of grid has always been selected in order to minimise the time of each run. Interphase linking is
solved in CFX4.1 using the Inter Phase Slip Algorithm (IPSA) of Spalding (1976). This solves the coupled equations in a segregated mode, with the option of accelerating convergence using the Partial Elimination Algorithm (PEA) of Spalding, or the SINCE method of Lo (1989). The pressure drop term is programmed into CFX4.1 via the user FORTRAN interface.

In the present study, steady flows were considered using DEC 3000 Alpha computer and more recently a Sun computer namely called BORG. The solutions were obtained by starting from some initial flow and temperature distributions set as boundary conditions and iterating towards steady flow. The conditions at outlet were set as a pressure boundary, at which the pressure was kept as atmospheric pressure. The initial distributions of velocities and temperatures for the two phases and the void fraction were given at the inlet. The air and steam vapour bubbles at inlet were all treated as being 3 mm diameter in spherical shape.

6.4 Results and Discussion

6.4.1 Air/Water and Steam/Water Flows in Equal Diameter Pipes

Bubbly air/water and steam/water flows in a horizontal pipe were investigated using the numerical model described above. Computations were carried out for horizontal pipe of 15 m length and 50 mm inside diameter assuming a uniform distribution for the initial mean velocity and temperature profile of the continuous phase and a uniform void fraction distribution. The computation was firstly carried out for the above conditions in order to validate the numerical results with the experimental results of other investigators. The migration of the bubbles between different lateral sections of the pipe have also been obtained.

Fig 6.1 shows the numerical results of local void fraction distribution for air/water bubbly flow at different lateral sections. In this figure r is the distance from the centre of the tube in vertical direction. It is can be seen from this figure that, buoyancy causes the migration of gas bubbles toward the top of the horizontal pipe and that consequently the flow becomes highly non-symmetric over the pipe cross-section. Another characteristic is that the local void fractions at L/D>20 have similar distributions. Generally, the published experimental data for air/water horizontal bubbly flow is very limited. Some of results
were reported by Kocamustafaogullari and Wang (1991), in which the bubbly flow in a 15m long horizontal pipe for fully developed flow was carried out (with superficial liquid and gas velocities of 4.9 and 1.34 m/s, respectively). We had to use other experimental results because we have not able to measure the void fraction at different cross-sections. The numerical results for local void fraction at \(L/D=10\) are reasonably consistent with the experimental results of Kocamustafaogullari and Wang (1991) as shown in Fig. 6.2, which were measured by means of quick closing valves. Therefore the effect of error caused by using the quick closing valves may have been included in the experimental results.

![Figure 6.1](image)

**Figure 6.1** Variation of void fraction of air/water bubbly flow at different lateral sections.

Similar void fraction profiles were also obtained for steam/water bubbly flows by numerical calculation. Fig. 6.3 shows the comparison of calculated void fraction distribution between air/water and steam/water bubbly flows. All these calculations are at similar input velocities of dispersed phase (gas and steam) and water as well as same initial bubble size and void fraction distribution. At the same lateral section, the void fraction of steam/water bubbly flow is higher than that of air/water bubbly flow. These differences arise from the different pressure and temperature distribution in the two systems reflecting the different mechanisms of interphase mass and heat transfer.
Results and Discussion

1.00

Figure 6.2 Comparison between numerical and experimental results.

Figure 6.3 Void fraction distribution for air/water and steam/water bubbly flows.
6.4.2 Flow in Venturi Pipes

As mentioned in chapter 4, venturis and orifices have often been used to measure flow rate in single and two-phase flows. Other applications of two-phase bubbly flow through restrictions are important in industrial processes. Two-phase bubbly flows through a horizontal venturi have been computed using the numerical model for the venturi, described in chapter 3. Fig. 6.4 compares the results for void fraction distribution at lateral sections and along the flow direction. It can be seen that air/water bubbly flow in the venturi has a quite different void fraction distribution from steam/water flow. The void fraction profile of air/water flow is relatively uniform from the throat area to the outlet, but it is not uniform for the steam/water flow because of mass and heat transfer between the phases. At the outlet section, the void fraction of steam/water flow is higher than air/water flow in the central area of the pipe and the value become close at the top area of the pipe. While at the throat section, the void fraction of steam/water flow is slightly lower than the air/water flow. This is because that, the large increase of the velocity in this area results in a larger change of the steam bubble temperature (T=372-366 K), so that condensation may become effective. However, for the air/water flow, the temperature change in this area was very small and no significant phase changes occur. A phenomenological comparison between Fig 6.4 and Fig 4.35 shows that the numerical results are consistent with the experimental results. However the numerical results do not show a significant change in void fraction at the outlet as exhibited in Fig 4.35. This implies that effect of phase change in the solution procedure (in the CFX code) has to be modified. This includes the modification of constant mass transfer and empirical correlation of heat transfer between the phases.

It should be pointed out that the aim of the present chapter was to validate the applicability of the numerical methods as well as detail calculation of parameters such as void fraction at different cross-sections, which we have not been able to determine in chapter 5. Despite the fact the numerical results were consistent with some the experimental results, however the accuracy of the numerical results deteriorates at low flow rates (<25 lit/min) due to divergence of numerical results at such flow rates. The new versions of the CFX code claim that they overcome the difficulty of interaction between the phases at low phase flow rates.
Figure 6.4 Comparison of void fraction distributions for air/water and steam/water bubbly flows through a venturi at the throat and outlet points.
CHAPTER 7

CONCLUSIONS AND FUTURE WORK
CONCLUSIONS AND FUTURE WORK

7.1 Conclusions

The aim of the present work was to study the differences between gas/liquid and vapour/liquid systems, which involves the experimental, theoretical and numerical studies. The literature survey reveals that although many studies appear to confirm similarities in the behaviour of air/water and steam/water mixtures, others report distinct differences between the two systems. The parameters involved in the phenomena include the differences in the physical properties of the mixtures, phase changes and rapid bubble coalescence or collapse in steam/water flow and in air/water flow near to its boiling point.

In order to establish the differences between the two systems a new test rig has been designed and constructed in both vertical and horizontal positions for either of a straight pipe or a venturi. A comprehensive set of experiments to find out the effect of liquid and gas/steam flow rates and temperature on the void fraction in both a normal pipe and through a venturi has been carried out. The main conclusions from the experiments are as follows.

- The effect of bubble size should be taken into account in correlations for the void fraction in the bubbly flow regime. An influence can be expected as a result of the change in bubble rise velocity and of wall effects altering the slip velocity between the phases.

- In dealing with differences between air/water and steam/water flows, the effect on the void fraction of increasing vapour pressure at elevated temperatures cannot be ignored. Without correcting for this, the experimental air/water data would show an excessive influence of temperature especially above 60 °C. This effect becomes greater at either higher liquid flow rates or lower inlet gas flow rates. The modified volumetric flow ratio (equation [4.4]) adequately includes the effect of vapour pressure and hydrostatic pressure on the void fraction at higher flow rates. The
results also reflect a change in flow regime as temperature rises due to a sharp increase of bubble size with temperature.

- The recorded images show that there is less bubble clustering at higher temperatures. This is probably due to a weak boundary layer of the partially saturated air bubbles leading to rapid bubble coalescence (Figs. 4.12 and 4.13). Experimental evidence has shown that the change in flow regime is caused by bubble coalescence at lower temperature, however clustering would be affected by growth and subsequent coalescence as the air bubbles saturate at elevated temperatures. This phenomenon is more significant at greater downstream distances.

- The results of the Laser Doppler Velocitometer measurement show that the local liquid velocity increases slightly near the centre of the pipe as gas flow rate increases at ambient temperature. The experiments also exhibit an increase in local liquid velocity, as temperature increases, particularly to the centre of the pipe at constant void fraction.

- Experiments in the steam/water system were found to be less reproducible, both as a result of rapid bubble coalescence or growth and also instability caused by rapid phase changes. Not unexpectedly the data show an increase of void fraction with the local heat load for both vertical and horizontal flows. Rapid bubble coalescence and collapse was observed in steam/water flow due to the high rate of heat and mass transfer between the phases. It was found to be more reliable and convenient always to decrease local heat load for measurement of void fraction in order to keep the flow at its saturation temperature.

- The local void fraction changes remarkably as an air/water flow passes through a venturi with the greatest change at the venturi throat. This shows a maximum or minimum peak at the throat as the liquid flow rate changes from stagnant to upflow. It has also been confirmed that the effect of increasing the liquid flow rate is much more important than changes in gas flow rate. The effect of temperature was found to be exactly the same as in a normal tube, but the minimum at the throat was deeper.
at higher temperatures, particularly near to the boiling point as a result of the contribution of the higher vapour pressure.

Perhaps the most distinctive behaviour of air/water and steam/water was observed at the throat of a venturi and beyond, where there is an abrupt increase of the void fraction as a result of depressurisation and subsequent flow flashing in steam/water flow (Fig. 4.35). The magnitude of this depends on the degree of superheat in the liquid phase.

Chapter 5 presented a theoretical model for the prediction of void fraction and pressure drop along a heated tube. The one-dimensional model is called simplified as it is based on neglecting the small terms compared to the others in the governing equations. The solution is given by equation [5.59]. In this equation the void fraction changes due to the change in local pressure and temperature. The theoretical results showed that in the prediction of local void fraction at saturation conditions the most important parameters are the heat load (Fig. 5.3) and local pressure. The changes in liquid and gas densities are also found to be negligible.

The results of the previous theoretical work done by Lisseter (1999) for the prediction of void fraction and pressure drop in the unheated tubes are also compared with the experimental results of chapter 4 and good accuracy was obtained. The accuracy of the model deteriorates at lower liquid flow rates. The theoretical results show that the degree of superheat is important in the prediction of the point at which the liquid phase turns into the vapour phase.

Finally, comparison between equations [5.92] for the prediction of the void fraction in the unheated tubes (steam/water), [5.59] for the heated tubes (steam/water) and [5.93] for air/water (ambient temperature) show a distinctive behaviour, mainly as a result of phase change in steam/water system. The equations may give a guide for the design of the straight tubes, as they are explicit and simple to use.

A numerical study has also been presented in chapter 6, in which only the results for horizontal pipe flow have been presented. The numerical studies was thought to be
important as the theoretical solutions given in chapter 5 were only for one direction and also because more information of local parameters such as the void fraction distribution at different cross-sections along the pipe were of interest. The numerical results exhibit the difference in local void fraction at different cross-sections between air/water and steam/water flows. This distinctive behaviour depends on the different temperature and pressure distributions, reflecting the different mechanism of interphase mass and heat transfer (Fig. 6.4).

7.2 Future Work

Some general remarks and suggestions for future work are given in the following:

It has been shown that void fraction effectively changes as a function of temperature with respect to variation of inlet volumetric flow ratio. Including the effect of vapour pressure in the calculation of volumetric flow ratio is generally sufficient to account for the effect of temperature. Nevertheless more study has to be done in order to extend the data for lower gas and higher liquid velocities.

Video recorded images showed the significant effect of temperature on the development of the flow structure in the straight pipes. The same experiment can be done for the venturi described in chapter 3. It is probable that information on the way that bubble shape changes during flow through the venturi will give the idea of the way the phases interact.

The dynamics of a single bubble plays a key role in the nature of air/water and steam/water flows. The results shown in chapter 4 present the distinctive behaviour of the air and steam bubbles. However more experiments need to be done. One possible way of doing this to study the bubble response as it passes through a venturi in air/water, particularly near to the boiling point. This influence should be established by including the effect of bubble size, shape and bubble velocity on the dynamics of the two-phase flow as characterised by the pressure field determinations mentioned above. The air bubbles have to be as small as possible, so the change in their shape can be recorded at the throat.
It has been emphasised that the behaviour of bubbles has a great influence on the structure of two-phase flow and interphase transport phenomena between the phases, including mass, heat and momentum. It is experimentally found that the bubble size changes with temperature at fixed flow rates. Moreover a change in bubble size has a significant effect on the slip velocity, as it depends on the bubble rise velocity ($d_0 < 3 \text{ mm}$). An investigation using image analysis is suggested in order to gain a better insight into the structure of two-phase flow at constant void fraction in order to confirm the results in Fig. 4.16.

Preliminary calculations show that when two-phase flow passes through a venturi, slip velocity changes, particularly at the throat. This leads to a considerable change in void fraction at the throat. LDV (Laser Doppler Velocimeter) measurements can be carried out to determine how the local liquid velocity changes at specified locations. The slip velocity between the phases can simply be determined from the local liquid velocity obtained from LDV and bubble velocity obtained from image analysis at the same position of the pipe.

The rate of mass transfer in a steam/water mixture is vastly higher than air/water flow. Accordingly, void fraction data in the axial direction will provide an excellent phenomenological comparison between the two systems. This will be particularly informative when it is carried out for both horizontal and vertical flows.

The theoretical results in chapter 5 show the reliability of the one-dimensional simplified model. The similar model can be developed for the constrictions such as venturis.

As mentioned in chapter 6, the current version of CFX (4.1) was unable to cope with simulation of the two-phase flow at low flow rates ($< 25 \text{ lit/min}$). Another difficulty arises from the fact that the mass transfer coefficient between the phases was assumed to be constant. Therefore, the new code of CFX which it is claimed that has been developed to deal with such difficulties has to be used. Moreover, computational fluid dynamic modelling should be carried out for movement of either of single air or steam bubble through a venturi in the unsteady-state condition. This
will provide crucial validation for the numerical procedures being developed for the flow of single bubble through a venturi and assesses the validity of the many assumptions about rates of heat and mass transfer that must be made in attempting such computations for a single bubble and liquid systems. To match the extensive data that will probably be available from the experimental programme which is suggested above, the numerical models must include the effects of various flow rates, fluid properties and of bubble size.
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APPENDIX A

GAMMA DENSITOMETER
APPENDIX B

FLOW STRUCTURE OF AIR/WATER AT AMBIENT AND ELEVATED TEMPERATURE
L/D=17, T=17 °C

$L_Q$ (lit/min)

0.0

6.0

20.8

35

$Q_g$ (lit/min)
L/D=45,  T=17^\circ C

\(Q_g\) (lit/min)

\(Q_l\) (lit/min)

- 6.0
- 20.78
- 35
L/D=17, T= 60°C

\( Q_g \) (lit/min)

<table>
<thead>
<tr>
<th>( Q_1 ) (lit/min)</th>
<th>0.4</th>
<th>0.6</th>
<th>1.0</th>
<th>2.0</th>
<th>4.0</th>
</tr>
</thead>
<tbody>
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<td>6.0</td>
<td></td>
<td></td>
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<td></td>
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<tr>
<td>20.78</td>
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<tr>
<td>35</td>
<td></td>
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</tbody>
</table>
L/D=45, $\ T=60^\circ$C

$Q_g$ (lit/min)
L/D=17,  T= 90 °C

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<th>0.6</th>
<th>1.0</th>
<th>2.0</th>
<th>4.0</th>
</tr>
</thead>
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<tr>
<td>6.0</td>
<td></td>
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<tr>
<td>20.78</td>
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<tr>
<td>35</td>
<td></td>
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</tr>
</tbody>
</table>
L/D=45,  \( T = 90^\circ C \)

\[ Q_1 \text{ (lit/min)} \]

\begin{align*}
0.4 & \quad 0.6 & \quad 1.0 & \quad 2.0 & \quad 4.0 \\
6.0 & & & & \\
20.78 & & & & \\
35 & & & & 
\end{align*}
L/D=17, T=95°C

Q_g (lit/min)

<table>
<thead>
<tr>
<th>Q_1</th>
<th>0.4</th>
<th>0.6</th>
<th>1.0</th>
<th>2.0</th>
<th>4.0</th>
</tr>
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<tbody>
<tr>
<td>6.0</td>
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<tr>
<td>20.78</td>
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<tr>
<td>35</td>
<td></td>
<td></td>
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<td></td>
</tr>
</tbody>
</table>
L/D=45, \quad T=95^\circ C

\begin{align*}
\text{Q}_g \text{(lit/min)} \\
\begin{array}{c}
0.4 \\
0.6 \\
1.0 \\
2.0 \\
\end{array}
\end{align*}

\begin{align*}
\text{Q}_1 \\
\quad \text{(lit/min)} \\
6.0 \\
20.78 \\
35
\end{align*}
L/D=45

T (°C) = 17  60  90  95
Q_g = 0.4 lit/min  Q_t = 6 lit/min

L/D=17

T (°C) = 17  60  90  95
Q_g = 1.0 lit/min  Q_t = 6 lit/min
L/D=45

\[ T \, (^{\circ}C) = 17 \quad 60 \quad 90 \quad 95 \]

\[ Q_g = 0.4 \, \text{lit/min} \quad Q_l = 20.78 \, \text{lit/min} \]

L/D=17

\[ T \, (^{\circ}C) = 17 \quad 60 \quad 90 \quad 95 \]

\[ Q_g = 0.4 \, \text{lit/min} \quad Q_l = 20.78 \, \text{lit/min} \]
L/D=45

T (°C) = 17 60 90 95

Q_g = 1.0 lit/min  Q_l = 35 lit/min

L/D=17

T (°C) = 17 60 90 95

Q_g = 2.0 lit/min  Q_l = 35 lit/min
L/D=45

Q_l = 35 lit/min
Q_g = 2.0 lit/min
T = 17 °C

L/D=17

Q_l = 35 lit/min
Q_g = 4.0 lit/min
T = 17 °C
$Q_I = 35 \text{ lit/min}$  \hspace{1cm} 20.8  \hspace{1cm} 6.0

$Q_g = 2.0 \text{ lit/min}$  \hspace{1cm} $T = 60 \, ^\circ C$

$L/D = 45$

$L/D = 17$

$Q_I = 35 \text{ lit/min}$  \hspace{1cm} 20.8  \hspace{1cm} 6.0

$Q_g = 2.0 \text{ lit/min}$  \hspace{1cm} $T = 90 \, ^\circ C$
L/D=45

Q_g=0.6(lit/min) 1.0 2.0 4.0

T=17°C, Q_t=6.0 lit/min

L/D=17

Q_g=0.6(lit/min) 1.0 2.0 4.0

T=17°C, Q_t=6.0 lit/min
L/D=45

$Q_g=0.6\, \text{(lit/min)}$  1.0  2.0  4.0

$T=60\, ^\circ\text{C}, \, Q_l=20.8 \, \text{lit/min}$

L/D=17

$Q_g=0.6\, \text{(lit/min)}$  1.0  2.0  4.0

$T=60\, ^\circ\text{C}, \, Q_l=20.8 \, \text{lit/min}$
L/D=45

Q_p=0.6(lit/min)  1.0  2.0  4.0

T=90 °C, Q_l=35.0 lit/min

L/D=17

Q_p=0.6(lit/min)  1.0  2.0  4.0

T=90 °C, Q_l=35.0 lit/min
APPENDIX C

FLOW STRUCTURE OF AIR/WATER AT ELEVATED TEMPERATURES AND CONSTANT VOID FRACTION
\begin{align*}
T &= 15 \, ^\circ C \\
95 \, ^\circ C \\
\text{Gas flow rate} &= 1.5 \text{ lit/min} \\
\alpha &= 0.06 \\
2.8 \text{ lit/min} \\
\alpha &= 0.105 \\
4 \text{ lit/min} \\
\alpha &= 0.145 \\
\text{Liquid flow rate} &= 21.8 \text{ lit/min}
\end{align*}

\begin{align*}
T &= 13 \, ^\circ C \\
95 \, ^\circ C \\
\text{Gas flow rate} &= 1.5 \text{ lit/min} \\
\alpha &= 0.04 \\
2.8 \text{ lit/min} \\
\alpha &= 0.07 \\
4 \text{ lit/min} \\
\alpha &= 0.100 \\
\text{Liquid flow rate} &= 35 \text{ lit/min}
\end{align*}
APPENDIX D

THEORETICAL STUDY’S BACKGROUND
Changes in Void Fraction due to Changes in Pressure in Vertical Saturated Vapour/Liquid Flows

This section presents a set of equations for prediction of void fraction along an unheated tube for saturated vapour/liquid flow. The complete derivation was presented by Dr. Pamela Lisseter (1999). The results of this section have already been discussed and compared with the experimental results in chapter 5. However the presentation of the complete model is thought to be indispensable here providing the background to the model posed in chapter 5.

The derivation of void fraction, in this section, along an unheated tube is based on the set of equations presented in chapter 5 from [5.1] to [5.20]. As far as the definition of problem is concerned, there are two differences between that described in the following and in chapter 5. These involve ignoring the generation of steam bubbles due to heat load and that changes in the vapour density are allowed, though as mentioned before these changes are negligible.

The solution procedure starts in section 1 of this appendix, when eight of the equations (Eqs. [5.2] to [5.20]) will be used to derive an expression for the local void fraction in terms of the local pressure. Equation [5.1] expresses the local pressure in terms of the void fraction profile downstream. Section 2 considers the pressure expression for vertical flow and iterates between it and the expression for the local void fraction in terms of the pressure to obtain explicit expressions for the void fraction and pressure in terms of the distance from the end of the tube.

1. Deriving an expression for the local void fraction in terms of the local pressure

A schematic diagram of the model is illustrated in Fig. 1. The boundary of the model starts from the top of the tube, similar to the assumptions used in chapter 5. In the following section, the variables that arise in the equations [5.2] to [5.20] are considered and estimated to derive an expression for the local void fraction in terms of the local pressure. This may be started by eliminating $\Gamma$ between [5.17] and [5.18]:

$$(\rho_G \varepsilon_G u_G A + \rho_L \varepsilon_L u_L A)_{x} = 0 \quad [1]$$
It is then integrated and boundary conditions at some point along the tube are applied, where variables have values indicated by the subscripts ‘O’. Equations [5.20] and [5.2] are used and the variable ‘A’ is cancelled throughout, since it is non-zero:

\[(\rho_G \varepsilon_G + \rho_L (1 - \varepsilon_G))u_L = (\rho_{GO} \varepsilon_{GO} + \rho_{LO} (1 - \varepsilon_{GO}))u_{LO},\]  

\[
\frac{u_L}{u_{LO}} = \frac{(\rho_{GO} \varepsilon_{GO} + \rho_{LO} (1 - \varepsilon_{GO}))}{(\rho_G \varepsilon_G + \rho_L (1 - \varepsilon_G))}.
\]

This equation will be considered later. For the moment, \(\Gamma\) can again be eliminated, this time between [5.18] and [5.19], therefore:

\[
\frac{1}{\rho_L u_L \varepsilon_L} (\rho_L u_L \varepsilon_L)_x = \left(\frac{C_v}{U_{LG}}\right) T_x.
\]

It should be pointed out that the second term in right hand side of Eq. [5.19] is neglected, as there is no contribution to the generation of vapour bubbles from an additional heat load. It is also assumed that \(U_{LG}\) and \(C_v\) are independent of pressure and temperature. This is reasonable as they only vary slightly over the range of values of pressure and temperature under consideration. The solution to [4], after applying the boundary condition at the end of the tube is:

\[
\rho_L u_L \varepsilon_L = \rho_{LO} u_{LO} \varepsilon_{LO} e^{(C_v/U_{LG})(T-T_0)}.
\]  

Applying [5.4] into [5.2] leads to:

\[
\frac{u_L}{u_{LO}} = \frac{\rho_{LO} (1 - \varepsilon_{GO})}{\rho_L (1 - \varepsilon_G)} e^{(C_v/U_{LG})(m_p \bar{p}_0)}
\]  

[6a]
where

$$\tilde{p}_o = \frac{p_p - p_o}{p_a} \quad [6b]$$

Now, equations [3] and [6] will be equated, and then the equation re-arranged to get the following expression for $F-G$:

$$\varepsilon_G = \frac{\left( \frac{p_{GO}}{p_{LO}} \right) \varepsilon_{GO} + (1 - \varepsilon_{GO})(1 - e^{\delta_2 \tilde{p}_o})}{\left( \frac{p_{GO}}{p_{LO}} \right) \varepsilon_{GO} + (1 - \varepsilon_{GO})(1 - e^{\delta_2 \tilde{p}_o}(1 - \frac{\rho_G}{\rho_L}))} \quad [7]$$

in which

$$\delta_2 = \left( \frac{C_v}{U_{LG}} \right) (m_1 p_a) = 0.0523 \quad [8]$$

Now, in vertical flow, for small void fractions and over 1 m length, it can be determined that $\tilde{p}_o$ was of order of $10^{-1}$. It is smaller over shorter lengths, as would be the region of bubbly flow in a boiling system (a pipe of diameter 0.0254 m is assumed). It is already seen that in [8] that $\delta_2$ is of order of $10^{-2}$ so $\delta_2 \tilde{p}_o$ is at most of order $10^{-3}$. Thus the term $\exp(\delta_2 \tilde{p}_o)$ can be expanded as a power series in $\delta_2 \tilde{p}_o$ and we can neglect terms which are second order in $\delta_2 \tilde{p}_o$. Thus:

$$e^{\delta_2 \tilde{p}_o} = 1 + \delta_2 \tilde{p}_o + O((\delta_2 \tilde{p}_o)^2) \quad [9]$$

$$1 - e^{\delta_2 \tilde{p}_o} = -\delta_2 \tilde{p}_o \quad [10]$$

Equations [5.11] and [5.8] should be recalled:

$$\rho_G = \rho_{Ga}[1 + \gamma \tilde{p}_a] \quad [11]$$

$$\rho_L = \rho_{La}[1 - \delta_1 \tilde{p}_a] \quad [12]$$

where $\gamma=O(1)$ and $\delta_1 = O(10^{-2})$.

By the arguments given above, $\delta_1 \tilde{p}_a$ is at most of order $10^{-3}$, and may therefore be neglected in the following analysis. Thus

$$\rho_L = \text{constant} \quad [13]$$

In [7] there was the term $\rho_{GO}$. From [11] this is:
A new parameter can be defined as:

\[ \frac{1}{R} = \frac{\rho_{GO}}{\rho_{Ga}} = 1 + \gamma \left( \frac{P_O - P_a}{P_a} \right) \]  

then

\[ \rho_G = \rho_{GO} R \left[ \frac{1}{R} + \gamma P_O \right] \]

so now the term \([\rho_G/\rho_L]\) may be expressed as a function of variables with the subscript 'O'. Thus:

\[ \frac{\rho_G}{\rho_L} e^{\delta_2 P_O} = \frac{\rho_{GO}}{\rho_L} R \left( \frac{1}{R} + \gamma P_O \right) \left( 1 + \delta_2 P_O \right) \]

In this expression, \(\delta_2 P_O\) may be neglected because it is much smaller than the other terms and considering the new definition:

\[ \delta_O = \frac{\rho_{GO}}{\rho_{LO}} \]

and substituting [9],[10],[17] and [18] into [7] lead to:

\[ \epsilon_G = \frac{\delta_O \epsilon_{GO} - (1 - \epsilon_{GO}) \delta_2 P_O}{\delta_O \epsilon_{GO} + (1 - \epsilon_{GO}) \left( -\delta_2 P_O + \delta_O R \left( \frac{1}{R} + \gamma P_O \right) \right)} \]

If the numerator and denominator are divided by \(\delta_O\), and define:

\[ D = \delta_2 (\rho_{La}/\rho_{Ga}) = 83.8 = O(10^2) \]

since

\[ \delta_a = \rho_{Ga}/\rho_{La} = 6.24 \times 10^{-4} = O(10^{-3}) \]

then

\[ \delta_2 = D \left( \frac{\rho_{Ga}}{\rho_{La}} \right) \left( \frac{\rho_{Lo}}{\rho_{GO}} \right) = DR \]

and

\[ \epsilon_G = \frac{\epsilon_{GO} - (1 - \epsilon_{GO}) DR P_O}{\epsilon_{GO} + (1 - \epsilon_{GO}) \left( -DR P_O + 1 + \gamma R P_O \right)} \]

\[ = \frac{\epsilon_{GO} - (1 - \epsilon_{GO}) DR P_O}{1 - (1 - \epsilon_{GO}) (D - \gamma) R P_O} \]
Note that because $D$ depends only on variables measured at atmospheric pressure, it is constant for a given system, whereas $R$ represents the density of gas at the point where the pressure and $\varepsilon_{GO}$ are measured, compared with the atmospheric gas density at boiling point. Note also that $\gamma$ is small compared with $D$. Thus the effects of the expansion of the gas are small compared with the effects of evaporation.

Where the pressure is measured, $\tilde{\rho}_O$ is zero, so $\varepsilon_G$ has the value $\varepsilon_{GO}$ as required by the boundary conditions. If we proceeded down the tube, in the opposite direction to the flow, $\tilde{\rho}_O$ increases, and so the second terms in both the numerator and denominator increase. The first term in the numerator is smaller than the first term in the denominator, and $(D-\gamma)$ is smaller than $D$, so the numerator will become zero before the denominator does. At the point at which the numerator becomes zero, evaporation is deemed to have started. This is when:

$$\tilde{\rho}_O = \frac{\varepsilon_{GO}}{D R (1 - \varepsilon_{GO})} \quad [24]$$

Since $D$ is fairly large ($O(10)$), and for bubbly flow to exist, $\varepsilon_{GO}$ is small (0.25 or less in vertical flow, unless there is much turbulence), [24] gives a fairly small value for $\tilde{\rho}_O$. This implies that the length over which bubbly flow pertains in evaporating flow may be quite short.

In order for bubbly flow to exist, the second term in the numerator of Eq. [23] will be less than $\varepsilon_{GO}$, so the second term in the denominator will be less than 1. Thus the denominator may binomially be expanded as:

$$\varepsilon_G = (\varepsilon_{GO} - (1 - \varepsilon_{GO}) DR \tilde{\rho}_O) \left\{ \left(1 + (1 - \varepsilon_{GO}) R (D - \gamma) \tilde{\rho}_O \right)^+ \left(1 - \varepsilon_{GO}\right)^2 R^2 (D - \gamma)^2 \tilde{\rho}_O^2 \right\} + \text{higher order terms} \quad [25]$$

Terms of order $D^3 \tilde{\rho}_O^3$ are neglected, since these will be small and get:

$$\varepsilon_G = \varepsilon_{GO} - R (1 - \varepsilon_{GO}) (D - \varepsilon_{GO} (D - \gamma)) \tilde{\rho}_O - R^2 (1 - \varepsilon_{GO})^2 (D - \gamma) (D - \varepsilon_{GO} (D - \gamma)) \tilde{\rho}_O^2 \quad [26]$$

This will be used in the solution procedure in the forthcoming section.
2. Deriving Expressions for the Local Void Fraction and Local Pressure in Terms of The Depth

The pressure gradient in vertical bubbly flow is \([5.1]\) in which expresses the pressure in terms of the void fraction. It is again assumed, as before that \(\rho_L\) is constant. At this point \([6]\) and \([8]\) need to be recalled:

\[
\varepsilon_L u_L = \varepsilon_{LO} u_{LO} e^{\delta_2 p_o}. \tag{27}
\]

Expanding the exponential term as a power series as before, lead to the fact that it is reasonable to neglect the terms of order \(10^{-3}\), which are the second and subsequent terms in the expansion (see \([9]\)). Thus these is the approximation

\[
(1 - \varepsilon_G) u_L = (1 - \varepsilon_{GO}) u_{LO} = \text{constant} \tag{28}
\]

That is, the mass of gas which produces void fractions up to 0.25 (when the flow regime changes to slug flow) which is negligible when liquid compared with that of the liquid surrounding the gas. Making this assumption in \([5.1]\) gives rise:

\[
\frac{p_o}{s} = \int_0^x (1 - \varepsilon_G) dx + \sigma \left[(1 - \varepsilon_G)^{-1}\right]_0^x \tag{29}
\]

where

\[
s = \frac{\rho L g}{\rho_a} = O(10^{-1}) \tag{30}
\]

\[
\sigma = \frac{\rho_L (\varepsilon_{LO} u_{LO})^2}{\rho_a} \tag{31}
\]

and if

\[
\varepsilon_{LO} u_{LO} = 1 \text{ms}^{-1} \tag{32}
\]

then

\[
\sigma = O(10^{-2}) \tag{33}
\]

and

\[
|D\sigma| < 1. \tag{34}
\]

The second term in \([29]\) can binomially be expanded:

\[
\left[(1 - \varepsilon_G)^{-1}\right]_0^x = \left[1 + \varepsilon_G + \varepsilon_G^2 - (1 - \varepsilon_{GO})^{-1}\right] \tag{35}
\]
Then it should be iterated between \([29]\) and \([26]\) to obtain expressions for \(\tilde{p}_0\) and \(\varepsilon_G\). It starts with the assumption that:

\[
\varepsilon_{G1} = \varepsilon_{GO} = \text{constant} \tag{26}
\]

Subscripts ‘1,2’ etc. indicate the number of the iteration, whilst ‘0’ indicates a boundary condition. Substituting \([36]\) in \([29]\) gives:

\[
\tilde{p}_{oi} = s(1 - \varepsilon_{GO})_x \tag{37}
\]

\[
\varepsilon_{G2} = \varepsilon_{GO} - (1 - \varepsilon_{GO})^2 R(D - \varepsilon_{GO}(D - \gamma))sx - \left( (1 - \varepsilon_{GO})^4 R^2 (D - \gamma)(D - \varepsilon_{GO}(D - \gamma))s^2 x^2 \right) \tag{38}
\]

Terms in \(x^3\) and higher are neglected, therefore:

\[
\tilde{p}_{o2} = sx(1 - \varepsilon_{GO})[1 + \sigma(1 - \varepsilon_{GO})(1 + 2\varepsilon_{GO})(D - \varepsilon_{GO}(D - \gamma))R] + s^2 x^2 (1 - \varepsilon_{GO})^2 \tag{39}
\]

\[
R(D - \varepsilon_{GO}(D - \gamma)[1/2 + \sigma(1 - \varepsilon_{GO})^2 R((D - \gamma)(1 + 2\varepsilon_{GO})(D - \varepsilon_{GO}(D - \gamma))])
\]

\[
\varepsilon_{G3} = \varepsilon_{GO} - (1 - \varepsilon_{GO})^2 R(D - \varepsilon_{GO}(D - \gamma))sx \left[ \begin{array}{c}
1 + \\
\sigma(1 - \varepsilon_{GO})(1 + 2\varepsilon_{GO}) \\
(D - \varepsilon_{GO}(D - \gamma))
\end{array} \right] \tag{40}
\]

\[
- s^2 x^2 (1 - \varepsilon_{GO})^3 R(D - \varepsilon_{GO}(D - \gamma)) \left[ \begin{array}{c}
1/2 R(D - \varepsilon_{GO}(D - \gamma)) + \\
R(D - \gamma)(1 - \varepsilon_{GO}) \\
+ \sigma R^2 (D - \varepsilon_{GO}(D - \gamma))(1 - \varepsilon_{GO})^2 \\
(D - \gamma)(1 + 2\varepsilon_{GO}) - (D - \varepsilon_{GO}(D - \gamma)) \\
+ \sigma^2 R^3 (1 - \varepsilon_{GO})^3 (1 + 2\varepsilon_{GO})^2 (D - \gamma) \\
(D - \varepsilon_{GO}(D - \gamma))^2
\end{array} \right]
\]
\[ p_{03} = s x (1 - \varepsilon_{GO}) \left[ \frac{1 + (1 + 2 \varepsilon_{GO})(1 - \varepsilon_{GO}) R (D - \varepsilon_{GO} (D - \gamma)) \sigma + }{\sigma^2 (1 - \varepsilon_{GO})^2 (1 + 2 \varepsilon_{GO})^2 R^2 (D - \varepsilon_{GO} (D - \gamma))^2} \right] \]

\[ + s^2 x^2 (1 - \varepsilon_{GO})^2 R \left[ \frac{1/2 + }{\sigma (1 - \varepsilon_{GO}) R} \left\{ \frac{3 \varepsilon_{GO} (D -)}{\varepsilon_{GO} (D - \gamma))} + \right\} \right] \]

\[ (D - \varepsilon_{GO} (D - \gamma)) \]

\[ \sigma (1 - \varepsilon_{GO}) R \left\{ \frac{3 \varepsilon_{GO} (D -)}{\varepsilon_{GO} (D - \gamma))} + \right\} \]

\[ 3 (1 + 2 \varepsilon_{GO}) - \]

\[ (D - \varepsilon_{GO} (D - \gamma)) \]

\[ 3 (1 + 2 \varepsilon_{GO}) - \]

\[ (D - \varepsilon_{GO} (D - \gamma)) \]

\[ \left[ \frac{1 + (1 + 2 \varepsilon_{GO})}{(1 - \varepsilon_{GO} (D - \varepsilon_{GO} (D - \gamma))) R \sigma} \right] \]

\[ \sigma^2 (1 - \varepsilon_{GO})^2 (1 + 2 \varepsilon_{GO})^2 \]

\[ R^2 (D - \varepsilon_{GO} (D - \gamma))^2 \]

\[ - s^2 x^2 (1 - \varepsilon_{GO})^3 R (D - \varepsilon_{GO} (D - \gamma)) \]

\[ \left[ \frac{1/2 R (D - \varepsilon_{GO} (D - \gamma)) + }{R (D - \gamma)(1 - \varepsilon_{GO})} \right] \]

\[ + \sigma (1 - \varepsilon_{GO}) R^2 (D - \varepsilon_{GO} (D - \gamma)) \]

\[ 3 (1 - \varepsilon_{GO}) (D - \gamma)(1 + 2 \varepsilon_{GO}) + \]

\[ 3 \varepsilon_{GO} (D - \varepsilon_{GO} (D - \gamma)) \]

\[ + \sigma^2 (1 + 2 \varepsilon_{GO}) \]

\[ (1 - \varepsilon_{GO})^3 R^3 (D - \varepsilon_{GO} (D - \gamma))^2 \]

\[ 3 (D - \gamma)(1 + 2 \varepsilon_{GO}) - \]

\[ (D - \varepsilon_{GO} (D - \gamma)) \]

\[ 3 \]
Further iterations confirm that the expressions for $\varepsilon_0$ and $\overline{p}_O$ have converged to terms in $x^2$ and $\sigma^2$, neglected terms being small for liquid fluxes not greater than approximated by [32]. Thus [43] and [44] are the solutions for $\overline{p}_O$ and $\varepsilon_0$. 

\[ p_{04} = s x (1 - \varepsilon_{GO}) \left[ \frac{1 + (1 + 2\varepsilon_{GO})(1 - \varepsilon_{GO})(D - \varepsilon_{GO}(D - \gamma))}{\sigma^2(1 - \varepsilon_{GO})^2(1 + 2\varepsilon_{GO})^2 R^2(D - \varepsilon_{GO}(D - \gamma))^2} \right] + \frac{1}{2} + \sigma(1 - \varepsilon_{GO})R \left\{ \frac{3\varepsilon_{GO}(D - \varepsilon_{GO}(D - \gamma))}{(D - \varepsilon_{GO}(D - \gamma))} \right\} \]

\[ + s^2 x^2 (1 - \varepsilon_{GO})^2 R(D - \varepsilon_{GO}(D - \gamma)) \left[ \frac{\sigma^2(1 - \varepsilon_{GO})^2}{(1 + 2\varepsilon_{GO})(D - \varepsilon_{GO}(D - \gamma))^2} \right] \]
3. Obtaining the Void Fraction at a Given Pressure and its Distance from the Boiling Boundary

In this section the aim is to derive some results which may be compared with the experimental data. The first stated description of the problem needs to be recalled, in which the fluid entering the start of the tube is at a given temperature, $T_1$, say, and that the pressure is above that which would allow boiling, which is

$$p_1 = \frac{T_1}{m_1} - c_1$$  \[45a\]

where

$$m_1 = 2.568 \times 10^{-4}$$  \[45b\]

and

$$c_1 = 2.88 \times 10^5$$  \[45c\]

Here the subscript ‘1’ is used to indicate values at saturation conditions. $T_1$ is in degrees Celsius rather than Kelvin.

Now the aim is to know what the void fraction is at a given pressure, $p_0$, further down the tube. The subscript ‘0’ refers to values of variables at this pressure. If the void fraction at this $p_0$ is $\varepsilon_{GO}$, then the void fraction is zero when the pressure is $p_1$, and thus when:

$$\tilde{p} = \tilde{p}_1 = \frac{p_1 - p_0}{p_a}$$  \[46\]

At this value of $\tilde{p}$, [5.43] gives that

$$\varepsilon_{GO} = \frac{DR\tilde{p}_1}{1 + DR\tilde{p}_1}$$  \[47\]

Where

$$DR = \frac{\rho_{LO}}{\rho_{GO}} \frac{C_v}{U_{LG}} m_1 p_a$$  \[48\]

and where $\rho_{LO}$ and $\rho_{GO}$ are defined for the pressure $p_0$, under boiling conditions.

The distance of the position where the pressure is $p_0$ from the boiling boundary is then given by the variable $x_1$. The first two terms of the expansion of the pressure are taken in [43] for vertical flow then the approximation can be made as:
\[ \bar{p}_1 = b x_1 + a x_1^2 \]  \[ \text{[49]} \]

where

\[ b = s(1 - \varepsilon_{GO}) \left( 1 + (1 + 2\varepsilon_{GO})(1 - \varepsilon_{GO}) R(D - \varepsilon_{GO}(D - \gamma)) \sigma + \sigma^2 (1 + 2\varepsilon_{GO})^2 (1 - \varepsilon_{GO})^2 R^2 (D - \varepsilon_{GO}(D - \gamma))^2 \right) \[ \text{[50]} \]

\[ a = s^2 (1 - \varepsilon_{GO})^2 R(D - \varepsilon_{GO}(D - \gamma)) + \sigma^2 (1 + 2\varepsilon_{GO})(1 - \varepsilon_{GO})^2 R^2 \left( \frac{3\varepsilon_{GO}(D - \varepsilon_{GO}(D - \gamma))}{1/2 + \sigma(1 - \varepsilon_{GO}) R + (D - \gamma)} \right) \left( \frac{(D - \gamma)(1 - \varepsilon_{GO})}{(1 + 2\varepsilon_{GO})} \right) \left( \frac{(1/2 - 2\varepsilon_{GO})}{(D - \varepsilon_{GO}(D - \gamma))} \right) \]  \[ \text{[51]} \]

\[ s = \frac{\rho_{LO}}{\rho_a}, \quad \gamma = 5.48 \times 10^{-6} \frac{p_a}{\rho_{Ga}}, \quad \sigma = \frac{\rho_{LO}}{p_a}(\varepsilon_{GO} u_{LO})^2 \]  \[ \text{[52]} \]

Then, to find \( x_1 \), [49] has to be solved as:

\[ x_1 = \frac{1}{2a} \left( -b + \sqrt{b^2 + 4a \bar{p}_1} \right) \]  \[ \text{[53]} \]

The above equation gives the point in which the saturated liquid phase turns to the boiling. This model is validated in chapter 5 with the steam/water experimental data obtained in chapter 4.
APPENDIX E

EXPERIMENTAL DATA
# Experimental Data Files Identification

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