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An Experimental Investigation of Two-Phase (Air–water) Flow Regimes in a Horizontal Tube at Near Atmospheric Conditions

by

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B.A.Sc., M. Eng. P.Eng.

A Thesis Submitted to the
Faculty of Graduate Studies and Research
in partial fulfillment of the requirements for
the degree of

Doctor of Philosophy

in
Engineering

The Ottawa-Carleton Institute for
Mechanical and Aerospace Engineering

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**The undersigned hereby recommend to
the Faculty of Graduate Studies and Research
acceptance of the thesis,**

**An Experimental Investigation of Two-Phase (Air-water) Flow Regimes
in a Horizontal Tube at Near Atmospheric Conditions**

submitted by

M.L. Lamari, B.A.Sc., M.Eng.

**in partial fulfilment of
the requirements for the degree of
Doctor of Philosophy**

Chair, Department of Mechanical and Aerospace Engineering

Thesis Supervisor

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**Carleton University
February 17, 2001**

Abstract

A series of air–water two-phase flow experiments was performed in a 25.4-mm horizontal tube at near atmospheric conditions. Measurements of the frictional pressure gradients, local void fractions, and gas–liquid interfacial area were taken. A twin-needle optical probe was used to measure the local void fractions at an axial location (l/d) of about 410. Simultaneous videotapes of the horizontal- and vertical-planes of the test section were also acquired.

Based on detailed analysis of the videotaped information of the flow, and using a methodology developed specifically for this study, the observed flow patterns were classified into stratified wavy, plug, slug, and annular flow regimes. The physical mechanisms that govern the transition between these regimes were identified and discussed. The transitions were observed to occur over a range of flow conditions. Based on the experimental data obtained in this study and on comparison with a number of published flow regime maps, a flow regime map is proposed to predict the prevailing flow regime for air–water flows in small diameter horizontal tubes at near-atmospheric conditions. Some of the literature shows that the range of applicability of this map may be extended to boiling and condensing flows in horizontal tubes. The boundaries of the regimes are represented by transition bands.

Preliminary analysis of the videotaped horizontal- and vertical-planes of the flow showed that the data could be used to provide accurate measurements of the concentration of the air-water interfaces in stratified wavy and plug flow regimes. The recorded signals of the optical probes could also be used to obtain information on other parameters, such as frequency of liquid slugs, length, and passing frequency of air bubbles.

Frictional pressure gradients and average void fractions were found to be flow regime-dependent. Models and correlations used to predict the friction losses and the average void fractions were reviewed and categorized into either (i) generalized models, which do not require prior knowledge of the flow regime, or (ii) regime-specific models,

which were developed based on the particular characteristics of the flow regimes. The adequacy of these models to predict the friction losses and the average void fractions in each of the observed flow regimes was evaluated and discussed.

The measured average void fractions were correlated using the drift-flux model. The results of this work extended the work of França and Lahey (1992) and showed that the weighted average drift velocity for horizontal flow is different from zero. Drift-flux parameters are proposed to predict the average cross-sectional void fraction in horizontal flow for each of the observed flow regimes. The proposed correlations were validated using the experimental data of França and Lahey (1992).

Based on the observed disposition of the phases in each flow regime, simplified two-fluid models were developed to predict the frictional pressure gradients. Results of these models clearly showed that the interactions between the phases at the interface have considerable effects on the velocity profiles within the phases such that, in general, the conventional definitions of the friction factors are invalidated. The measured void fractions and pressure gradients were used to obtain an expression for the interfacial friction factors for each flow regime. These expressions include a waviness coefficient to correct for the effects of the gas–liquid interface conditions on the velocity profiles in the phases, and for the increase in the interfacial area of the phases. Correlations are proposed to predict the interfacial friction factor for certain regimes. The proposed correlations were compared with existing correlations.

The contributions of this experimental investigation are as follows:

- (i) an improved fundamental understanding of mechanisms that govern the transition between the flow patterns;
- (ii) a flow regime map, which can be used to identify the prevailing flow regimes for air–water flows in horizontal tubes at near atmospheric conditions;

- (iii) validated drift-flux models to predict the average void fractions in each flow regime;
- (iv) review and assessment of adequacy of models used to predict the frictional pressure gradients and average cross-sectional void fractions in each flow regime;
- (v) experimental data that could be used to provide accurate measurements of the concentration of the gas–liquid interfaces, and important characteristics of the slug and plug flow regimes.

الإهداء

إلى أمّي ، أبي وكلّ العائلة.

Acknowledgements

I wish to express my profound gratitude to my supervisor, Professor J.T. (Terry) Rogers, for his patient guidance and constant encouragement. Throughout my graduate studies at Carleton University, Terry was always there to explain things to me, from two-phase flow to reactor safety and thermalhydraulics. His help and advice are greatly appreciated.

The skilful work and the assistance provided by the technical staff of the department of Mechanical and Aerospace Engineering during the preparation of the experimental facility is recognized.

I would like to thank my colleagues at SAIC Canada and the Ottawa branch of the Canadian Nuclear Society for their support and encouragement. The generosity of SAIC Canada (Terry Jamieson, Vice-President) in giving me access to their facilities while writing this thesis is also acknowledged.

Finally, I would like to thank my parents, family, and friends for their long-term support and encouragement. They certainly helped me maintain my morale and sense of equilibrium, especially during the final stage of preparing this manuscript.

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Glossary

Terms designated with a dagger (†) the first time they are introduced in this dissertation are explained in this section.

Term	Description
average density	weighted mean density of the two-phase (gas–liquid) mixture.
CANDU	CANada Deuterium Uranium, a water cooled nuclear power reactor developed in Canada by AECL (Atomic Energy of Canada Limited.)
CATHENA	Canadian Algorithm for Thermal Hydraulic Network Analysis, a state-of-the-art thermalhydraulics code developed by AECL to study the thermal behaviour of the CANDU reactors and related equipment.
CISE	empirical correlation to calculate the average void fraction developed at CISE Laboratories in Milan, Italy.
flow regime	identifies the relative topographical configuration assumed by the gas and liquid phases. This term is used interchangeably throughout this dissertation with “flow pattern”.
homogeneous model	model assumes the phases are thoroughly mixed and behave like single-phase flow with weighted average properties of the phases.
mass quality	the ratio of the gas mass flow rate to the total mass flow rate of the mixture.
mass velocity	mass flow rate per unit area.
separate-cylinders model	model assumes the phases are separated by one continuous boundary and flow with a certain relative velocity.
slip ratio	is the ratio of the gas velocity to the liquid velocity.
superficial mass velocity	mass flow rate per unit area calculated using the superficial velocity.
superficial velocity	the velocity the phase would attain if it occupies the total channel’s cross-sectional area alone at the same volumetric flow rate.
two-component mixture	when the gas and liquid phases are for different substances, e.g., gas–oil flow, air–water flow.
two-fluid model	in this model, the phases are modeled separately with mass, momentum and energy exchanged between the phases.
two-phase flow friction multipliers	multipliers used to relate frictional pressure gradients in two-phase flow to that of the gas or liquid phases flowing alone in the channel.

Term	Description
void fraction	time-averaged fraction of flow area or volume which is occupied by the gas phase. Note that the term void is actually misleading since there is no void but gas or vapour.
volumetric quality	the ratio of the volume flow rate of the gas phase to the total volume flow rate of the mixture.
wall shear stress	shearing stress imposed on the fluid by the wall of the channel.

Nomenclature

The choice of symbols used in this document is, in as much as possible, consistent with the symbols used by Collier and Thome (1994).

Symbol	Units	Description (Value / Equation)
A	m^2	cross-sectional area of test tube (total flow area of the mixture)
A_r	m^2	flow area occupied by the liquid phase
A_g	m^2	flow area occupied by the gas phase
C	-	constant in friction factor equation (Eq. 3.16)
C	-	constant in Lockhart–Martinelli correlation (Eq. 3.1)
C_{go}	-	friction factor calculated based on the mass velocity of the mixture and using the properties of the gas phase (Eq. 3.30b)
C_{lo}	-	friction factor calculated based on the mass velocity of the mixture and using the properties of the liquid phase (Eq. 3.30b)
C_o	-	phase distribution in drift-flux model (Eq. 3.33)
C_w	-	waviness coefficient used in the two-fluid models (Eq. 3.53)
d	m	internal diameter of test tube (0.0254 m)
d_{eg}	m	equivalent hydraulic diameter of the gas flow area
d_{el}	m	equivalent hydraulic diameter of the liquid flow area
E	-	parameter used in Friedel's (1954) correlation (Eq. 3.30b)
E_1	-	parameter used in the CISE correlation (Eq. 3.31d)
E_2	-	parameter used in the CISE correlation (Eq. 3.31e)
f	-	friction factor (Eq. 3.16)
F	-	modified Froude number, Taitel and Dukler (Eq. 2.5)
F	-	parameter used in Friedel's (1954) correlation (Eq. 3.30c)
f_i	-	gas–liquid interfacial friction coefficient
$f_{i,eq}$	-	equivalent friction factor calculated using the measured pressure gradients (Eq. 3.55)
Fr	-	Froude number (Eq. 3.30e)
G	$\text{kg/m}^2\cdot\text{s}$	mass velocity of the mixture
g	m/s^2	gravitational acceleration (9.81 m/s^2)
G_g	$\text{kg/m}^2\cdot\text{s}$	gas mass velocity
G_l	$\text{kg/m}^2\cdot\text{s}$	liquid mass velocity
H	-	parameter used in Friedel's (1954) correlation (Eq. 3.30d)
h_L	m	average height of the liquid film (Eq. 6.7)

Symbol	Units	Description (Value / Equation)
J	m/s	superficial velocity or volumetric flux of the mixture ($=J_g+J_l$)
J_g	m/s	gas superficial velocity
J_g^*	-	dimensionless gas volumetric flux (Eq. 2.12)
J_l	m/s	liquid superficial velocity
K	-	parameter of Taitel and Dukler (Eq. 2.6)
l/d	-	length to diameter ration of the test section
L_B	m	average length of the gas bubbles in plug flow regime
L_S	m	average length of the liquid slugs in plug flow regime
m	-	exponent in equation (Eq. 3.16)
$\dot{m}$	kg/s·m ²	total mass flux of the mixture ($=\dot{m}_g + \dot{m}_l$)
$\dot{m}_g$	kg/s·m ²	mass flux of the gas phase
$\dot{m}_l$	kg/s·m ²	mass flux of the liquid phase
$\dot{m}_t$	kg/s	total mass flow rate of the liquid phase
$\dot{m}_e$	kg/s	mass flow rate of the liquid phase entrained in the gas (Eq. 6.7)
Re_f	-	Reynolds number of liquid film in annular flow (Eqs. 6.5 and 6.6)
$\dot{m}_t$	kg/s	total mass flow rate of the mixture
P_{avg}	Pa	average pressure in the test section (calculated as the arithmetic average of the inlet and exit pressures)
P_i	m	gas–liquid interfacial perimeter
P_{wg}	m	wall–gas contact perimeter
P_{wl}	m	wall–liquis contact perimeter
Q	m ³ /s	volume flow rate
Re	-	Reynolds number (Eq. 3.17)
Re_g	-	gas phase Reynolds number
Re_i	-	interfacial Reynolds number (Eq. 3.39d)
Re_l	-	liquid phase Reynolds number
Re_f	-	Reynolds number of liquid film in annular flow (Eqs. 6.5 and 6.6)
$Re_{f,crit}$	-	critical film Reynolds number in annular flow (Eqs. 3.43)
S	-	ratio of gas to liquid velocities (slip ratio) (Eq. 3.6)
S	-	parameter used in the CISE correlation (Eq. 3.31a)
T	-	parameter of Taitel and Dukler (Eq. 2.4)
T_m	°C	average temperature of the mixture
u_g	m/s	gas velocity averaged over the gas flow area
u_l	m/s	liquid velocity averaged over the liquid flow area
V_{gi}	m/s	weighted mean drift velocity in drift flux model (Eq. 3.34)

Symbol	Units	Description (Value / Equation)
We	-	Weber number (Eq. 3.30f)
X	-	Lockhart-Martinelli parameter (Eq. 2.3)
x	-	axial coordinate (along the test tube axis)
X_{tt}	-	Lockhart-Martinelli parameter assuming turbulent gas and liquid phase (Eq. 3.25)
y	-	parameter used in the CISE correlation (Eq. 3.31b)
Y	-	Taitel and Dukler (1976) parameter (Eq. 2.7) ($Y=0$ for horizontal flow)
$\left(\frac{dP}{dx}\right)_{TP,fric}$	Pa/m	frictional pressure gradient (Eq. 3.12)
$\left(\frac{dP}{dx}\right)_{TP,gr}$	Pa/m	gravitational pressure gradient (Eq. 3.14)
$\left(\frac{dP}{dx}\right)_{TP,acc}$	Pa/m	accelerational pressure gradient (Eq. 3.13)
$\left(\frac{dP}{dx}\right)_{g,fric}$	Pa/m	single-phase gas frictional pressure gradient
$\left(\frac{dP}{dx}\right)_{l,fric}$	Pa/m	single-phase liquid frictional pressure gradient
$\left(\frac{dP}{dx}\right)_{lo,fric}$	Pa/m	liquid phase frictional pressure gradient calculated using the total mass velocity of the mixture (Eq. 3.21)
$\left(\frac{dP}{dx}\right)_{go,fric}$	Pa/m	liquid phase frictional pressure gradient calculated using the total mass velocity of the mixture (Eq. 3.21)
$\left(\frac{dP}{dx}\right)_{exp,l-3}$	Pa/m	measured pressure gradient defined in Section 6.3.1
$\left(\frac{dP}{dx}\right)_{A,min}$	Pa/m	limiting pressure for annular flow, Bandel's (1973) correlation (Eq. 3.35a)
$\left(\frac{dP}{dx}\right)_{St,max}$	Pa/m	limiting pressure for stratified flow, Bandel's (1973) correlation (Eq. 3.35b)
$\left(\frac{dP}{dx}\right)_{TP,tot}$	Pa/m	total pressure gradient (Eq. 3.11)

Greek Symbols

<i>Symbol</i>	Units	Description (Value / Equation)
$\bar{\rho}$	kg/m ³	average density of the two-phase flow mixture
ρ_{air}	kg/m ³	density of air at standard atmospheric conditions (=1.2 kg/m ³)
ρ_g	kg/m ³	density of the gas phase
ρ_l	kg/m ³	density of the liquid phase
ρ_{wat}	kg/m ³	density of water at standard atmospheric conditions (=998 kg/m ³)
α	-	Time-averaged cross-sectional average void fraction
α_h	-	average void fraction using homogeneous flow model
α_p	-	void fraction measured averaged over the length of the air bubbles
$\alpha_{\text{loc},9}$	-	time-averaged local void fraction measure at the most upper location in the channel
β	-	volumetric quality of the mixture ($\beta = Q_g / (Q_g + Q_l)$)
β	-	parameter used in the CISE correlation (Eq. 3.31c)
δ	m	film thickness
σ	N/m	liquid surface tension
σ_{wat}	N/m	water surface tension at standard atmospheric conditions
λ	-	Baker's (1954) correlation parameter (Eq. 2.1)
ψ	-	Baker's (1954) correction parameter (Eq. 2.2)
θ	°	angle of inclination of the tube from horizontal
θ	°	angle of defined in Fig. 3.9, two-fluid stratified flow model.
π	-	constant (3.1416)
χ	-	mass quality of the mixture
ϕ_g	-	two-phase flow friction multiplier (Eq. 3.18)
ϕ_{go}	-	two-phase flow friction multiplier (Eq. 3.20)
ϕ_l	-	two-phase flow friction multiplier (Eq. 3.19)
ϕ_{lo}	-	two-phase flow friction multiplier (Eq. 3.20)
μ_g	N·s/m ²	dynamic viscosity of the gas
μ_h	N·s/m ²	dynamic viscosity of the mixture using homogeneous flow model (Eq. 3.23)
μ_l	N·s/m ²	dynamic viscosity of the liquid
μ_m	N·s/m ²	dynamic viscosity of the mixture
μ_{wat}	N·s/m ²	dynamic viscosity of water at standard atmospheric conditions
τ_i	N/m ²	gas-liquid interfacial shear stress
τ_{wg}	N/m ²	wall-gas shear stress
τ_{wl}	N/m ²	wall-liquid shear stress

Subscripts

Symbol	Description
air	air at standard atmospheric conditions (Pressure = 10^5 Pa (abs.), and temperature = 20°C)
ave	average value
exp	experimental value
crit	critical
fric	friction
g	gas phase
go	gas only
h	homogeneous model
l	liquid phase
lo	liquid only
loc	local
m	mixture
min	minimum
max	maximum
pre	predicted value
s	superficial
TP	two-phase flow
w	wall
wat	water at standard atmospheric conditions (Pressure = 10^5 Pa (abs.), and temperature = 20°C)

Chapter 1

Introduction

1.1 General

Many engineering processes involve the presence of more than one phase of a substance, and some necessitate a phase change for processing materials or for energy transfer, such as in petroleum refining and production, chemical and food processing, treatment of wastes, and energy production (from both conventional and alternative sources). This type of flow, originally identified as “flow of complex mixtures,” is known as multi-phase flow. Since the 1940s, engineers and scientists have been aware of the large effects caused by the presence of the second phase in the system.

1.2 Two-phase Flow

The term two-phase flow refers to the simultaneous flow of any two of the three phases of gas, liquid and solid. The flow of two immiscible liquids, such as oil and water, is commonly referred to as two-component flow. In this thesis, the term two-phase flow is used only to refer to the simultaneous flow of the gas and liquid phases.

Gas–liquid flow is probably the most widely encountered type of two-phase mixture, and for that reason it has been the subject of many theoretical and experimental investigations. Gas–liquid mixtures are encountered in a number of engineering processes. This type of flow is of great interest to mechanical, nuclear, and chemical engineers because of the simultaneous presence of the gas and liquid phases in process equipment. Examples of such equipment include water-cooled nuclear power plants, steam generators, condensers, heat exchangers and, oil and gas pipelines. Verification of the safety of nuclear reactors requires a good understanding of heat and mass transfer under transient conditions in two-phase flow. The power of nuclear reactors is often governed by the thermalhydraulic behaviour of a two-phase coolant. The CANDU[†] reactor is unique in having horizontal rather than vertical fuel channels: the two-phase

behaviour of the coolant plays an important role under normal and potential accident conditions in a CANDU. Additional applications of horizontal gas–liquid flow are found in boilers, condensers, refrigeration, and air-conditioning equipment.

The flow of gas–liquid mixtures is a complex phenomenon. Much of this complexity can be attributed to the unstable and deformable interfaces of the phases, which have a destabilizing effect on the flow, thus resulting in a disorderly and random distribution of the phases.

1.2.1 Distribution of the Gas and Liquid Phases

In most single-phase flow problems it is important to distinguish whether the flow is laminar or turbulent, and whether flow separation or secondary flows exist. These distinctions provide useful information about the behaviour of the flow and help select the most appropriate models to analyze the flow. Although these distinctions are equally useful in the study of two-phase flow systems, the distribution of the phases is perhaps of greater importance. As will be discussed later, many of the two-phase thermalhydraulics parameters are strongly dependent on the distribution of the phases within the channel.

The flow of a gas–liquid mixture in a pipe exhibits a variety of distributions of the gas–liquid interfacial surfaces. The distributions of the phases are usually categorized into different regimes or patterns¹ (analogous to laminar, turbulent flow regimes in single-phase flow). The ranges of occurrence of the two-phase flow patterns are generally determined experimentally. Results of these experimental studies are usually presented in diagrams called flow-regime[†] maps that indicate the relative conditions under which each pattern can be expected to prevail.

1.2.2 Methods of Analysis of Two-phase Flows

Most of the methods proposed for the treatment of two-phase flow systems are basically extensions of those already developed for single-phase flow systems. As a

¹ The terms regime and pattern are used interchangeably throughout this document.

simplified approach to the treatment of the mixtures, earlier workers considered the phases to be thoroughly mixed, homogeneously distributed throughout the conduit, and flowing at the same velocity. Suitable averaged properties are determined and the mixture is treated as a single-phase flow. This model is known as the homogeneous model.

Over the past five decades, great strides have been made in the knowledge of two-phase flow, stimulated mainly by the need of the nuclear industry. Progress in qualitative and quantitative understanding of flow behaviour has permitted some improvement in deterministic modelling and analysis of the transient thermalhydraulics performance of many two-phase flow systems, such as drift-flux and two-fluid models. However, more accurate and reliable models are needed to predict the transient thermalhydraulics behaviour of two-phase flow systems. Some models now used in the analysis of two-phase flow systems are generally viewed as crude representations of the flow behaviour.

The two-fluid model is considered to be the most accurate two-phase flow formulation because of its detailed treatment of the phase interactions at the interface. This model allows for the transfer of mass, momentum, and energy from one phase to another through the interfacial area separating the two phases. The main obstacle to a workable theoretical model is the lack of detailed information about the topological structure of the mixture.

As will be discussed later, most of the models used in two-phase flow studies are general, i.e., they were developed independently of the flow regimes. Whalley (1980) and others argued that it is unlikely that substantial improvements in accuracy can be made until modelling methods are based on the individual characteristics of the flow patterns. Therefore, the preferred approach is similar to that used in single-phase flow studies, where the flow is classified as either laminar or turbulent, and specific models based on experimental results are used to analyze it. Therefore, models developed for two-phase flow studies should be regime-specific. This highlights the need for proper

classification and identification of the various configurations assumed by the phases in the channel. Detailed information on the disposition and interaction of the phases, for each flow regime, is considered by many investigators to be the foundation upon which more reliable theoretical models can be established.

1.2.3 Experimental Studies in Two-phase Flow

Well-documented theoretical solutions are available only for the treatment of single-phase flow in simple geometries and under certain idealized conditions. For most practical problems, e.g., flows in arbitrary geometries, investigators rely on experimental results to achieve reasonable predictions of flow behaviour. Similarly, in two-phase flow, experimental data are the main source of information about the behaviour of flow. As indicated earlier, most of the models currently used in the analysis of two-phase flow systems are either empirical or based on simplified assumptions regarding the behaviour of the flow.

It is important to note that uncertainties in the measurements of two-phase flow parameters are considerably higher than those associated with the measurements of single-phase flow parameters. Collier and Thome (1994) indicated that typical standard deviations in the measurement of friction factors or pressure gradients are about 40–50%, which is large with respect to single-phase flows. Schmidt and Friedel (1997) consider models that predict pressure gradients in two-phase flow to be sufficiently accurate for engineering purposes if 80% of the data are within $\pm 30\%$. Hashizume and Ogawa (1987) collected 2281 data points representing 23 sets from 11 different studies from 1956 to 1972 for air–water and steam–water flows from the heat transfer and fluid flow services (HTFS) data bank in England. Only 60% of the pressure drop measurements were predicted within $\pm 30\%$.

1.3 Objectives of the Present Study

The objectives of the present study were to obtain detailed fundamental information on the flow patterns of the air and water phases in a horizontal tube at near

atmospheric conditions, the identification and classification of these patterns, and their effects on some of the thermalhydraulics parameters of the flow. In more explicit terms, this study was undertaken for the following:

- to gain a fundamental understanding of distribution of the phases and the mechanisms governing the transitions between the different flow patterns;
- to assess the ability of existing flow regime maps in predicting the observed flow patterns;
- to obtain accurate measurements of the frictional pressure gradient and the void fraction[†] for different flow patterns;
- to review the existing models and correlations used to predict the measured friction losses and void fractions and to assess their applicability for each of the observed flow patterns;
- to develop validated models to predict the frictional pressure gradient and void fraction for each of the observed flow patterns; and
- to obtain measurements in certain flow patterns of the interface area concentration, i.e., the interface area per unit volume.

1.4 Thesis Organization

This thesis is organized as follows:

- (i) The various classifications of flow patterns and mechanisms governing the transitions between these patterns are presented and discussed in Chapter 2. Some of the most widely accepted flow regime maps are also presented.

- (ii) A summary of the models and correlations used in the analysis of the gas–liquid two-phase flow systems is given in Chapter 3. Various models and methods of analysis currently used for the calculations of the pressure losses, the average cross-sectional void fraction and the gas–liquid interfacial area concentration are presented.
- (iii) Chapter 4 provides general descriptions of the experimental facilities and the measurement techniques used in this experimental investigation. Reduction and analysis of the experimental data are also described.
- (iv) The observed flow patterns and the transition mechanisms are identified and described in Chapter 5. The existing flow regime maps are evaluated and compared with the experimental results. This chapter also provides detailed discussions of the observed flow pattern transition mechanisms. Based on these results a flow regime map is proposed for the prediction of the prevailing flow regime under various conditions.
- (v) The measured friction losses, average void fraction, and interfacial area concentration are presented in Chapter 6. The experimental results are compared with the prediction of the models and correlations discussed in Chapter 3.
- (vi) Chapter 7 summarizes the main conclusions drawn from this investigation and provides recommendations for further work.

Raw experimental data, the experimental procedures, listings of various computer programs (used to acquire or to analyze the experimental data), and derivations of some models are placed in a number of appendices placed at the end of the thesis.

Literature Review: Two-phase Flow Patterns and Pattern Transitions

2.1 Introduction

A gas–liquid mixture flowing in a pipe can exhibit a variety of shapes and distributions of the interfacial surfaces of the phases. The configurations of these interfacial surfaces are of critical importance in determining the characteristics of the flow. Since it is impossible to specify the spatial and temporal distribution of the phases, the internal structure of the flow is usually classified in a number of characteristic patterns or regimes. The conditions under which these patterns are expected to occur are usually presented as diagrams called flow regime maps.

Based on the findings of many experimental studies, it is well established that the distribution of the interfacial surfaces depends on the fluid properties, the flow rates of the two phases, and the channel geometry and orientation.

The identification of flow patterns and the definitions of the transitions between them were the subject of numerous investigations. Govier and Aziz (1972), Delhayé et al. (1981), and Hewitt (1982) published detailed reviews of the earlier work on two-phase flow patterns and pattern transitions. Rogers (1985) provided a comprehensive assessment of the regime boundaries and interphase drag models used in advanced thermalhydraulics codes for two-phase vertical up-flow. Collier and Thome (1994) provided reviews of more recent experimental and semitheoretical studies.

This chapter provides brief descriptions of the flow patterns, and a review of some of the pattern classifications, flow regime maps, and the mechanisms governing the

transitions between these patterns. Detailed discussions of the literature are included in the appropriate chapters of the thesis to allow the discussion points to be supported by the results of this experimental study.

2.2 Gas–Liquid Flow Patterns

A number of classifications and subdivisions of the spectrum of the two-phase flow patterns were reported in the literature. Before discussing these classifications, brief descriptions are given of the most widely accepted classifications of flow patterns in both vertical and horizontal flows. The vertical flow patterns are presented for comparison purposes and because some of the transition mechanisms between the patterns are the same in both cases. Barnea (1987) proposed a unified model for predicting the flow-pattern transitions for the whole range of pipe inclinations, i.e., from vertical up-flow to vertical down-flow, including horizontal flow.

2.2.1 Patterns in Vertical Co-current Up-flow

A number of flow patterns have been identified in upward co-current² flow in a vertical pipe. These patterns are usually grouped in the major regimes described below and are illustrated in Fig. 2.1.

- (i) *Bubble flow*: This regime is characterized by a dispersion of discrete bubbles flowing in a continuous liquid phase. The bubbles are generally spherical if their diameter is less than 1 mm; if greater than 1 mm in diameter, the bubbles tend to have various distorted shapes. The number of bubbles increases with increasing the gas flow rate.
- (ii) *Slug flow*: This flow pattern is composed of a series of large elongated bubbles separated by aerated slugs of water. As illustrated in Fig. 2.1, the bubbles usually have nearly the same diameter as that of the pipe.

² when both phases are flowing in the same direction.

(iii) *Churn flow*: This regime is similar to slug flow, except it is much more chaotic and unstable. At relatively higher flow rates, structures of the large bubbles in the slug flow pattern become unstable and eventually break down, leading to an oscillatory motion of the liquid. This regime is often considered as a transitional regime because of its unstable behaviour. The churn flow pattern is sometimes referred to as semiannular flow.

(iv) *Annular flow*: In this pattern, the liquid is swept up around the entire wall of the pipe in an annular form, while the faster moving air occupies the core of the tube and entrains small droplets of liquid dispersed in the form of a mist (see Fig. 2.1). The liquid film has a wavy and irregular surface and may contain gas bubbles. This regime is encountered at relatively high mass velocities[†].

Under certain annular flow conditions, as the liquid flow rate is increased the droplets entrained in the gas core coalesce into large lumps or “wisps” of liquid, as illustrated in Fig. 2.1. Hewitt et al. (1978) considered this type of flow as a separate regime, which they identified as wispy-annular flow”

2.2.2 Patterns in Horizontal Flow

The following descriptions are based on the most widely accepted classifications of the gas–liquid flow patterns in horizontal flow, i.e., those of Alves (1954), Baker (1954), Govier and Omer (1962), and Mandhane et al. (1974). These patterns are shown in Fig. 2.2.

In horizontal two-phase flow, because of the density differences of the gas and liquid phases, gravity causes the gas, the less dense phase, to migrate to the top of the channel. As a result, horizontal two-phase flow has flow patterns that are not axisymmetrical.

(i) *Dispersed bubble flow*: This flow pattern is characterized by bubbles of gas randomly dispersed in a continuous liquid phase. The bubbles are relatively small

in comparison with the diameter of the tube. Because of gravity, the gas bubbles tend to travel in the upper half of the tube as illustrated in Fig. 2.2a.

- (ii) *Plug flow*: In plug flow, the gas phase is present in the form of elongated bubbles transported in an aerated continuous liquid phase in the upper half of the tube (see Fig. 2.2b). This flow is also referred to as elongated bubble flow by Govier and Aziz (1972) and Mandhane et al. (1974).
- (iii) *Stratified smooth flow*: In this flow, the gas phase travels along the upper half of the tube while the liquid flows along the bottom with no significant interfacial waves. As shown in Fig. 2.2c, this flow has the simplest configuration of all the horizontal flow patterns. The gravity separation is complete, and the stratified flow regime occurs at relatively low gas and liquid mass flow rates.
- (iv) *Stratified wavy flow*: As the gas velocity increases, the faster moving gas causes waves at the gas–liquid interfaces, resulting in a wavy flow pattern. As shown in Fig. 2.2d, the gas phase might contain some liquid droplets, and at the same time the liquid waves might be aerated.
- (v) *Slug flow*: With increased gas velocity, the magnitude of the waves increases. Ultimately, the waves build up and reach the upper wall of the tube to form some liquid packets, also liquid slugs. These liquid slugs are then transported at the higher velocity of the gas (Fig. 2.2e). Unlike plug flow, in which the elongated bubbles of gas are transported by the liquid phase, in slug flow the liquid slugs are carried by the faster moving gas flow.

The slug flow regime is highly undesirable in practical applications. The faster moving liquid slugs are usually associated with sudden pressure pulses and severe pressure oscillations that can cause damage to downstream equipment. Prediction of the onset of slugging has been the focus of many theoretical and experimental studies, e.g., Kordyban (1985), Hubbard and Dukler (1966), Wallis and Dobson

(1973), Taitel and Dukler (1977), Kubie (1979), Lin and Hanratty (1987a), Barnea and Taitel (1993), and Jepson and Taylor (1993).

(vi) *Annular flow regime*: In this regime, the liquid is swept up around the perimeter of the tube in an annular form, while the faster moving air flows in the core of the tube and entrain small droplets of liquid dispersed in the form of a mist. Small bubbles of gas may also be entrained in the liquid film. As shown in Fig. 2.2f, because of gravity, the thickness of the liquid film is higher at the bottom of the tube than at top. This regime occurs at a relatively high mass flow rate of the gas phase.

2.3 Classifications and Identification of Flow Patterns

As stated earlier, a number of classifications and subdivisions of the spectrum of the two-phase flow patterns have been identified in the literature. Hewitt (1978), Delhay (1981), Vince and Lahey (1982), and Jones (1983) have published reviews of the methods used to identify and classify the flow patterns.

Kosterin (1949), Alves (1954), Baker (1954) and Hoogendoorn (1959) were among the first investigators to make a systematic attempt to classify the distributions of the interfacial surfaces of the phases into a number of flow patterns. Most of these earlier classifications were based, largely on visual observations of the flow, supplemented by still or high-speed photography. Some investigators attempted to classify the flow regimes using gamma- or X-ray techniques or by analyzing variations of some of the two-phase flow parameters, such as pressure gradient and liquid holdup³. Jones and Zuber (1975) and Vince and Lahey (1982) developed X-ray techniques to identify the flow patterns. Govier et al. (1972) supplemented their visual observation by attempting to relate the transitions between the regimes to variations of the pressure gradient. Hubbard and Dukler (1966), Nishikawa et al. (1969) and Kutateladze (1972) are among a number of investigators who attempted to classify the flow patterns by performing spectral

³ term used mainly by chemical engineers and refers to the average volume fraction of the liquid phase.

analyses of fluctuations of the wall pressure and time traces of void fraction measurement instruments. Unfortunately, these techniques have had only limited success, as discussed in the following sections. Many discrepancies still exist between the proposed pattern classifications and flow regime maps. The main reason for the limited success is that the time traces of these instruments capture the random and transitional behaviour of the flow instead of its organized behaviour. Despite the wide range of flow configurations, analysis of the time traces usually results in only three distinct spectra representing clearly distinguishable flow patterns (see Fig. 2.3). For flow patterns at or near the transition boundaries, a superposition of the three spectra is usually observed.

Weisman et al. (1979) realized this difficulty and extended their analysis of pressure drop fluctuations by introducing amplitude analysis in the characterization of the time traces. They suggested specific number criteria on amplitude and frequency to characterize all patterns. Dukler and Taitel (1986) suggested that these specific numbers depend on the specific system being used in the experiment and are not general.

2.3.1 Patterns in Horizontal Flow

Taitel and Dukler (1976a) divided the horizontal two-phase flow patterns spectrum into what they considered the five basic flow regimes: intermittent, stratified smooth, stratified wavy, dispersed bubble, and annular flow patterns. This classification is similar to that of Mandhane et al. (1974), except that in their study, Taitel and Dukler (1976a) amalgamated the slug- and plug-flow patterns, which they considered to be different conditions of the intermittent flow pattern.

Hewitt (1982) identified a subdivision of the slug flow, called semislug flow, as a separate flow pattern. The criterion used to distinguish between the two regimes is that under semislug flow conditions, the liquid slugs do not touch the top of the tube. Lin and Hanratty (1987b) combined visual and photographic observations with spectral analysis of conductance probe signals and wall-pressure fluctuation techniques to identify and categorize the flow patterns. They identified a new flow pattern that they termed pseudo-slug flow. This regime was distinguished from slug flow by the fact that the liquid slugs

touch the top of the tube only momentarily. The characteristics of this regime are different from those of the semislug flow identified by Hewitt (1982). It can be argued that these regimes are different conditions of either the wavy flow or slug flow.

Spedding and Nguyen (1980) observed over 13 distinguishable horizontal flow patterns. Since some of these regimes were basically similar, they were grouped together to give the following four broad flow regimes:

- (i) *Stratified flow*: This includes those regimes in which both phases are continuous, such as stratified smooth and stratified wavy flows. This group also includes cases of annular flow without entrained liquid droplets or gas bubbles.
- (ii) *Bubble and slug flows*: This group is characterized by the gas phase being discontinuous while the liquid is continuous.
- (iii) *Droplet flow*: This refers to regimes in which the liquid phase is discontinuous while gas phase is continuous.
- (iv) *Mixed flow*: This group includes patterns in which both phases are discontinuous, such as annular film and droplet flow.

The above classification of Spedding and Nguyen (1980) considers cases of annular flow as subregions of the broader stratified flow pattern, which they defined as flow without droplets or bubbles. This contradicts the accepted description of the annular flow regime. Researchers have almost invariably observed the occurrence of a mist of liquid droplets in the core of annular flow.

Kokal and Stanislav (1989) studied the flow of oil and air in horizontal and near-horizontal pipes. They divided the flow into the following three basic flow patterns:

- (i) *Gas-dominated flow*: This includes stratified, stratified wavy and annular flow patterns.
- (ii) *Intermittent flow*: This represents plug and slug flow, similar to the intermittent flow pattern defined by Taitel and Dukler (1976a). The authors indicated that this flow pattern is the dominant flow in horizontal pipes.
- (iii) *Liquid-dominated flows*: This represents the dispersed bubble flow.

Again, this classification is very simplistic as the annular flow is grouped with the stratified flow. As will be discussed later in this chapter and in Chapter 5, it is well established in the literature that the annular and stratified flow occur over different flow conditions.

Breber et al. (1980) and Ewing et al. (1999) categorized the horizontal flow patterns into the following four groups:

- (i) *Shear-controlled flows*: This includes cases of annular-like flow patterns.
- (ii) *Gravity-controlled flows*: This includes stratified smooth and wavy flow patterns.
- (iii) *Intermittent flows*: which include plug and slug flow regimes.
- (iv) *Dispersed flows*: such as the bubble flow regime.

The above classification is similar to the one suggested by Taitel and Dukler (1976a), except that the stratified smooth and wavy flow regimes are grouped together.

For condensing and boiling two-phase flow systems, the matter is further complicated by the gradual and continuous change of the vapor and liquid fractions, which is accompanied by a similar change in the spatial distribution of the phases. In a

recent experimental study of condensing flow in horizontal and near-horizontal tubes, Nitheanandan (1993) identified seven major flow regimes and three transitional regimes.

2.3.2 Patterns in Vertical Flow

There are fewer distinguishable flow patterns in vertical flow than in horizontal flow. This may be explained by the fact that in vertical flow, the distribution of the phases' interfacial surfaces are roughly symmetrical around the tube axis, since the gravitational force acts along the flow axis. Exceptions to this observation are the random and highly turbulent bubble and churn flow patterns, which may not be axisymmetrical at any moment. However, on a time average basis, the spatial distribution of the phases in these flow patterns approaches axial symmetry.

For vertical up-flow, the classification presented in Section 2.2.1 is generally accepted by most investigators, such as Collier and Thome (1994), Wallis (1972), and Rogers (1985). Only few other classifications are found in the literature, such as the wispy-annular flow regime identified by Hewitt (1981). Serizawa (1974) suggested that the vertical up-flow patterns be classified as bubble flow, developing slug flow, slug flow, ripple flow, annular or dispersed-annular flow, and mist flow. The mist flow occurs when the liquid film on the wall does not exist or is disrupted.

For downward co-current gas-liquid flow, Oshinowo and Charles (1974) identified six different flow configurations. These configurations include the main four flow patterns of the up-flow described in Section 2.2.1, and two other flow patterns that they called falling film and bubbly-falling film. These patterns are shown in Fig. 2.4.

2.3.3 Summary of Pattern Classifications

Numerous investigators have tried to classify the two-phase flow patterns. Earlier classifications of the flow patterns were subjective since they were based almost entirely on visual observations of the flow. Attempts to classify the regimes by analyzing variations in some of the two-phase flow parameters, such as pressure drop and void

concentration, had limited success at best. This can be explained by the fact that the transitional behaviour of the flow is reflected in the measured parameters.

The flow patterns described in Sections 2.2.1 and 2.2.2 are the most widely accepted classifications of horizontal and vertical flows. See Chapter 5 for further discussion of the classifications of the flow patterns.

2.3.4 Boundaries of Flow Patterns

Weisman et al. (1979), Breber et al. (1980), Pearce (1982), and Rogers (1985) indicated that all classifications of flow regimes are somewhat idealized, as such definite and distinct patterns do not usually exist, and the various patterns gradually merge into one another. Weisman et al. (1979) suggested that the transition from one regime to another generally takes place over a range of flow conditions. They indicated that during the transition from slug to dispersed flow regime, an annular-like pattern was observed. As shown in Fig. 2.5, Weisman et al. (1979) plotted the transition curves as shaded areas to indicate the regions in which the flow pattern transition occurred. These transition bands can be explained by the observer's doubt about the nature of the flow. Pearce (1982) recognized this difficulty and suggested that the regime boundaries be situated in the middle of the uncertainty bands.

The transition behaviour of the patterns was also observed with vertical flow patterns. In studying the steam–water flow regime transitions in vertical upward flows, Rogers (1985) stated that the transition phenomena between the flow patterns are gradual. Following others, Rogers (1985) recommended that in presenting flow regime maps, the transition boundaries of the patterns should be shown by bands rather than lines.

2.4 Horizontal Flow Pattern Maps

The literature indicates that Kosterin (1949) was possibly the first to suggest the use of flow regime maps. Since then, a number of flow regime maps were proposed for identifying the prevailing flow pattern in a given two-phase flow situation. Some of the

well-cited maps in the literature were developed by Baker (1954), Hoogendoorn (1959), Mandhane et al. (1974), Taitel and Dukler (1976a), Spedding and Nguyen (1980), Breber et al. (1980), and Lin and Hanratty (1987b).

A flow regime map is essentially a diagram that indicates the range of occurrence of the various flow patterns. A significant number of two-phase (gas–liquid) horizontal flow maps were suggested. The following are the most frequently cited flow maps in the literature:

2.4.1 Baker's Map (1954)

Based on the experimental data of Jenkins (1947), Gazely (1948), Kosterin (1949) and Alves (1954), Baker (1954) developed a two-phase flow regime map that is still widely used, especially in the gas and oil industries. Bell et al. (1970) simplified the coordinates used by Baker. The resulting generalized map is shown in Fig. 2.6, where G_g and G_l are the gas and liquid superficial mass[†] velocities, λ and ψ are the correction coefficients proposed by Baker to adjust for variations in the fluid density and liquid surface tension and viscosity. Since most of the data collected by Baker were for an air–water system, the correction factors are referenced to that of air and water at standard atmospheric conditions. The corrections factors proposed by Baker (1954) are defined as follows:

$$\lambda = \left[\left(\frac{\rho_g}{\rho_{air}} \right) \left(\frac{\rho_l}{\rho_{wat}} \right) \right]^{1/2} \quad (2.1)$$

$$\psi = \left[\left(\frac{\mu_l}{\mu_{wat}} \right) \left(\frac{\rho_{wat}}{\rho_l} \right) \right]^{1/3} \left(\frac{\sigma_{wat}}{\sigma} \right) \quad (2.2)$$

where ρ_{air} is the density of air, ρ_{wat} , μ_{wat} and σ_{wat} are the density, viscosity, and surface tension of water, respectively. All reference air and water properties are at standard atmospheric conditions.

Spedding and Spence (1993) indicated that the flow regime map of Baker (1954) failed to handle the effects of pipe diameter. This is to be expected, since most of the data used by Baker, with the exception of that of Kosterin (1949), were for a 25.4-mm pipe.

Troniewski and Ulbrich (1984) analyzed 21 flow pattern maps for gas–liquid flow in horizontal pipes. They suggested an improvement on Baker's (1954) map by placing the bubble flow regime below the stratified pattern, i.e., at the lower left corner of the map (at low gas mass velocity). Their reasoning for this variation was explained by the fact that when only a liquid is flowing it would fill the entire cross-section area. The introduction of gas would result in bubbly flow. However, they provided no quantitative evidence to support this hypothesis. Spedding and Spence (1993) compared the transitions proposed by Troniewski and Ulbrich (1984) with their experimental data. They concluded that the map proposed by Troniewski and Ulbrich showed no significant improvement over the Baker map, and that bubble flow regime should in fact be placed to the right of the mapping area, as suggested by Baker (1954).

2.4.2 Hoogendoorn's Map (1959)

Hoogendoorn (1959) conducted an experimental study of the flow of gas–liquid in horizontal tubes. He studied the effects of pipe diameter and fluid properties, i.e., viscosity, density, and surface tension, as well as effects of pipe roughness on the transition lines between the flow patterns.

Hoogendoorn correlated his results using the mapping parameters proposed by Kosterin (1949), i.e., volumetric quality[†] (β) as the ordinate and the mixture mean velocity[†] (V_m) as the abscissa (Fig. 2.7). Hoogendoorn also concluded that pipe diameter and liquid properties have no effect on the locations of the transition lines between the regimes.

2.4.3 Mandhane et al. (1974)

Mandhane, Gregory and Aziz (1974) collected about 6000 flow pattern observations of gas–liquid flow in horizontal pipes. Data were collected from published research results prior to that date. After a careful examination of the data bank, about 1178 flow pattern observations for air–water systems were chosen to produce, perhaps, the most commonly accepted horizontal flow patterns map (Fig. 2.8). Its success may be as a result of the wide range of parameter values for the data used to produce the map. The Mandhane map is valid for the range of parameter values listed in Table 2.1.

Mandhane et al. (1974) used the superficial velocities[†] of the phases as mapping parameters. They indicated that the boundaries of the flow patterns are least sensitive to variations of pipe diameter and fluid properties if the superficial velocities were used as mapping parameters.

Table 2.1: Range of parameter values of flow regime map of Mandhane et al. (1974).

Parameter	Range
Pipe diameter (d)	12.7 – 165.1 mm
Liquid density (ρ_l)	705 – 1009 kg/m ³
Gas density (ρ_g)	0.80 – 50.5 kg/m ³
Liquid viscosity (μ_l)	0.3×10^{-3} – 90×10^{-3} Pa.s
Gas viscosity (μ_g)	1×10^{-5} – 2.2×10^{-5} Pa.s
Surface tension (σ)	24 – 103 mN/m
Liquid superficial velocity (J_l)	0.09 – 7.31 m/s
Gas superficial velocity (J_g)	0.04 – 171 m/s

2.4.4 Taitel and Dukler (1976a)

Taitel and Dukler (1976a) developed simple physical models for predicting the flow regime transitions in horizontal and near horizontal gas–liquid flows. This was considered the first clear attempt to break away from reliance on visual observations to predict the transition of flow patterns. However, the models still relied on visual

observations for confirmation of the prevailing flow regime and for identification of the transition mechanisms.

Based on some simple physical concepts and assumptions that will be discussed later, Taitel and Dukler (1976a) derived a mathematical expression for each of the pattern boundaries. Their models, which were supported with some experimental data, incorporated the effects of fluid properties, conduit size, and conduit inclination from the horizontal.

Taitel and Dukler (1976a) categorized the flow into stratified smooth (SS), stratified wavy (SW), intermittent (I), dispersed bubble (DB), and annular (AD) flow patterns. As indicated earlier, the intermittent flow includes plug and slug flow patterns. The resulting map proposed by Taitel and Dukler (1976a) for gas–liquid flow in a horizontal pipe is shown in Fig. 2.9. The map uses different coordinate systems depending on the pattern transition considered. Taitel and Dukler (1976a) argued that since the transitions between the various flow regimes are governed by different mechanisms, it is uncertain that any single set of two groups will be able to correlate all transitions.

The transitions are expressed in terms of the five dimensionless parameters (X , T , F , K , and Y), defined by the following equations:

$$X = \left[\frac{|(dP/dx)_{l,fric}|}{|(dP/dx)_{g,fric}|} \right]^{1/2} \quad (2.3)$$

$$T = \left[\frac{|(dP/dx)_{l,fric}|}{(\rho_l - \rho_g) g \cos \theta} \right]^{1/2} \quad (2.4)$$

$$F = \left(\frac{\rho_g}{(\rho_l - \rho_g)} \right)^{1/2} \frac{J_g}{(d g \cos \theta)^{1/2}} \quad (2.5)$$

$$K = F \times (Re)^{1/2} \quad (2.6)$$

$$Y = \frac{(\rho - \rho_g) g \sin \theta}{|(dP/dx)_{l,fric}|} \quad (2.7)$$

Parameters in the above equations are defined in the glossary section. Taitel and Dukler (1976a) introduced parameter (Y) to correct for channel inclination from the horizontal. Therefore each inclination angle requires a different flow regime map. For horizontal flow, Y is equal to zero (see Fig. 2.9). Variations of the transition curves with inclinations of the flow channel are discussed in Section 2.6.1.1.

For horizontal flow, the patterns are separated by four transition curves controlled by the above dimensionless parameters as follows:

(i) transition from (SW) to (AD) or (I):	X, F
(ii) transition from (DB) or (I) to (AD):	X
(iii) transition from (SS) to (SW):	X, K
(iv) transition from (I) to (DB):	X, T

Transition from stratified wavy flow to either intermittent or annular flow is represented in Fig. 2.9 by curve A. Taitel and Dukler (1976a) indicated that this transition takes place when a finite amplitude wave becomes unstable and grows to reach the top of the tube to form either slug or annular flow regime. They used Kelvin–Helmholtz wave stability criterion to model this transition phenomenon.

In defining the transition between annular flow and intermittent flow, represented by curve B, Taitel and Dukler (1976a) simply assumed that when the equilibrium liquid level occupies more than half of the pipe, intermittent flow will result; otherwise annular flow will develop. This criterion has received a great deal of criticism by other researchers. For instance, Lin and Hanratty (1987b) believe that annular flow is caused by entrainment and deposition phenomenon.

The transition between stratified smooth and stratified wavy (curve C) is modeled using the phenomenon of wave initiation. For horizontal flow, this transition curve can be plotted in terms of parameters K and X .

To separate the dispersed bubble and the intermittent subregions, Taitel and Dukler (1976a) suggested that the dispersed bubble flow regime will develop when there are enough turbulent fluctuations within the liquid phase to overcome the buoyant forces that tend to keep the gas at the top of the tube. The transition between intermittent and dispersed bubble flow regimes is represented in Fig. 2.9 by curve D.

In Fig. 2.10, Taitel and Dukler (1976a) compare the predictions of their models with the flow regime map developed by Mandhane et al. (1974) for air–water flow at atmospheric conditions in a 25-mm horizontal pipe. Although these semitheoretical predictions agree fairly well with the experimental results presented by Mandhane et al. (1974), recent studies have uncovered some deficiencies in the Taitel and Dukler (1976a) models. Experimental studies conducted by Weisman et al. (1979) showed shifts in the location of the transition curves caused by surface tension variations; the parameters suggested by Taitel and Dukler (1976a) do not account for surface tension effects. On the other hand, Weisman et al. (1979) confirmed that there is little or no effect of viscosity on transition lines, whereas the stratified to intermittent transition model presented by Taitel and Dukler (1976a) predicts large shifts due to viscosity effects.

2.4.5 Weisman et al. (1979)

Weisman et al. (1979) studied effects of fluid properties and pipe diameter on two-phase flow patterns in horizontal pipes. The observed boundaries between the patterns were fitted using the following empirical correlations:

Stratified to intermittent transition:

$$\frac{J_g}{\sqrt{g d}} = 0.25 \left(\frac{J_g}{J_l} \right)^{1.1} \quad (2.8)$$

Transition to annular flow:

$$1.9 \left(\frac{J_g}{J_l} \right)^{1/8} = \left[\frac{J_g \rho_g^{1/2}}{(\rho_l - \rho_g) g \sigma} \right]^{0.2} \left[\frac{J_g^2}{g d} \right]^{0.18} \quad (2.9)$$

Transition to dispersed bubble flow⁴:

$$\left[\frac{(dP/dx)_{i,fric}}{(\rho_l - \rho_g) g} \right]^{1/2} \left[\frac{\sigma}{(\rho_l - \rho_g) g d^2} \right]^{-1/4} = 9.7 \quad (2.10)$$

Stratified smooth to stratified wavy transition:

$$\left[\frac{\sigma}{(\rho_l - \rho_g) g d^2} \right]^{0.2} \left(\frac{d J_g \rho_g}{\mu_g} \right)^{0.45} = 8 \left(\frac{J_g}{J_l} \right)^{0.16} \quad (2.11)$$

A comparison of these empirical correlations with other transition models, such as those of Taitel and Dukler (1976a), is given in Chapter 5. In Fig. 2.5, Weisman et al. (1979) compared their results with the well-known map of Mandhane et al. (1974).

Spedding and Spence (1993) compared data collected for 45.5- and 93.5-mm pipes with the correlations proposed by Weisman et al. (1979). They indicated that with the exception of the stratified to stratified wavy transition, which gave good prediction, the other proposed transitions generally showed poor agreement with the experimental data. It should be noted that Spedding and Spence (1993) used the wrong coefficients and constants in the equations proposed by Weisman et al. (1979). For the transition to annular flow (Eq. 2.9 above), Spedding and Spence (1993) used 1.8 instead of 1/8, and $G^{0.5}$ instead of $\rho_g^{0.5}$. Also, in the transition to dispersed bubble flow (Eq. 2.10 above), they used 1.7 instead of 9.7. It is not clear whether these errors were introduced only in typing the equations or in the analysis upon which they based their assessment of the

⁴ This transition is defined by equation (9) in Weisman et al. (1979). It should be noted that the equation, as given by Weisman et al. (1979), contains an error. The equation should read $(g d^2)$ and not $(g^2 d)$. From their discussion, e.g., paragraph ahead of Eq. 9, it clear that this just a typing error.

transitions suggested by Weisman et al. (1979). In any case, the flow pattern classification of Spedding and Spence (1993) is completely different from that used by Weisman et al (1979). As indicated earlier, Spedding and his co-workers, Nguyen in (1980) and Spence in (1993), observed over 13 different flow patterns. This obviously makes it difficult to compare their observations with the transitions between the six flow patterns observed by Weisman et al. (1979) (see Section 2.2.1).

2.4.6 Spedding and Nguyen (1980)

Spedding and Nguyen (1980) studied air–water flow patterns in a 45.5-mm pipe for the entire range of pipe inclination, i.e., from vertical up-flow to down-flow. Based on their experimental data they proposed a series of empirical flow regime maps for different pipe inclinations from the horizontal.

Spedding and Nguyen (1980) observed over 13 distinguishable horizontal flow patterns that they classified into four broader patterns (as discussed in Section 2.3.1) and identified five possible variables for mapping the boundaries of the flow patterns. They concluded that the liquid to gas volume flow rate ratio (Q_l/Q_g) and Froude number (Fr) were the most satisfactory parameters for presenting horizontal two-phase flow pattern maps. The flow regime map proposed by Spedding and Nguyen is shown in Fig. 2.11. The solid lines separate the four main regimes and the broken lines separate the subdivisions of the main regimes.

2.4.7 Breber et al. (1980)

Breber et al. (1980) proposed a flow regime map for use in condensing flows based on the semitheoretical models proposed by Taitel and Dukler (1976a) and data from 10 experimental studies of condensing flow. Breber et al. (1980) classified the horizontal flow into the four flow patterns described in Section 2.3.1 and used the Lockhart–Martinelli parameter (X) and the dimensionless gas superficial velocity (J_g^*) as mapping parameters. Note that (J_g^*) is often referred to as the Wallis dimensionless gas velocity and is defined by the following equation:

$$J_g^* = \frac{G_g}{[d g \rho_g (\rho_l - \rho_g)]^{1/2}} \quad (2.12)$$

Substituting $(\rho_g J_g)$ for G_g , the above equation can be written as:

$$J_g^* = \left(\frac{\rho_g}{(\rho_l - \rho_g)} \right)^{1/2} \frac{J_g}{(d g)^{1/2}} \quad (2.13)$$

For horizontal flow ($\cos(\theta) = 0$), (J_g^*) is equivalent to the modified Froude number introduced by Taitel and Dukler (1976a), defined by Eq. 2.5 in this document.

As shown in Fig. 2.12, Breber et al. (1980) allowed for transition bands between the regimes and suggested that the transitions between the flow patterns occur at constant X or J_g^* . Although the map of Breber et al. (1980) was suggested for use in condensing flows, Ewing et al. (1999) showed that it could be used to predict adiabatic air–water flows.

2.4.8 Lin and Hanratty (1987b)

Lin and Hanratty (1987b) proposed a flow regime map based on experimental data collected from an air–water system flowing in horizontal 25.4- and 95.3-mm pipes at near atmospheric conditions. Lin and Hanratty (1987b) examined the reliability of the Mandhane map in predicting the regime transition in large-diameter pipes. They indicated that the Mandhane map failed to correlate the transition from annular to slug flow. This disagreement may be explained by the new pseudo-slug regime introduced by Lin and Hanratty (1987b). Results of this study are shown in Fig. 2.13.

2.5 Flow Pattern Transition Mechanisms in Horizontal Flows

This section provides brief descriptions of the mechanisms that govern the transitions between flow patterns. Fig. 2.14 shows the flow patterns and summarizes the various transition mechanisms suggested in the literature.

2.5.1 Transition to Annular Flow

Two mechanisms have been suggested for the transition from stratified to annular flow. Hoogendoorn (1959) concluded that annular flow is established by entrainment and deposition mechanism. However, Govier and Aziz (1972) and Taitel and Dukler (1976a) argued that annular flow is initiated by large-amplitude waves reaching the top of the tube. If these waves are frequent, the liquid is unable to drain along the walls of the pipe before the following wave adds more liquid to the top of the tube.

Lin and Hanratty (1987b) indicated that both mechanisms are operative. They observed that at low liquid flow rates, the transition to annular flow takes place mainly because of entrainment and deposition, while at high liquid flow rates the transition is initiated by wave disturbances. The relative importance of each transition mechanism depends on the pipe diameter and the liquid flow rate. The entrainment deposition mechanism is dominant if the liquid superficial velocity (J_l) is less than 0.015 m/s for the 25.4-mm pipe and 0.1 m/s for the 95.3-mm pipe.

2.5.2 Transition from Stratified Wavy to Plug or Slug Flow

Taitel and Dukler (1976a) suggested that the transition from stratified wavy to intermittent (plug or slug) flow occurs when a finite amplitude wave becomes unstable and grows to reach the top wall of the tube. They used Kelvin–Helmholtz wave stability criteria in establishing this transition (curve A in Fig. 2.9).

Lin and Hanratty (1987b) observed that the amplitude of the waves grows with an increase in the gas velocity. Then the waves coalesce and form larger waves. These larger waves can subsequently either break up and form a pseudo-slug flow or coalesce with a large-amplitude wave to form a slug flow. The coalescence or break-up of these waves depends on whether enough liquid is present in front of the point of coalescence to sustain the fast moving slug.

2.5.3 Transition to Dispersed Bubble Flow

The semitheoretical models of Taitel and Dukler (1976a) postulate that the transition to dispersed bubble flow regime occurs when the turbulent forces in the liquid phase are strong enough to overcome the buoyancy forces that tend to separate the phases. This suggests that the bubble flow regime occurs at high liquid mass velocities.

2.5.4 Transition between Plug and Slug Flow Regimes

No specific mechanism has been proposed to explain the transition between the plug and slug flow regimes. The literature seems to indicate that either plug or slug flow is established from stratified wavy flow depending on the liquid and gas mass velocities.

2.5.5 Transition between Stratified Smooth and Stratified Wavy Flow Regimes

It is generally accepted that the transition from stratified smooth to stratified wavy is governed by wave generation phenomena. The waves are initiated by the fast moving gas on the free surface of the stratified liquid phase.

2.6 Parameters Affecting the Boundaries Between the Regimes

A number of researchers studied the effects of different parameters, such as fluid properties and pipe geometry and orientation, on the boundaries of the flow patterns. The following is a brief review of the expected effects of some of these parameters.

2.6.1 Inclination of the Flow Channel

The tube inclination is perhaps the most studied parameter because of its wide occurrence in applications such as oil and gas transport pipelines.

The semitheoretical models proposed by Taitel and Dukler (1976a) predict pronounced effects of tube inclination on the boundaries of the flow patterns, especially the wavy to intermittent transition curve. With slight downward inclinations, the liquid tends to move more rapidly resulting in a thinner liquid film. Therefore, the transition from wavy flow is expected to occur at a higher gas flow rate, thus making wavy flow

more prominent than the intermittent flow, i.e., it would occur over a wider range of flow conditions. On the other hand, for upward inclinations the transition to intermittent flow occurs at a lower gas flow rate and the intermittent flow is more prominent.

Barnea et al. (1980, 1986, and 1987) studied the effects of flow inclination on the range of occurrence of flow regimes. Barnea (1987) proposed a unified model for predicting the flow-pattern transitions for the whole range of pipe inclinations, i.e., from vertical up-flow to vertical down-flow and including horizontal flow. The problem with this approach is that the flow patterns change with the channel orientation. Also, the mechanisms that govern the transition between the regimes in horizontal flows are different from those in vertical flows. For example, the entrainment and deposition mechanism, which has been observed to initiate the transition to the annular regime in horizontal flows, is not expected to initiate the same transition in vertical flows. Therefore, it is extremely difficult to establish a set of conditions for the transition of flow regimes over a wide range of flow inclinations.

Nitheanandan (1993) studied the influence of tube inclination on the flow regime boundaries of condensing flows in a 13.6-mm diameter pipe. Within a tube inclination of $\pm 10^\circ$ from the horizontal, Nitheanandan (1993) observed that the boundaries of the annular flow regime do not experience any significant shifts. However, small inclinations of $\pm 1^\circ$ from the horizontal resulted in significant shifts in the locations of the wavy and slug regions. With changes in inclination angles from 5 to 10° , the shifts in these transition boundaries were found to be insignificant.

2.6.2 Tube Diameter

The semitheoretical models proposed by Taitel and Dukler (1976a) (Eq. 2.3 to 2.7) suggest that the transition between stratified smooth and stratified wavy regimes (curve C in Fig. 2.9) and the intermittent to annular transition (curve B) are independent of the pipe diameter. Also, for variations of tube size between 20 and 50 mm, the locations of the boundaries are not sensitive to the tube diameter. These predictions are supported by the experimental observations of Weisman et al. (1979), who indicated that

the pipe diameter has only moderate effects on the locations of the boundaries of the flow regimes.

For larger pipe diameters, such as 300 mm, shifts in the flow regime boundaries are expected. Jepson and Taylor (1993) studied slug flow and its transitions in a 300-mm diameter horizontal pipe. Their results show that the pipe diameter has a large effect on the regime transitions and on the slug characteristics, such as its frequency and length.

2.6.3 Fluid Properties

Hoogendoorn (1959), Taitel and Dukler (1976a), Weisman et al. (1979) and Barajas (1993) studied the effects of properties of the fluid on the locations of the pattern boundaries. Some of the parameters that were studied include viscosity, density of the phases, liquid surface tension, and contact angle.

Hoogendoorn (1959) concluded that the properties of the liquid have no effect on the locations of the transition lines between the regimes. The models proposed by Taitel and Dukler (1976a) predict shifts of the transition lines with variations in the properties of the phases. Significant shifts are observed in the transition from stratified smooth to stratified wavy flow regimes and in the transition to annular flow. This contradicts the findings of Weisman et al. (1979), which predict that the properties of the liquid have only moderate effects on the boundaries of the flow regimes. Taitel and Dukler (1976a) also suggested that using the superficial velocities as mapping parameters could not adequately account for variations in the fluid properties.

Barajas (1993) studied the effect of contact angle on the regime boundaries in a capillary tube (1.6-mm (i.d.) pipe) using different types of tube material. For partially wetting systems ($\theta < 90^\circ$), Barajas (1993) observed that the contact angle has little effect on the transition boundaries. For partially non-wetting systems ($\theta > 90^\circ$), the contact angle has a significant effect on the locations of the transition boundaries. For large diameter pipes (> 10 mm), the effects of the contact angle are expected to be negligible.

2.7 Summary and Discussion

Reliable prediction of the prevailing flow patterns is of considerable practical importance in the analysis of two-phase flow systems. As will be discussed in the following chapters, correlations and models developed for a specific flow regime provide more accurate predictions of the two-phase flow parameters than general models. Therefore, if the prevailing flow pattern can be identified with some confidence, better predictions of the flow parameters can be achieved.

Several flow regime maps were proposed over the past several decades. Unfortunately, significant discrepancies were observed when these maps were compared. These discrepancies were attributed to one or more of the following:

- use of different two-phase flow systems, such as air–water, steam–water, or natural gas and crude oil;
- differences of the test apparatus and instrumentation used;
- differences in the terminology used by the various investigators to describe and identify the flow patterns;
- incorrect drawing of the boundaries between the flow regimes;
- unsuitable selection of mapping parameters that do not provide proper correlation for all regime transitions.

Some of the most referenced horizontal flow regime maps are compared and discussed in Chapter 5. Table 2.2 gives a summary of the experimental studies of horizontal two-phase flow.

Table 2.2: Experimental studies of horizontal two-phase flow patterns and pattern transitions.

Author(s)	Two-phase systems	Mapping parameters	Pipe diameter (mm)	Experimental techniques	Comments
Kosterin (1949)	Air–water	$\beta; V_m$	25, 50, 76, and 101	-	First to suggest use of regime maps
Baker (1954)	Air–water and oil–gas		data from other sources	-	-
Hoogendoorn (1959)	Gas–liquid	$\beta; V_m$	24, 50, 91, and 140	Liquid holdup meter	Effects of fluid properties and pipe (i.d.)
Govier et al. (1972)	-	$G_g; G_l$	-	-	-
Mandhane et al. (1974)	Gas–liquid	$J_g; J_l$	Data from other sources	Data from others	Widely accepted map
Taitel and Dukler (1976a)	Semi-theoretical models	Dimensionless groups	-	-	Widely used correlation
Weisman et. al. (1979)	Various	$G_l; G_g$	12, 25, and 50	Visual, pressure fluctuations	Correlation of exp. results
Breber et al. (1980)	-	$X; J_g^*$	-	-	Simplified flow regime map
Spedding & Nguyen (1980)	Air–water		45.5	Visual observations	-
Pearce (1982)	R-12	$J_l; J_g$	40.9	Axial viewer, light probe	Regime transitions under dif. press.
Lin and Hanratty (1987b)	Air–water	$J_g; J_l$	25.4 and 95.3	Video, conductivity wire probe	studied transition mechanisms
Kokal and Stanislav (1989)	Oil–air	$J_g; J_l$	25, 51, and 76	Pressure, volume sensor	Effects of inclinations on transitions
Spedding and Spence (1993)	Air–water	Various	45 and 93.5	Video, pressure fluctuations, quick closing valves	Studied other theoretical and empirical models

Figure 2.1: Gas-liquid flow patterns in vertical up-flow (reproduced from Collier et al., 1994).

Figure 2.2: Gas-liquid flow patterns in horizontal flow (reproduced from Hewitt, 1970).

Figure 2.3: Power spectral density of wall pressure fluctuations (reproduced from Dukler and Taitel, 1986).

Figure 2.4: Patterns classification in downward cocurrent flow in a vertical pipe observed by Oshinowo and Charles, 1974 (reproduced from Delhaye et al., 1981).

Figure 2.5: Horizontal flow regime map proposed by Weisman et al. (1979).

Figure 2.6: Modified Baker's flow regime map (reproduced from Whalley, 1987).

Figure 2.7: Hoogendoorn (1959) gas-oil flow regime map (reproduced from Govier and Aziz, 1972).

Figure 2.8: Flow regime map proposed by Mandhane et al. (1974), (reproduced from Delhaye, 1981).

Figure 2.9: Diagram showing theoretical boundaries proposed by Taitel and Dukler (1976a) for flow in horizontal channels ($Y=0$).

Figure 2.10: Comparison of horizontal flow regime maps by Taitel and Dukler (1976a) and Mandhane et al. (1974).

Figure 2.11: Flow regime map proposed by Spedding and Nguyen (1980).

Figure 2.12: Simplified flow regime map proposed by Breber et al. (1980).

Figure 2.13: Flow regime map proposed by Lin and Hanratty (1987*b*) compared with that of Mandhane et al. (1974).

Figure 2.14: Some of the suggested mechanisms for flow pattern transitions.

Two-Phase Flow Models and Methods of Analysis

3.1 Introduction

Over the past few decades a number of models and correlations were proposed for the analysis of two-phase flow systems. For this investigation, which seeks, in part, to identify some of the effects that flow patterns may have on the two-phase flow friction losses and void fraction, the models and correlations proposed for the analysis of two-phase flow systems are categorized in two groups:

- (i) *Generalized models and correlations:* This includes all models and correlations developed independently of the flow patterns. Therefore, their use does not require prior knowledge of the prevailing flow pattern. Examples of these models include the homogeneous flow model[†] and a number of empirical correlations.
- (ii) *Flow pattern-dependent models and correlations:* This includes all models developed based on the particular hydrodynamic features of the flow regime, e.g., mechanistic models.

Govier and Aziz (1972), Delhay (1981), Whalley (1987), and Collier and Thome (1994) published reviews of the various models and correlations used in the analysis of two-phase flow systems. The following sections provide definitions and introductions to the parameters considered in this investigation and a review of the most commonly accepted models and correlations used to predict these parameters. This review is not intended to be a detailed examination of all the models and correlations presented in the literature.

3.2 General Relationships

This section will present some of the two-phase flow relationships that can be developed independent of the flow models, such as void fraction and average density[†] of the mixture.

3.2.1 Average Void Fraction

Void fraction (α) is defined as time-averaged concentration of the gas phase. For gas–liquid flow in a channel, the void fraction is defined as the average cross-sectional area occupied by the gas phase to the total cross-sectional area of the channel. Thus,

$$\alpha = \frac{A_g}{A} \quad (3.1)$$

where A is the cross-sectional area of the channel, and A_g is the cross-sectional area occupied by the gas phase. In boiling and condensing flows it is often convenient to express the void fraction as a function of the mass quality[†] of the mixture (χ) given by

$$\chi = \frac{m_g}{m_g + m_l} \quad (3.2)$$

where m_g and m_l are gas and liquid mass flow rates. Using Eq. 3.1, the mass velocities of the phases are given by

$$\text{gas phase} \quad G_g = \rho_g u_g \alpha \quad (3.3)$$

$$\text{liquid phase} \quad G_l = \rho_l u_l (1 - \alpha) \quad (3.4)$$

where u_g and u_l are the average velocities of the phases. A general equation for the void fraction can be obtained by substituting Eqs. 3.3 and 3.4 into Eq. 3.2;

$$\alpha = \left[1 + \frac{u_g}{u_l} \frac{(1-\chi)}{\chi} \frac{\rho_g}{\rho_l} \right]^{-1} \quad (3.5)$$

where (u_g/u_l) is the ratio of the average gas and liquid velocities or the slip ratio[†] S .

$$\alpha = \left[1 + S \frac{(1-\chi)}{\chi} \frac{\rho_g}{\rho_l} \right]^{-1} \quad (3.6)$$

3.2.2 Some Relevant Parameters

The average density of the two-phase mixture is defined as

$$\bar{\rho} = (1-\alpha) \rho_l + \alpha \rho_g \quad (3.7)$$

The superficial velocities or volumetric flux densities of the phases are defined as

$$J_g = u_g \alpha \quad (3.8)$$

$$J_l = u_l (1-\alpha) \quad (3.9)$$

The total volumetric flux density of the total superficial velocity of the mixture is given by

$$J = J_g + J_l \quad (3.10)$$

3.2.3 Pressure Losses

As in single-phase flow, the total pressure gradient in two-phase flow systems is composed of three components: a gravity term, an acceleration or momentum change term, and a friction term. The total pressure gradient is therefore defined as

$$\left(\frac{dP}{dx} \right)_{TP, tot} = \left(\frac{dP}{dx} \right)_{TP, fric} + \left(\frac{dP}{dx} \right)_{TP, acc} + \left(\frac{dP}{dx} \right)_{TP, gr} \quad (3.11)$$

The equations used to calculate these terms are obtained by applying the principle of conservation of momentum to an element along the axis of the channel. The following equations are derived for the calculation of the pressure terms in a channel with a constant cross-sectional area (Collier and Thome, 1994):

$$\left(\frac{dP}{dx}\right)_{TP,fric} = -\tau_w \frac{P_w}{A} \quad (3.12)$$

$$\left(\frac{dP}{dx}\right)_{TP,acc} = -G^2 \frac{d}{dx} \left[\frac{(1-\chi)^2}{(1-\alpha)\rho_l} + \frac{\chi^2}{\alpha\rho_g} \right] \quad (3.13)$$

$$\left(\frac{dP}{dx}\right)_{TP,gr} = -[(1-\alpha)\rho_l + \alpha\rho_g]g \quad (3.14)$$

Definitions of the variables used in the above equations are given in the nomenclature.

Gravitational and Acceleration Pressure Gradients

From Eqs. 3.12 and 3.13, the acceleration and the gravitation pressure gradient terms depend on the correlation used to predict the average void fraction (α). It is widely accepted that, in most cases, using the homogeneous equation for the average void fraction (Eq. 3.5 with $S = 1$) provides reasonable predictions of the accelerational and gravitational pressure gradients (Beattie, 1973).

Andeen and Griffith (1968) measured the momentum flux[†] through a pipe with steam–water and air–water two-phase flow systems. Their measurements were reasonably well correlated by the homogeneous model. The homogeneous model is presented in Section 3.4.

It should be noted that many of the process engineering systems involve steady-state the flow of the two-phase mixture in long horizontal conduits of constant cross-

sectional area and under adiabatic conditions. For these cases, the gravity term of the pressure gradient is zero (no change in elevation) and the acceleration term is negligible, since under steady-state adiabatic conditions there are no phase changes and thus no change of momentum⁵.

Frictional Pressure Gradient

Frictional losses were the subject of numerous experimental and theoretical two-phase flow studies for the following reasons:

- (i) the acceleration and gravity losses can be neglected in many cases, as discussed above;
- (ii) at relatively high flow rates, frictional is the dominant contributor to the total pressure loss;
- (iii) the frictional losses depend on the prevailing flow regime of the mixture.

Equation 3.12 shows that the frictional pressure gradient depends on the wall shear stress[†] (τ_w), the wetted perimeter of the channel (P_w), and the cross-sectional area of the flow channel (A).

Wall Shear Stress

It is well established in most common practices that the wall shear stress (τ_w) is defined in a manner consistent with that of single-phase flow studies, using the well-known Fanning equation

$$\tau_w = \frac{1}{2} f_k \rho_k u_k^2 \quad (3.15)$$

where the subscript k indicates either the gas or liquid phase and f_k is the friction factor that is commonly estimated using a standard single-phase flow relationship

⁵ Note that for adiabatic cases in which the pressure drop is significant compared to the average pressure, the acceleration term is not negligible.

$$f_k = C Re_k^m \quad (3.16)$$

with Re is the Reynolds number given by

$$Re_k = \frac{G_k d}{\mu_k} \quad (3.17)$$

In Eq. 3.16, $C = 16$ and $m = -1$ for laminar flow, and $C = 0.046$ and $m = -0.2$ for turbulent flow in smooth tubes. The values of C and m for turbulent flow are those suggested by Knudsen and Katz (1958), which are widely used in two-phase flow studies, e.g., Taitel and Dukler (1976b), Barnea (1986), Hetsroni (1982). These values are different from those suggested by Blasius (1913) which are widely recognized in single-phase flow studies ($C = 0.079$ and $m = -0.25$).

It should be noted that the implicit equation of Colebrook (1939), which allows for the roughness of the conduit wall, and its explicit forms suggested by Haland (1983) and Eck (1973) are often used to determine the two-phase flow friction coefficient.

Measurements of Liquid-wall Shear Stress

A number of experimental investigations were carried out to measure the liquid-wall shear stress, e.g., Hart et al. (1989), Grolman (1994), and Vlachos et al. (1997). Vlachos et al. (1997) measured the wetted wall fraction of the pipe and the liquid-wall shear stresses at various locations around a 24-mm horizontal pipe. Their experiment used an electrochemical solution as the liquid phase and covered mainly the stratified flow regime (liquid superficial velocity (J_l) varied from 0.01 to 0.05 m/s). From the results of these experiments, Vlachos et al. (1997) concluded that the time-averaged shear stress decreases in the lateral direction away from the bottom of the pipe. However, for relatively low gas superficial velocity, ($J_g < 15$ m/s), the measured shear stress is almost constant in the region where the liquid film is relatively thick, up to about a 45° angle.

Considering the low overall accuracy of the models used to predict the frictional pressure drop in two-phase flow systems, it can be argued that the effect due to variations in the wall shear stress around the pipe's circumference is considered to be modest and may be neglected.

Two-phase Friction Multipliers[†]

Some of the formulations used to predict friction losses in two-phase flow systems are closely related to the well-established formulations of single-phase flow. In fact, the two-phase flow frictional pressure gradient is often expressed in terms of that calculated for the gas phase or liquid phase based on the phase flowing alone in the channel. These expressions are termed frictional multipliers (ϕ_g and ϕ_l) and are defined as follows:

$$\phi_g^2 = (dP/dx)_{TP,fric} / (dP/dx)_{g,fric} \quad (3.18)$$

$$\phi_l^2 = (dP/dx)_{TP,fric} / (dP/dx)_{l,fric} \quad (3.19)$$

where $(dP/dz)_{l,fric}$ and $(dP/dz)_{g,fric}$ are pressure gradients for the liquid phase and gas phase, respectively, flowing alone in the conduit.

For boiling and condensing flows, where the mass flow rate of the phases continuously changes along the channel, the two-phase friction multipliers are defined as

$$\phi_{go}^2 = (dP/dx)_{TP,fric} / (dP/dx)_{go,fric} \quad (3.20)$$

$$\phi_{lo}^2 = (dP/dx)_{TP,fric} / (dP/dx)_{lo,fric} \quad (3.21)$$

where $(dP/dz)_{go,fric}$ and $(dP/dz)_{lo,fric}$ are the pressure gradients for the gas phase and liquid phase, respectively, flowing alone in the conduit at the total mass velocity of the mixture.

3.3 Generalized Models and Correlations

The models and correlations discussed in this section are those developed independently of the flow regime. Thus, their use does not require prior knowledge of the prevailing flow regime.

3.3.1 Homogeneous Flow Model

The homogeneous treatment of the flow assumes the phases are thoroughly mixed, homogeneously distributed throughout the conduit, in thermal equilibrium, and flowing at the same velocity. The model then considers the mixture to possess certain mean properties, defined from the properties of the phases, and abides by the usual equations of single-phase flow. This is the simplest approach to the problem.

Average Void Fraction

The homogeneous void fraction (α_h) can be obtained from Eq. 3.6 using the assumption that the phases are flowing at the same velocity, i.e., slip ratio equal to unity ($S = 1$)

$$\alpha_h = \left[1 + \frac{(1 - \chi) \rho_g}{\chi \rho_l} \right]^{-1} \quad (3.22)$$

The homogeneous void fraction is equivalent to the volumetric quality of the mixture (β).

Friction Losses

As indicated earlier, the homogeneous model treats the mixture as a single-phase flow with suitable averaged properties based on the properties of the phases. Therefore, the frictional pressure losses are calculated using the Fanning equation with suitable weighted average values of the mixture properties of density and viscosity.

The homogeneous density of the mixture is defined using Eq. 3.7. The friction factor is defined as a function of the two-phase flow Reynolds number, i.e., $Re_h = G d / \mu_h$, where μ_h is the mixture viscosity. A number of definitions were proposed for the calculation of the mean dynamic viscosity of the mixture. The most widely accepted formulations are the following:

$$\text{Owens et al. (1962)} \quad \mu_h = \mu_l \quad (3.23a)$$

$$\text{McAdams et al. (1942)} \quad 1/\mu_h = x/\mu_g + (1-x)/\mu_l \quad (3.23b)$$

$$\text{Cicchitti et al. (1960)} \quad \mu_h = x \mu_g + (1-x) \mu_l \quad (3.23c)$$

$$\text{Dukler et al. (1964)} \quad \mu_h = \alpha_h \mu_g + (1 - \alpha_h) \mu_l \quad (3.23d)$$

$$\text{Beattie and Whalley (1982)} \quad \mu_h = \alpha_h \mu_g + (1 - \alpha_h) (1 + 2.5 \alpha_h) \mu_l \quad (3.23e)$$

Figure 3.1 shows the calculated mean viscosity using the above five formulations for an air–water mixture at standard atmospheric conditions (101.3 kPa and 20 °C). The air and water properties at standard atmospheric conditions are $\mu_g = 1.8 \times 10^{-5}$ Pa·s, $\mu_l = 1.05 \times 10^{-3}$ Pa·s, $\rho_g = 1.2$ kg/m³, and $\rho_l = 998$ kg/m³. The formulations proposed by Dukler et al. (1964) and Beattie and Whalley (1982) predict a lower mean viscosity for the mixture than the other formulations. Both of these formulations are given as a function of the mixture volumetric flow fraction (β), which is equivalent to the homogeneous void fraction (α_h) given by Eq. 3.22. It is interesting to note that at low mass quality of the mixture ($\chi < 10^{-3}$), the definition suggested by Beattie and Whalley (1982) predicts a mean viscosity greater than that of the water. The definition proposed by McAdams et al. (1942) predicts a mean mixture viscosity slightly higher than that predicted by the definitions proposed by Dukler et al. (1964) and Beattie and Whalley (1982).

The definition proposed by Cicchitti et al. (1960) is similar to that of Owens et al. (1962), except the mixture viscosity approaches that of the gas phase when the mass quality approaches unity. Both of these definitions assume that the mean viscosity of the mixture is more dependent on the viscosity of the water since in most two-phase flow

patterns (bubble, slug, plug and annular flows) the liquid phase covers most of the wall of the conduit.

The effect of the definition of the mean viscosity of the mixture on the calculated friction losses using the homogeneous model is discussed in Chapter 6.

Discussion

The homogeneous model is considered too simplistic and is often used only to provide first-cut estimates of the frictional pressure drop in the system. However, under certain flow conditions, specifically when the two phases are thoroughly mixed as with dispersed bubble flow regime, the homogeneous model can provide reasonable predictions. Whalley (1987) indicated that the homogeneous model provides good results if the liquid to gas density ratio is less than 10 or if the mixture mass velocity is higher than 2000 kg/m²·s.

3.3.2 Lockhart–Martinelli and Related Correlations

Lockhart and Martinelli (1949) developed the most widely used correlation for the calculations of friction losses in low pressure two-phase flow systems. This correlation is still being used by many investigators to correlate their experimental data, e.g., Dowlati et al. (1995), Xin et al. (1996), and Wang et al. (1999).

Lockhart and Martinelli (1949) related the two-phase flow friction multipliers ϕ_l and ϕ_g , given by Eqs. 3.18 and 3.19, to the parameter X defined as follows:

$$X^2 = (dP/dx)_{l,fric} / (dP/dx)_{g,fric} \quad (3.24)$$

The relationship between the friction multipliers and the Lockhart–Martinelli parameter X was determined based on experimental data for isothermal two-component[†] flow in horizontal pipes at near atmospheric conditions. The correlation was presented in graphical form as shown in Fig. 3.2. Different curves are shown for the flow condition of

each of the phases: laminar (ν) or turbulent flow (t). Figure 3.2 also shows the empirical correlation suggested by Lockhart and Martinelli (1949) for the calculation of the average void fraction. This correlation is presented at the end of this section.

The Lockhart–Martinelli parameter (X) can be written as a function of the properties of the phases and the mass quality of the mixture (Eq. 3.25a) or the superficial velocities of the phases (Eq. 3.25b). Assuming both phases are turbulent and using Eqs. 3.12 and 3.15 to 17, X can be written as

$$X_{tt} = \left(\frac{\rho_g}{\rho_l} \right)^{0.5} \left(\frac{1 - \chi}{\chi} \right)^{0.9} \left(\frac{\mu_l}{\mu_g} \right)^{0.1} \quad (3.25a)$$

$$X_{tt} = \left(\frac{\rho_l}{\rho_g} \right)^{0.4} \left(\frac{J_l}{J_g} \right)^{0.9} \left(\frac{\mu_l}{\mu_g} \right)^{0.1} \quad (3.25b)$$

The subscript term “ tt ” in the above equations indicates that both phases are assumed to be under turbulent conditions.

Although the original correlation of Lockhart and Martinelli (1949) was entirely empirical, it was later given some theoretical justification by a number of researchers, e.g., Chisholm (1967) and Johannessen (1972). Using the separate-cylinders⁶ model and assuming that the pressure drop is the same for the segregated phases regardless of the flow pattern, Chisholm (1967) provided the following general representation of the graphical relationships of the two-phase flow friction multipliers of Lockhart and Martinelli (1949).

$$\phi_l^2 = 1 + C/X + 1/X^2 \quad (3.26a)$$

$$\phi_g^2 = 1 + CX + X^2 \quad (3.26b)$$

where X is defined in Eq. 3.24, and C is a dimensionless parameter whose value depends on the viscous or turbulent nature of each phase. Chisholm (1967) showed that

⁶ The separate-cylinders model is based on the assumption that the phases are separated by one continuous boundary and flow with a certain relative velocity.

the Lockhart–Martinelli curves shown in Fig. 3.2 are well represented by the values of C given in Table 3.1.

Table 3.1: Values of the parameter C in Eq. 3.26a and Eq. 3.26b.

Liquid Phase	Gas Phase	Parameter “C” In Eq. 3.18
Turbulent	Turbulent	20
Laminar	Turbulent	12
Turbulent	Laminar	10
Laminar	Laminar	5

Martinelli–Nelson Correlation

Martinelli and Nelson (1948) noted that the Lockhart–Martinelli correlation failed to predict the friction losses in high-pressure systems. Based on experimental data taken for steam-water mixtures over a wide range of system pressures, Martinelli and Nelson (1948) produced a revised correlation for friction losses in high-pressure systems. The correlation was given in terms of the friction multiplier ϕ_o as a function of the mixture quality for different system pressure levels (Fig. 3.3). Note that the friction multiplier is calculated based on the total mass flow rate of the mixture.

The Martinelli–Nelson correlation is used mainly to predict friction losses in high-pressure boiling flows. Since in boiling flows the mass quality of the mixture changes along the flow channel, Martinelli and Nelson (1948) integrated their correlation, which provides local values of ϕ_o , over the length of the test section. The results are shown in Fig. 3.4, which gives the integrated value of ϕ_o as a function of the average system pressure and the mass quality of the mixture at the end of the test section.

Discussion

Over the years, a number of other modifications of the Lockhart–Martinelli correlation were suggested. However, the original correlation, in the form suggested by

Chisholm (1967), is perhaps the most widely used as it is practical and easy to use. It is still being used as the basis of a number of new formulations of two-phase flow friction losses. Xin et al. (1996) used the Lockhart–Martinelli correlation to correlate their measurements of friction losses in vertical helicoidal pipes. Wang et al. (1999) extended the Lockhart–Martinelli correlations to predict the pressure drop across plate heat exchangers. Their experimental results were well correlated with $C = 16$. Mishima et al. (1996) carried out experimental investigations of friction losses in a vertical rectangular test section. Their data were well correlated with the Lockhart–Martinelli correlation. Mishima et al. gave an expression for the parameter C as a function of the hydraulic diameter of the channel. For pipe diameters > 10 mm, their expression gives a value of C of about 20.

The major shortcoming of the Lockhart–Martinelli and Martinelli–Nelson correlations is their failure to account for the effect of total mass velocity of the mixture, as shown by Baroczy (1966), Cicchitti et al. (1960), Chisholm (1968), and Friedel (1979). The experimental data of Cicchitti et al. (1960) showed a significant effect of the mass velocity on the friction multipliers. Cicchitti et al. (1960) also noted that at relatively high mass velocity, the homogeneous model provided better predictions than the Martinelli–Nelson correlation. The empirical correlations of Baroczy (1966), Chisholm (1968), and Friedel (1979) are presented in the following section.

Average Void Fraction

As indicated earlier, Lockhart and Martinelli (1949) provide a graphical correlation for the average void fraction (Fig. 3.2). This correlation is well represented by the following equation (Wallis, 1969):

$$\alpha = \frac{1}{(1 + X_{tt}^{0.8})^{0.378}} \quad (3.27)$$

where X_{tt} is the Lockhart–Martinelli parameter for turbulent liquid and turbulent gas flow, given by Eq. 3.25.

A similar expression for the void fraction can be developed from the separate-cylinders model as

$$\alpha = 1 - \frac{1}{(\phi^2)^{1/n}} \quad (3.28)$$

where ϕ is the two-phase flow friction multiplier defined by Eq. 3.26a and n is given by $n=(5-m)/2$, where m is the exponent in the Reynolds number in Eq. 3.16, (Rogers, 1989). For turbulent liquid flow in a smooth pipe $m = 0.2$, the average void fraction calculated using the separate-cylinders model is given by

$$\alpha = 1 - \frac{1}{\phi^{0.833}} \quad (3.29)$$

3.3.3 Empirical Correlations

A large number of correlations were proposed for predicting the two-phase flow friction losses. Usually these correlations are obtained by curve fitting experimental data without reference to any particular flow regime. The main disadvantage of this approach is that the resulting correlation is strongly dependent on the composition of the data used and provides no insight into the mechanisms affecting the friction losses. To overcome this difficulty, a number of correlations were based on data taken over a wide range of flow conditions. Other investigators suggested the use of correction factors to account for variations in the properties of the mixture.

The following are some of the correlations most referred to in the literature. Govier and Aziz (1972) provided a detailed review of most of the earlier empirical correlations.

3.3.3.1 Baroczy Correlation for Friction Losses

The Baroczy (1966) correlation was intended to overcome the deficiency of the Lockhart–Martinelli and the Martinelli–Nelson correlations, which is their inability to

incorporate the mass velocity effects on the frictional pressure drop. The Baroczy correlation consists of two sets of curves: the first is a plot of the friction multiplier ϕ_o as a function of a physical property index $[(\mu/\mu_g)^{0.2} (\rho_g/\rho_l)]$ for a reference mass velocity (Fig. 3.5); and the second provides a correcting factor for different mass velocities of the mixture (Fig. 3.6). The correlation is suggested for use with a wide variety of liquids, including sodium, mercury, and Freon-22. This correlation is difficult to use because it is entirely graphical. The error associated with using this correlation may be large, mainly because interpolating between the meandering curves is difficult.

3.3.3.2 Chisholm Method for Friction Losses

Chisholm (1968) expanded on the expression he suggested for the Lockhart–Martinelli correlation (Eq. 3.18) and proposed a complex expression for parameter C in Eq. 3.26a. This expression includes the influence of the mass velocity of the mixture, the system pressure, and the roughness of the conduit wall surface on the two-phase flow friction multiplier (ϕ^2). The Chisholm method is usually used for steam–water flows (Collier et al., 1994). Whalley (1987) indicated that the Chisholm method is not particularly accurate. Therefore, this correlation is not considered in this study.

3.3.3.3 Friedel's Correlation for Friction Losses

Friedel (1979) developed what has been considered as the most accurate correlation for predicting pressure losses in two-phase systems. It is considered to be an improvement over the correlations of Baroczy (1966) and Chisholm (1968). Using a large database of two-phase pressure drop measurements, Friedel (1979) derived an expression for the friction multiplier (ϕ_o), and thus the friction losses. The correlation proposed by Friedel (1979) is presented in Appendix A.

Friedel's (1979) correlation is applicable for vertical up-flow as well as horizontal flow. A separate correlation was provided for vertical downward flows.

3.3.3.4 CISE[†] Correlation for Average Void Fraction

Premoli et al. (1970) suggested a correlation for the slip ratio S and thus the average cross-sectional void fraction. This correlation, known as the CISE correlation, is given in Whalley (1980).

The CISE Correlation is presented in Appendix A.

3.4 Flow Pattern-Dependent Correlations

Dependence of the average void fraction and pressure drop on flow patterns was recognized and demonstrated by many investigators, e.g., Baker (1961), Simpson et al. (1977), and Kokal et al. (1989). Baker (1961) suggested a separate correlation for each of the flow regimes. Simpson et al. (1977) studied the flow of air–water mixtures in large-diameter horizontal pipes (127 and 216 mm) at near atmospheric conditions; they performed ~1000 tests. Their measurements of the void fraction and pressure drop showed a strong flow pattern dependency. Kokal et al. (1989) conducted a series of oil–water experiments in a 25-m long test section. They found that the pressure drop and liquid hold-up are flow-pattern dependent.

The models and correlations presented in this section were developed for specific flow patterns. Therefore their use requires the proper identification of the prevailing flow pattern.

3.4.1 Drift-flux Model for Average Void Fraction

The drift-flux⁷ model, developed principally by Zuber and Findlay (1965), is a separated-flow model with a special focus on the relative motion between the phases rather than on their individual motion. The model was mainly used in vertical two-phase flows, e.g., Wallis (1969), Ishii et al. (1975, 1977), and Mishima et al. (1993). However, França and Lahey (1992) showed that it could be used to correlate void fractions in

⁷ Note that Collier et al. (1994) considered the drift-flux model as an empirical correlation, since C_o and V_{gj} parameters are usually determined experimentally.

horizontal flows. The results of the work of França and Lahey (1992) are presented and discussed at the end of this section.

Detailed description of the drift-flux model is given in Wallis (1969); only the essential relationships of the model are presented here. Zuber and Findlay (1965) introduced an expression for the average void fraction, which takes into account the relative velocity between the phases and the distribution of the phases over the cross-sectional area of the conduit. The expression implies a linear relationship between the average gas velocity and total volumetric flux of the mixture. The relationship is given as

$$\langle u_g \rangle_g = \frac{\langle J_g \rangle}{\langle \alpha \rangle} = C_o \langle J \rangle + V_{gj} \quad (3.30)$$

where $\langle u_g \rangle_g$ is the gas velocity averaged over the gas flow area, $\langle J_g \rangle$ is the superficial gas velocity, and $\langle J \rangle$ is the average volumetric flux of the mixture, which is the sum of the gas and liquid superficial velocities, i.e., $\langle J \rangle = \langle J_g \rangle + \langle J_l \rangle$. The drift-flux model parameters C_o and V_{gj} are defined by the following:

$$C_o = \frac{\langle \alpha J \rangle}{\langle \alpha \rangle \langle J \rangle} \quad (3.31)$$

$$V_{gj} = \frac{\langle \alpha (u_g - J) \rangle}{\langle \alpha \rangle} \quad (3.32)$$

In the above equations, the angle brackets indicate that the parameters are averaged over the cross-sectional area of the tube. C_o is the distribution parameter and V_{gj} is the weighted-mean drift velocity. These drift-flux parameters are usually determined experimentally by plotting $\langle J_g \rangle / \langle \alpha \rangle$ against $\langle J \rangle$. C_o is the slope of the linear regression of the data, and V_{gj} is the intercept of the regression line with the $\langle J_g \rangle / \langle \alpha \rangle$ axis.

The distribution coefficient C_o depends mainly on the void fraction and volumetric flux (J) profiles along the pipe diameter. Assuming axisymmetric profiles of the volumetric flux and the void fraction, Zuber and Findlay (1965) showed that, for flow

in vertical conduits, C_o is usually less than unity when the void fraction at the centre on the conduit is less than the void fraction near the wall. C_o is equal to unity when the gas phase is equally distributed over the channel's cross-sectional area. $C_o > 1$ when the void fraction near the centre on the conduit is higher than the void fraction near the wall of the conduit. These variations of C_o with the void fraction and the volumetric flux profiles cannot be extended to horizontal flow regimes because of the non-axisymmetric nature of the flow. The literature surveyed does not provide any information on this issue.

The weighted mean drift velocity, V_{gj} , takes into account the average relative velocity of the gas phase to the average volumetric flux of the two phases. In vertical flow, the drift velocity is governed mainly by balance between the buoyancy and drag forces. Earlier correlations of horizontal flow data using the drift-flux model assumed that since the buoyant forces can be neglected in horizontal flows, V_{gj} can be set equal to zero. However, as noted by França and Lahey (1992), there is a drift flux velocity in horizontal flows that is a result of a balance between momentum and drag forces (observed slip between the phases).

Ishii (1977) provided expressions to calculate C_o and V_{gj} for each of the flow patterns in vertical channels. The correlations of França and Lahey (1992) for horizontal flows are discussed in the following section.

Horizontal Flows:

França and Lahey (1992) found that the drift-flux expression, given in Eq. 3.30, correlates very well their measurements of the average void fraction for plug and slug flow regimes. However, for annular and stratified wavy flow regimes, they indicated that their experimental data are more appropriately correlated by the $\langle \alpha \rangle / (1 - \langle \alpha \rangle)$ and $\langle J_g \rangle / \langle J_l \rangle$ set of variables. França and Lahey (1992) argued that the ratios of $\langle J_g \rangle / \langle J_l \rangle$ in annular and stratified wavy flow are so high that the graphical method for determining C_o and V_{gj} does not show sufficient sensitivity to changes in $\langle J_l \rangle$. This does not agree with the results of most of the flow regime maps presented in Chapter 2, which show that slug flow could occur at $\langle J_g \rangle / \langle J_l \rangle$ ratios as high as those calculated for annular and stratified

wavy flow regimes. For example, the flow regime map of Mandhane et al. (1974) (Fig. 2.8), predicts annular flow occurs at $J_g = 30$ m/s and $J_l = 0.5$ m/s, which gives a $\langle J_g \rangle / \langle J_l \rangle$ ratio of 60. The same map predicts that slug flow occurs at $J_g = 8$ m/s and $J_l = 0.1$ m/s, which gives a $\langle J_g \rangle / \langle J_l \rangle$ ratio equal to 80.

The values of the drift-flux parameters recommended by França and Lahey (1992) for horizontal flow are given Table 3.2.

Table 3.2: Values of the drift-flux model parameters for horizontal flow suggested by França and Lahey (1992).

Flow Regime	C_o	V_{ei} (m/s)
Plug	$\cong 1.0$	0.16
Slug	1.2	-0.20
Stratified wavy and annular	$\cong 1.0$	0.2 for $\langle J_l \rangle = 0.005$ m/s 2.7 for $\langle J_l \rangle = 0.27$ m/s

Correlation of the Data of França and Lahey (1992) using the Standard Drift-Flux Variables

França and Lahey (1992) included the experimental data with their published results. Therefore, it is possible to correlate their data using the standard variables of the drift-flux model, $\langle J_g \rangle / \langle \alpha \rangle$ and $\langle J \rangle$. The results of this correlation are shown in Fig. 3.7 and the calculated values of C_o and V_{gi} are given in Table 3.3.

Table 3.3: Results of the correlation of the data of França and Lahey (1992) with the standard drift-flux variables.

Flow Regime	C_o	V_{ei} (m/s)
Plug	1.01	0.145
Slug	1.22	-0.218
Stratified wavy	1.01	0.756
Annular	1.01	1.790

Table 3.3 shows that the values C_o and V_{gj} for the slug and plug flow regimes are consistent with those determined by França and Lahey (1992) (Table 3.2). The observed small variations may be explained by the fact that different tools were used to perform the least-squares fit of the data. In the present study, Microsoft Excel-97 SR-1 © is used to perform all regression analysis.

Figure 3.7 shows that the experimental data of França and Lahey (1992) for annular and stratified wavy flow regimes can be correlated using the standard variables of the drift-flux model: a strong linear dependency between the variables is observed for the regimes. The calculated coefficient of determination (R^2) is better than 0.99 for all of flow patterns. The results of this correlation are discussed further and compared with the results of the present study in Chapter 6.

3.4.2 Friction Losses Models and Correlations

The models and correlations that were developed based on the specific-configuration of flow patterns are classified as mechanistic⁸ models and empirical correlations.

(a) Mechanistic Models

The flow pattern-dependent mechanistic models are listed in Tables 3.4 to 3.7. These models are generally based on an idealized representation of the flow pattern. These models rely on experimental data.

⁸ Mechanistic models are also referred to as “phenomenological” models.

Table 3.4: Models developed for stratified smooth and wavy flow regimes.

Regime-specific models	Comments
Agrawal et al. (1973)	<ul style="list-style-type: none"> • developed for stratified smooth flow regime • based on extension of the same velocity profile the gas and liquid regions • requires iterative procedure • predicts lower values of the two-phase friction multiplier than the Lockhart–Martinelli model
Johannessen (1972)	<ul style="list-style-type: none"> • developed based on the Lockhart–Martinelli model • predicts lower values of the two-phase friction multiplier than the Lockhart–Martinelli model
Taitel and Dukler (1976-b)	<ul style="list-style-type: none"> • based on simple theoretical considerations of the flow • assumed that the interfacial friction factor (f_i) is constant and is equal to the gas-wall friction factor (f_g) • showed that the frictional pressure drop is a function of the Lockhart–Martinelli parameter only
Parsen (1984)	<ul style="list-style-type: none"> • developed for stratified flow with no interactions between the phases
Hashizume et al.(1985 / 87) Note: This model is discussed further at the end of this section.	<ul style="list-style-type: none"> • assumed that velocity profiles for gas and liquid phases are described with the Prandtl mixing length theory • developed simplified models for stratified smooth and annular flow regimes • for wavy flow; suggested to interpolate between values calculated for stratified smooth and annular regimes, at the transition based on a modified flow regime map of Baker; • original model tested against refrigerant data • later (1987) tested model against air–water and steam–water data (stratified, wavy, slug and annular flow regimes) – plug flow was not considered • for wavy used linear interpolation between stratified and annular flow model, and for slug flow used annular flow model
Hart et al. (1989)	<ul style="list-style-type: none"> • developed for horizontal and slightly inclined conduits; • identified as Apparent Rough Surface (ARS) model • suggested for wavy and annular flow with low liquid mass velocities (low liquid-holdup; $(1-\alpha) < 6\%$) • requires iterative procedure
Grolman (1994)*	<ul style="list-style-type: none"> • Modified ARS (MARS) model • covers a wider range of flow conditions than the ARS model; ($(1-\alpha) < 42\%$) • requires iterative procedure • for horizontal and slightly inclined flows (-3° to $+6^\circ$)

Table 3.5: Models developed for annular flow regime.

Regime-specific models	Comments
Kadambi (1985)	<ul style="list-style-type: none"> assumed uniform film thickness around the tube, initially neglected the liquid entrained in the gas core, and assumed matching conditions of the phases at the interface model requires information of the interfacial friction factor; compared prediction with experimental data in vertical and horizontal flows liquid entrained in the core is accounted for using empirical correlations good agreement obtained with measured pressure drop in vertical flow indicated that better predictions of interfacial friction factor and liquid entrainment are required for the model to provide reasonable predictions for flow in horizontal conduits
Hashizume and Ogiwara (1985)	<ul style="list-style-type: none"> same comments as in Table 3.4
Hart et al. (1989)	<ul style="list-style-type: none"> same comments as in Table 3.4
Grolman (1994)	<ul style="list-style-type: none"> same comment as in Table 3.4

Table 3.6: Models developed for plug and slug flow regimes.

Regime-specific models	Comments
Dukler and Hubbard (1975)	<ul style="list-style-type: none"> modelled the flow as a repeating liquid slugs with trailing liquid film model includes details, such as liquid pickup at the front of the liquid slugs and liquid shedding at the back of the liquid slug model assumes zero drift velocity used experimental data to approximate the average length of the liquid slugs
Nicholson et al. (1978)	<ul style="list-style-type: none"> proposed two models: (i) modified the model of Dukler and Hubbard (1975); and (ii) slightly different model which assumes a constant length of liquid slugs equal to $30d$ model requires the use of empirical correlations to estimate the slug frequency and the liquid holdup requires iterations
Kokal & Stanislav (1989)	<ul style="list-style-type: none"> developed for flow in horizontal and slightly inclined pipes; used drift-flux model with values of C_o and V_{ei} proposed by Zuber and Findlay (1965) for vertical flows

Table 3. 7: Models developed for dispersed bubbly flow regime.

Regime-specific models	Comments
Nicholson et al. (1978)	<ul style="list-style-type: none"> extended use of their intermittent flow model to cover the transition into dispersed flow regime

(b) Empirical Correlations

This section presents the empirical correlations which were also suggested for predicting the friction losses in each of the horizontal flow regimes.

Correlations of Baker (1961)

Baker (1961) provided separate correlations for annular, plug, slug and stratified flow regimes (Eqs. 3.33a to 3.33d). The correlations, given in Govier et al. (1972), were developed for large diameter pipes from the Lockhart–Martinelli correlation. However, Baker (1961) showed that his correlations gave reasonable prediction of friction losses in 25 mm pipes.

The correlations were given in terms the two-phase flow friction multiplier ϕ_g as a function of the Lockhart–Martinelli parameter (X) and the mass velocity of the liquid phase G_l in ($\text{lb}_m/\text{ft}^2\cdot\text{h}$). The correlation proposed for the annular flow is given as a function of X and the pipe diameter d (units in ft). The gas and liquid phases are assumed to be under turbulent conditions.

$$\text{stratified flow:} \quad \phi_g = \frac{15400 X}{G_l^{0.8}} \quad (3.33a)$$

$$\text{elongated bubble (plug) flow:} \quad \phi_g = \frac{27.315 X^{0.855}}{G_l^{0.17}} \quad (3.33b)$$

$$\text{dispersed bubble flow:} \quad \phi_g = \frac{14.2 X^{0.75}}{G_l^{0.1}} \quad (3.33c)$$

$$\text{slug flow:} \quad \phi_g = \frac{1190 X^{0.815}}{G_l^{0.5}} \quad (3.33d)$$

annular flow:
$$\phi_g = (4.8 - 0.3125 d) X^{0.343 - 0.021 d} \quad (3.33e)$$

For wavy flow and slug flow regimes in small diameter pipes (25 mm and smaller) Baker suggested the use of the Lockhart–Martinelli correlation.

Correlations of Beattie (1973)

Beattie (1973) argued that the friction losses in two-phase flow can be calculated in a manner consistent with that of that of single-phase flow provided that the friction factor (f) and the Reynolds number (Re) are appropriately defined for the particular type of two-phase flow. Beattie (1973) developed separate expressions for ϕ_o as a function of the properties of the phases and the mass quality of the mixture for a number of flow regimes depending on the distribution of the phases near the wall of the pipe. The conditions of the flow at the wall of the conduit and examples of flow regimes considered by Beattie as follows:

(i) Bubble (i.e., bubble flow)

$$\phi_o^2 = \left\{ 1 + \chi \left(\frac{\rho_l}{\rho_g} - 1 \right) \right\}^{0.8} \left\{ 1 + \chi \left(\frac{(3.5 \mu_g + 2 \mu) \rho_l}{(\mu_g + \mu) \rho_g} - 1 \right) \right\}^{0.2} \quad (3.34a)$$

(ii) Wavy gas–liquid interface (i.e., annular flow with no entrainment)

$$\phi_o^2 = \left\{ 1 + \chi \left(\frac{\rho_l}{\rho_g} - 1 \right) \right\}^{0.8} \left\{ 1 + \chi \left(\frac{\mu_g \rho_l}{\mu \rho_g} - 1 \right) \right\}^{0.2} \quad (3.34b)$$

(iii) Very small bubbles (i.e., bubble and annular flow)

$$\phi_o^2 = \left\{ 1 + \chi \left(\frac{\rho_l}{\rho_g} - 1 \right) \right\}^{0.8} \left\{ 1 + \chi \left(\frac{3.5 \rho_l}{\rho_g} - 1 \right) \right\}^{0.2} \quad (3.34c)$$

(iv) Dry wall (i.e., post burnout flow)

$$\phi_o^2 = \left(\frac{\mu_g}{\mu} \right)^{0.2} \left(\frac{\rho_g}{\rho_l} \right)^{0.8} \left\{ 1 + \chi \left(\frac{\rho_l}{\rho_g} - 1 \right) \right\}^{1.8} \quad (3.34d)$$

The above expressions proposed by Beattie (1973) are based on simplifying assumptions about the distribution of the phases.

The major difficulty in using the above expressions is the fact that the flow regime classification considered by Beattie (1973) is not consistent with the classification discussed in Chapter 2. It is therefore difficult to compare it with the different flow patterns observed.

Correlations of Bandel et al. (1973)

Bandel (1973) developed empirical correlations for friction losses in stratified smooth and annular flow regimes. Bandel's (1993) correlations are given in Appendix A. For a given two-phase flow system, the pressure drop is first calculated using the annular flow correlation. If the calculated pressure gradient is higher than the minimum pressure gradient for annular flow $\left(\frac{dP}{dx}\right)_{A,min}$ (given by Eq. 3.35a) the flow is considered annular; if not, the pressure drop is calculated using the stratified smooth flow correlation. The calculated pressure is then compared against the upper limiting pressure gradient of the stratified smooth flow $\left(\frac{dP}{dx}\right)_{St,max}$ (given by Eq. 3.35b). If the calculated pressure gradient is lower than $\left(\frac{dP}{dx}\right)_{St,max}$, the pressure gradient is correctly predicted by the stratified smooth flow correlation; if not, the flow is considered to be a transitional flow and the pressure gradient is calculated by interpolating between the stratified and the annular regimes.

The limiting pressure gradients are calculated using the following equations. Figure 3.8 shows the limiting pressure gradients calculated for air and water at standard temperature and pressure plotted as a function of mass quality of the mixture (χ).

$$\left(\frac{dP}{dx}\right)_{A,min} = \frac{g \rho_l}{10 \chi^{0.3}} \quad (3.35a)$$

$$\left(\frac{dP}{dx}\right)_{St,max} = \frac{g \rho_l \chi}{15 \left(\frac{\mu_l}{\mu_g}\right)^{0.2}} \quad (3.35b)$$

Bandel's (1973) correlations do not require prior knowledge of the prevailing flow regime. It can therefore be argued that these correlations should be considered with the generalized correlations considered earlier in this chapter. However, since the correlations were developed for specific flow patterns, they are categorized in this study as flow-pattern dependent correlations.

Hashizume et al. (1985/87)

Hashizume and Ogiwara (1985) developed two separate mechanistic models for calculating the frictional pressure drop in the stratified and annular flow regimes. For wavy and intermittent flow regimes, Hashizume and Ogiwara (1985) suggested that the friction losses be calculated by interpolating between the stratified and annular flow, using the interpolation method of Bandel (1973). Hashizume and Ogiwara (1985) indicated that their models showed good agreement with experimental data for refrigerants. Later Hashizume and Ogawa (1987) compared the predictions of this model against 2300 experimental data points from 23 data sets for air–water and steam–water flows in horizontal pipes (data were taken from the HTFS⁹ data bank). The data cover different mass velocity, mass quality, and pipe diameter, and wall roughness. Hashizume and Ogawa (1987) used the annular flow model to calculate the friction losses in slug flow, and interpolated for the friction losses in wavy flow. The composition of the data is as follows: 79% annular flow, 15% wavy flow, 3% stratified flow, and 3% slug flow. The regime classification was based on their modification of the horizontal flow regime map of Baker (1964).

⁹ HTFS (Heat Transfer and Fluid Flow Service, England) data bank.

The results of this comparison showed that about two-thirds of the experimental readings are predicted within $\pm 30\%$ and concluded that the model also applies to air–water and steam–water systems. Hashizume and Ogawa (1987) concluded that their model gave the best agreement with the refrigerant data, while the model of Bandel (1973) gave better agreement with the air–water and steam–water data.

◆ *Discussion*

Tribbe and Müller-Steinhagen (2000) evaluated the predictions of a number of mechanistic models against a large experimental data bank containing over 7000 measurements of frictional pressure drop in horizontal pipes. The data bank¹⁰ comprises different fluid systems (air–water, steam–water and different refrigerants) and covers a wide range of flow conditions. The mechanistic models considered by Tribbe and Müller-Steinhagen (2000) are listed in Table 3.3 with the exception of the models of Kadambi (1985), Kokal and Stanislav (1989), and Grolman (1994).

Tribbe and Müller-Steinhagen (2000) concluded that the performance of the mechanistic models is comparable to that of the well-known empirical methods, such as Lockhart and Martinelli (1949), Bandel (1973) and Friedel (1979). The mechanistic models were found to be less influenced by changes of fluid system. Based on the results of this comparison, Tribbe and Müller-Steinhagen (2000) recommended the models listed in Table 3.8 for predicting the friction losses in horizontal pipes.

Table 3.8: Mechanistic models suggested by Tribbe and Müller-Steinhagen (2000).

Flow Regime	Proposed Method(s)
Stratified flow	Agrawal et al. (1973)
Wavy flow	Hashizume and Ogiwara (1985) and Hart et al. (1989)
Annular flow	Hashizume and Ogiwara (1985)
Plug and slug (intermittent) flow	Nicholson et al. (1978)
Dispersed bubble flow	Nicholson et al. (1978)

¹⁰ Identified as TVT-Dukler data bank.

Hashizume and Ogawa (1987) indicated that the models of Bandel (1973) gave better predictions of the friction losses in air–water systems than the models of Hashizume and Ogiwara (1985). The models of Hart et al. (1989) are developed for low liquid hold-up (< 6%). The model Nicholson et al. (1978) uses drift-flux parameters developed for vertical flow and assumes constant slug frequency. As will be discussed in Chapter 5, none of the test runs of the present study are classified as stratified smooth or dispersed bubble flow. The models listed Table 3.8 are not considered in the present study.

3.5 Two-Fluid[†] Model

The two-fluid model is considered to be the most accurate two-phase flow formulation because of its detailed treatment of the distribution of the phases and their interactions at the interface. In this model, the phases are modeled separately, with mass, momentum and energy exchanged through the interfacial area separating the phases. Conservation equations are written for each phase, and interactions between the phases are accounted for by constitutive relationships. The model is valid for any two-phase flow pattern, if appropriate constitutive equations for all the wall-to-fluid and interfacial transfer terms are provided.

The most common approach used in developing the one-dimensional two-fluid model is to start with the momentum equations for an idealized representation of the two-phase flow configuration. For steady state, fully developed incompressible gas–liquid flow in a horizontal pipe with a constant cross-sectional area, the one-dimensional¹¹ momentum equation for each of the phases is given by

$$\text{gas phase:} \quad -\langle \alpha \rangle A \left(\frac{dP}{dx} \right)_{TP} - \tau_{wg} P_{wg} - \tau_i P_i = 0 \quad (3.36a)$$

¹¹ Two-dimensional flow models were developed by a number of investigators, such as Shoham and Taitel (1986), Issa (1988) and Hall (1992). These models are rarely used in most practical applications; most state-of-the-art codes, e.g., CATHENA and TUF, are one-dimensional codes.

$$\text{liquid phase:} \quad -(1 - \langle \alpha \rangle) A \left(\frac{dP}{dx} \right)_{TP} - \tau_{wl} P_{wl} + \tau_i P_i = 0 \quad (3.36b)$$

The above equations represent the basic momentum equations for a simplified one-dimensional approach, which assumes that the pressure drop in the gas and liquid phases is the same. Similar equations are derived in Taitel and Dukler (1976b) and Collier and Thome (1994).

In the above equations, $\langle \alpha \rangle$ is the average void fraction and A is the total flow area of the channel. τ_{wl} , τ_{wg} and τ_i are the wall-liquid, wall-gas and gas-liquid interfacial shear stresses, respectively. P_{wl} , P_{wg} and P_i are the wall-liquid, wall-gas and gas-liquid contact perimeters over which the shear stresses act. The contact perimeters are defined based on the disposition of the phases.

Most studies assume that the shear stresses can be evaluated using the following conventional single-phase flow formulations (Fanning relationship):

$$\tau_{wg} = \frac{1}{2} f_g \rho_g u_g^2 \quad (3.37a)$$

$$\tau_{wl} = \frac{1}{2} f_l \rho_l u_l^2 \quad (3.37b)$$

$$\tau_i = \frac{1}{2} f_i \rho_g (u_g - u_l)^2 \quad (3.37c)$$

where f_g , f_l , and f_i are the gas, liquid, and interfacial friction factors, respectively. The friction factors are evaluated using Eq. 3.16

$$f_g = C \left(\frac{\rho_g u_g d_{eg}}{\mu_g} \right)^m \quad (3.38a)$$

$$f_l = C \left(\frac{\rho_l u_l d_{el}}{\mu_l} \right)^m \quad (3.38b)$$

$$f_i = C \left(\frac{\rho_g |u_g - u_l| d_{eg}}{\mu_g} \right)^m \quad (3.38c)$$

In the above equations, the terms inside the parentheses represent the gas, liquid and interfacial Reynolds numbers. In Eqs. 3.38a and 3.38b, the gas and liquid Reynolds numbers (Re_g and Re_l) are evaluated based on the properties and the actual velocities of the phases. The interfacial Reynolds number (Re_i) and interfacial shear stress (τ_i) are calculated based on the relative velocities between the phases and the properties of the gas phase, since the gas velocity is almost always higher than the liquid velocity. Thus the situation is similar to that of a gas flow along a solid surface. Taitel and Dukler (1976a), Vlachos et al. (1997), Khor et al. (1997) are among many of the investigators who used the definition of the interfacial shear stress discussed above.

The constants C and m in Eqs. 3.38a to 3.38c are defined as described earlier (i.e., for laminar flow, $C = 16$ and $m = -1$, and for turbulent flow, $C = 0.046$ and $m = -0.2$). To avoid the need of specifying a transition Re , and to eliminate any discontinuity in the transition between laminar and turbulent flow, the friction factor is calculated as $f = (f_{lam}^3 + f_{turb}^3)^{1/3}$.

The gas and liquid equivalent hydraulic diameters (d_{eg} and d_{el}) in Eqs. 3.38a to 3.38c are defined as follow:

$$d_{eg} = \frac{4 \langle \alpha \rangle A}{P_{wg} + P_i} \quad (3.39a)$$

$$d_{el} = \frac{4 (1 - \langle \alpha \rangle) A}{P_{wl}} \quad (3.39b)$$

The definitions of d_{eg} and d_{el} in Eqs. 3.39a and 3.39b follow the definition of Taitel and Dukler (1976b). These definitions assume that the gas flows in a closed channel, i.e., both the tube and the liquid interface make-up the walls surrounding the gas,

while the liquid is assumed to flow in an open channel. Taitel and Dukler (1976b) provided no justifications for their assumption. This assumption is somewhat arbitrary since it represents an inconsistency in the definition of d_{eg} and d_{cl} , so its effect on the friction factor has been assessed here. However, for the case of stratified flow, excluding P_i in Eq. 3.39a is estimated to result in only a small decrease of the friction factor (less than 5%). Note that this definition of d_{eg} is widely used in two-phase flow studies, e.g., Kokal and Stanislav (1989), and Khor et al. (1997). It is for that reason that it is considered in the present study.

3.5.1 Two-phase Flow Pressure Gradient

An expression for the frictional pressure gradient $\left(\frac{dP}{dx}\right)_{TP}$ can be obtained by summing-up Eqs. 3.36a and 3.36b to eliminate the interfacial friction term ($\tau_i P_i$) and the average void fraction $\langle\alpha\rangle$

$$\left(\frac{dP}{dx}\right)_{TP} = -\tau_{wl} \frac{P_{wl}}{A} - \tau_{wg} \frac{P_{wg}}{A} \quad (3.40)$$

It should be noted that although $\langle\alpha\rangle$ does not appear in the above equation, it is still required to determine the shear stresses and the contact perimeters.

The hydraulic diameters and the contact perimeters of the different flow regimes are given in the following sections.

(a) Stratified Flow Regime

Figure 3.9 shows typical distribution of the gas and liquid phases in stratified wavy flow regimes. Stratified smooth flow assumes a similar configuration with the exception of the smooth gas–liquid interface. Therefore, the contact perimeters for both regimes are defined in a similar manner.

The hydraulic diameters and the contact perimeters for stratified flow are given as a function of θ , which is half of the angle subtended at the centre of the pipe by the width of the gas–liquid interface as shown in Fig. 3.9.

$$P_{wl} = \theta d \quad (3.41a)$$

$$P_{wg} = (\pi - \theta) d \quad (3.41b)$$

$$P_i = \sin(\theta) d \quad (3.41c)$$

$$d_{el} = \frac{\pi d}{\theta} (1 - \langle \alpha \rangle) \quad (3.41d)$$

$$d_{eg} = \frac{\pi d}{(\pi - \theta) + \sin(\theta)} \langle \alpha \rangle \quad (3.41e)$$

The following expression relates the angle (θ) to the void fraction (α):

$$\langle \alpha \rangle = 1 - \frac{1}{2\pi} (2\theta - \sin(2\theta)) \quad (3.42)$$

(b) Annular Flow Regime

As discussed in Chapter 2, annular flow is characterized by a liquid film flowing along the wall of the channel, while the gas phase flows in the core of the tube entraining small droplets of liquid. The fraction of liquid entrained in the gas core depends on the gas and liquid flow-rates. Govan et al. (1988) suggested that entrainment starts at a certain liquid film Reynolds number given by

$$Re_{f, crit} = \exp \left[5.8504 + 0.429 \left(\frac{\mu_g}{\mu_l} \right) \left(\frac{\rho_l}{\rho_g} \right)^{1/2} \right] \quad (3.43)$$

Collier and Thome (1994) indicated that for film Reynolds number above 3000, the onset of entrainment is a function of the gas flow rate only. Wallis (1969) suggested that for turbulent film flow, $Re_f > 3000$, entrainment starts at

$$J_{g, crit} = 2.42 \cdot 10^{-4} \left(\frac{\rho_l}{\rho_g} \right)^{1/2} \frac{\sigma}{\mu_g} \quad (3.44)$$

Note that $Re_{f, crit}$ and $J_{g, crit}$ are a function of the gas and liquid properties only. Using the properties of air and water at standard atmospheric conditions, $Re_{f, crit} = 429$ and $J_{g, crit} = 28.3$ m/s.

Simple expressions for the hydraulic diameters and the contact perimeters can be obtained if the fraction of liquid phase entrained in the gas core can be ignored.

$$P_{wl} = \pi d \quad (3.45a)$$

$$P_{wg} = 0 \quad (3.45b)$$

$$P_i = \pi \sqrt{\alpha} d \quad (3.45c)$$

$$d_{el} = (1 - \alpha) d \quad (3.45d)$$

$$d_{eg} = \sqrt{\alpha} d \quad (3.45e)$$

(c) Plug Flow Regime

A simplified model is developed to calculate the friction losses in the plug flow regime. The model is based on the idealized representation of the flow shown in Fig. 3.10. The disposition of the phases along the test section is divided into single-phase liquid and two-phase gas–liquid sections. The average pressure gradient along the test section is calculated as the weighted-average of the pressure gradients in the single-phase and two-phase regions using the following equation:

$$\left(\frac{dP}{dx}\right)_{TP,P} = \frac{L_S}{L_S + L_B} \left(\frac{dP}{dx}\right)_l + \frac{L_B}{L_S + L_B} \left(\frac{dP}{dx}\right)_{TP,St} \quad (3.46)$$

where L_B is the average length of the elongated bubbles, and L_S is the average length of the liquid slugs that separate the bubbles (see Fig. 3.10). The ratio $L_B:(L_B+L_S)$ represents the average fraction of the test section simultaneously occupied by the two phases, i.e., two-phase flow regions. This ratio can be determined experimentally.

In the single-phase regions, the average frictional pressure gradient $\left(\frac{dP}{dx}\right)_l$ is calculated using the conventional single-phase flow relationship

$$\left(\frac{dP}{dx}\right)_l = \frac{1}{2} f_l \rho_l u_l^2 \frac{P}{A} \quad (3.47)$$

where P and A are the perimeter and the cross-sectional area of the channel, respectively. u_l is the average velocity of the liquid in the single phase flow regions, and f_l is the friction factor calculated based on u_l , assuming fully developed flow. Note that u_l is calculated based on the measured liquid flow rate and the overall average void fraction (α).

In the two-phase flow regions, the flow is assumed to behave like the stratified smooth flow regime. As discussed in Chapter 2, in plug flow the gas-liquid interfaces are smooth and the phases move at relatively the same velocity. Therefore, within the two-phase flow sections, the disposition of the phases can be approximated as stratified smooth flow with a certain void fraction (α_p), which is the average void fraction within the two-phase flow regions.

The pressure gradient in the two-phase flow regions $\left(\frac{dP}{dx}\right)_{TP,St}$ is calculated in a similar manner as in stratified wavy flow using Eqs. 3.41a to 3.41e, with contact

perimeters and the hydraulic diameters of the phases based on the average void fraction within the two-phase flow regions α_p , which α_p can be obtained from the overall void fraction (α) and the fraction of the test section simultaneously occupied by both phases (i.e., $L_B:(L_B+L_S)$)

$$\alpha_p = \frac{\alpha}{\left(\frac{L_B}{L_B + L_S} \right)} \quad (3.48)$$

(d) Slug Flow Regime

In slug flow, the gas and liquid phases co-exist throughout the entire length of the test section. The liquid slugs are usually aerated and considerably shorter than those observed in plug flow. On average, the aerated liquid slugs that completely block the flow area of the gas phase represents less than ~20% of the entire length of the test section. Therefore, slug flow cannot be modeled in a similar manner as plug flow, i.e., the flow cannot be divided into single-phase and two-phase flow regions.

Slug flow can be viewed as stratified flow with extremely wavy and unstable gas–liquid interfaces. Measurement of the frictional pressure gradients can be correlated as in stratified wavy flow to determine the increase in the interfacial friction factor. The interfacial drag term is expected to be significant in this case because of the large difference between the velocities of the phases.

3.5.2 Correlation of the Gas–liquid Interfacial Friction Factor

A number of experimental studies used measurements of the frictional pressure gradient to provide expressions for the gas–liquid interfacial friction factors (f_i). Vlachos et al. (1997) used measurements of the liquid–wall shear stress and pressure gradients in stratified and droplet¹² flow regimes to develop a correlation for f_i . The liquid–wall shear stress was measured at different locations around the circumference of 24- and 50-mm pipes using an electrochemical technique. Their measurements showed that the liquid–wall shear stress decreased in the lateral direction away from the bottom of the pipe as the

¹² In droplet flow most of the liquid phase is carried by the gas phase in the form of small liquid droplets.

thickness of the film decreased. However, for relatively low gas superficial velocity ($J_g < 15$ m/s), the time averaged shear stress was almost constant near the bottom of the tube (up to about 45°). Vlachos et al. (1997) proposed the following expression for the liquid–wall shear stresses:

$$f_l = 0.7 Re_{lf}^{-0.5} \quad (3.49)$$

where Re_{lf} is the liquid film Reynolds number based on the average thickness of the film at the bottom of the pipe. For the 24-mm pipe, $Re_{lf} < 1000$, and for the 50-mm pipe, $Re_{lf} < 10,000$. It is interesting to note that, for Re_{lf} within the range of their experimental data (i.e., $Re_{lf} < \sim 10,000$), Eq. 3.49 gives higher friction factors than the conventional single-phase flow equation. However, at $Re_{lf} > \sim 10,000$, Eq. 3.49 predicts lower friction factors than the conventional single-phase flow equation. Therefore, Eq. 3.49 can not be used with confidence for Re_{lf} outside the range of the experimental data (i.e., $Re_{lf} > 10,000$).

Vlachos et al. (1997) used the momentum balance equations, Eq. 3.49 for the liquid–wall friction factor, Eq. 3.16 for the gas–wall friction factor, and expressed the interfacial shear stress in terms of f_i and J_g to obtain the following correlation for f_i in stratified and droplet flow regimes:

$$f_i = 0.024 (1 - \alpha)^{0.35} Re_l^{0.18} \quad (3.50)$$

where Re_l is the liquid Reynolds number based on the liquid superficial velocity and the pipe diameter.

Newton and Behnia (1998) developed expressions for the interfacial friction factors for stratified flow in 50- and 80-mm horizontal pipes. Using measurements of the gas–wall shear stress, the gas pressure drop, and the thickness of the liquid film, Newton and Behnia (1998) suggested the following expressions for f_i :

$$50\text{-mm pipe} \quad f_i = 8.9 \cdot 10^{-7} h_L^{-0.5} Re_g^{0.41} Re_l^{0.6} \quad (3.51a)$$

$$80\text{-mm pipe} \quad f_i = 8.6 \cdot 10^{-7} h_L^{-0.23} Re_g^{-0.19} Re_l^{1.2} \quad (3.51b)$$

where h_L is the average height of the liquid film, Re_g and Re_l are gas and liquid Reynolds numbers calculated based on the flow area of each phase. Note that Newton and Behnia (1998) used a Preston tube to measure the gas–wall shear stresses and did not provide a correlation for f_g from their measurements.

The following describes the approach developed in the present study to correlate the interfacial friction factors, for the different flow patterns, using measurements of the frictional pressure gradients and the average void fractions. An expression for the frictional pressure gradients that includes the interfacial shear stress was obtained from simplified momentum balance equations of the gas and liquid phases, Eqs. 3.36a and 3.36b. Subtracting Eq. 3.36a from Eq. 3.36b gives the following expression:

$$\left(\frac{dP}{dx} \right)_{TP} = \frac{l}{(1 - 2 \langle \alpha \rangle) A} \left[-\tau_{wl} P_{wl} + \tau_{wg} P_{wg} + 2 \tau_i P_i \right] \quad (3.52)$$

The contact perimeters (P_{wl} and P_{wg}) were calculated based on measurements of the average void fraction, and the simplified disposition of the phases. Since no measurements of either the liquid–wall or the gas–wall shear stresses were taken, τ_{wl} and τ_{wg} were calculated using the conventional single-phase flow equations. It is recognized that the conditions of the gas–liquid interfaces may affect the velocity profiles within the phases such that the conventional single-phase flow relationships are invalidated. The best approach would be to correlate the interfacial friction factors based on measurements of both the liquid–wall and the gas–wall shear stresses. However, in the absence of accurate relationships for the wall friction factors, the present study uses the conventional relationship and introduces a coefficient to correct for these effects and for the increased interfacial friction factor as a result of the waves. Note that Vlachos et al. (1997) and

Newton and Behnia (1998) measured the wall shear stress for one of the phases and used the conventional relationship to calculate the friction factor for the other phase.

The definition of the interfacial shear stress was modified to include a correction coefficient C_w to account for; (i) the effects of the relative velocity of the gas–liquid interfacial conditions on the velocity profiles in the phases, and (ii) the increased friction factor as a result of the waves. Therefore, the interfacial shear stress was defined as follows:

$$\tau_i = \frac{l}{2} C_w f_i \rho_g (u_g - u_l)^2 \quad (3.53)$$

where f_i is the interfacial friction factor for smooth gas–liquid interfaces calculated using Eq. 3.38c, and C_w is a waviness coefficient defined earlier.

Substituting Eq. 3.53 into Eq. 3.52 gives the following expression for the interfacial shear stress:

$$\frac{l}{2} C_w f_i \rho_g (u_g - u_l)^2 P_i = \frac{l}{2} \left[(1 - 2 \langle \alpha \rangle) A \left(\frac{dP}{dx} \right)_{TP} + \tau_{wl} P_{wl} - \tau_{wg} P_{wg} \right] \quad (3.54)$$

Rearranging the above equation gives the following expression for the equivalent interfacial friction factor ($f_{i,eq} = C_w f_i$):

$$f_{i,eq} = \frac{l}{\rho_g (u_g - u_l)^2 P_i} \left[(1 - 2 \langle \alpha \rangle) A \left(\frac{dP}{dx} \right)_{TP} + \tau_{wl} P_{wl} - \tau_{wg} P_{wg} \right] \quad (3.55)$$

This equation can be used to calculate the interfacial friction factors using the measured void fraction $\langle \alpha \rangle$, and the measured pressure gradient $\left(\frac{dP}{dx} \right)_{TP}$. The contact perimeters and hydraulic diameters were calculated using the measured void fractions and the simplified dispositions of the phases presented in Section 3.5.1.

3.5.3 Interfacial Area Concentration

As indicated earlier, the two-fluid models require detailed information about the concentration and the distribution of the gas–liquid interfaces to allow for momentum, heat, and mass transfer between the phases. The interfacial area concentration is defined as the area of the gas–liquid interfaces per unit volume of the mixture (usually given in units of m^2/m^3). The interfacial area concentration was identified as one of the most important parameters in two-phase flow studies (Ishii, 1992).

Two-fluid models require accurate information about the distribution and concentration of the gas–liquid interfaces. A number of models and correlations were provided to predict the concentration of the gas–liquid interfaces in different flow regimes, such as Tomida et al. (1978), Ishii and Mishima (1984), Tabei et al. (1989), Zietoun et al. (1994), and Riznic et al. (1996). Riznic et al. (1996) published the results of their measurements of the interfacial area in horizontal slug flow. Most other models and correlations were developed for vertical flow regimes. Because of differences in the flow patterns and the non-axisymmetric nature of the horizontal flow, these models may not be applicable to horizontal flows.

In this study, measurements of the interfacial area were obtained for all flow regimes; however, analysis of these measurements was limited to those of the stratified flow test runs. For other flow regimes, the analysis will be performed as follow-up work.

Stratified Flow

For stratified smooth flow, an expression for the gas–liquid interfacial area can be obtained directly from geometrical considerations,

$$a_i = \frac{4 \sin(\theta)}{\pi d} \quad (3.56)$$

where θ is the angle defined in Fig. 3.9, which can be determined by

$$\theta = \arccos\left(1 - 2\frac{h_L}{d}\right) \quad (3.57)$$

where h_L is the average height of the liquid layer in stratified flow. Equation 3.56 gives the interfacial area for stratified smooth flow pattern. As relative velocity between the phases increase, waves are formed, thus resulting in an increase in interfacial area concentration of the gas–liquid interfaces.

3.6 Concluding Remarks

A number of models and correlations were suggested for predicting the average void fraction and the frictional pressure gradient. These models and correlations are categorized into two groups: (i) generalized models and correlations, which were developed independently of the flow regimes; and (ii) regime-specific models and correlations, which were developed based on the particular configuration of the flow regime.

Generalized Models

A number of models and correlations were proposed to predict the average void fractions and the frictional pressure gradients without prior knowledge of the prevailing flow regime. The frictional pressure drop and the void fractions are evaluated in a similar manner for all flow patterns and often regardless of the orientation of the conduit. However, as the void distribution and the velocity profile change with the prevailing flow pattern and tube orientation, it can be expected that the pressure loss due to friction will vary. The most widely used models and correlations were reviewed in this study. These include the following:

- (i) The homogeneous model for frictional pressure gradient and the average void fraction. A number of definitions of the mean viscosity of the mixture viscosity

were proposed for use with the homogeneous model to predict the frictional pressure gradient.

- (ii) The Lockhart–Martinelli (1948) correlation, which is still widely used to predict friction losses in two-phase flow systems and to correlate experimental data.
- (iii) Empirical correlations of Martinelli-Nelson (1948), Baroczy (1966), Chisholm (1968), and Friedel (1979) for predicting the frictional pressure gradients and the CISE correlation for predicting the average void fraction.

The correlations of Martinelli-Nelson (1948), Baroczy (1966), and Chisholm (1968), are entirely graphical. These correlations are difficult to use since they require interpolation between the meandering lines. Therefore, they are not considered any further in this study.

The correlation of Friedel (1979) is considered by many investigators to provide the best prediction of frictional pressure gradients.

Regime-specific models

These models and correlations are expected to provide better predictions of the friction losses and the pressure gradients since they were based on the particular configuration of the flow pattern. The main disadvantage of these models and correlations is that in practical situations often the prevailing flow regime is unknown. Also, as discussed in Chapter 2, different classifications of the flow patterns were proposed.

Over the past three decades, a number of regime-specific correlations were proposed for the average void fraction and frictional pressure gradients. The drift-flux model is widely used to predict the average void fraction. However, most of the correlations were suggested for vertical channels. França and Lahey (1992) showed that the drift-flux model could be used to correlate the average void fraction in horizontal

channels. França and Lahey (1992) used different drift-flux variables for stratified wavy and annular flow regimes. However, it was shown in Section 3.4.1 that the standard variables can also be used to correlate the data for these flow regimes.

The models of Agrawal et al. (1973), Dukler and Hubbard (1975), Nicholson et al. (1978), Hashizume et al. (1985, 1987), and Hart et al. (1989) are some of those proposed to predict the pressure gradients for particular flow regimes. As indicated earlier, these models were expected to give better predictions of the friction losses. However, the comparative study of Tribbe and Müller-Steinhagen (2000) showed that the accuracy of these models is similar to those of the general models.

Baker (1961), Beattie (1973), and Bandel (1973) proposed correlations for predicting the frictional pressure gradients in each of the flow regimes. Only the correlation of Bandel (1973) provides criteria for the identification of the prevailing flow regime.

Two-fluid models were developed for the stratified wavy, slug, annular, and plug flow regimes. The models were developed using the momentum balance equations and were based on simplified configurations of the phases. Equations were also developed to calculate the gas-liquid interfacial friction factors for the different flow regimes from measurements of the frictional pressure gradients and void fractions.

Figure 3.1: Estimates of the mean viscosity of the two-phase mixture.

Figure 3.2: Empirical Correlations of Lockhart and Martinelli (1949) for friction multiplier and average void fraction (from Collier et al., 1994).

Figure 3.3: Empirical Correlation of Martinelli and Nelson (1948) for friction multiplier for steam-water systems (from Collier et al., 1994).

Figure 3.4: Correlation of Martinelli and Nelson (1948) for friction multiplier at different exit quality (from Collier et al., 1994).

Figure 3.5: Empirical correlation of Baroczy (1965) for friction multiplier, (from Collier and Thome, 1994).

Figure 3.6: Empirical correlation of Baroczy (1965) and correction factor for mass velocities (from Collier et al., 1994).

Figure 3.7: Correlation of the experimental data of França and Lahey (1992).

Figure 3.8: The limiting pressure gradients of Bandel (1973).

Figure 3.9: Simplified model for the two-fluid approach.

Figure 3.10: Idealized representation of the plug flow regime.

Experimental Facility and Measurement Techniques

4.1 Introduction

Two-phase flow is an area in which theoretical prediction of flow parameters is not generally possible. Thus, the role of experiments and accurate parametric measurements is particularly important. The degree of precision of the experimental results obtained depends on, among other factors, the apparatus, the instruments used and the test procedure.

The following parameters are known to govern the characteristics of the two-phase mixture: flow conditions of the phases, such as flow rate, static pressure, and temperature; properties of the phases, such as density, viscosity, and the liquid surface tension and contact angle; void distribution and test section geometry and orientation. These parameters have a direct impact on the characteristics of the prevailing two-phase flow regime. The inlet flow conditions of the phases were measured using common single-phase flow measurement techniques, which will be discussed in this chapter.

Hewitt (1978), Jones (1980) and Delhaye (1982) published detailed reviews of measurement techniques of various two-phase flow parameters. Only those techniques used in the present experimental study are discussed in this chapter.

4.2 Description of Test Facility

4.2.1 General Description

Descriptions of the experimental test rig were previously presented by Lamari (1990) and should be consulted for information not presented here. Some modifications to the apparatus were made for the present experimental investigation. These changes include the implementation of an interfacial area measurement system. The system was

implemented after the test rig was moved in 1994 to its new location in Room 2354 of the Engineering Building (C.J. Mackenzie) at Carleton University, Ottawa.

The experimental test rig consists of the following major components:

- (i) a circulation loop and test tube;
- (ii) general instrumentation;
- (iii) void distribution measurement system;
- (iv) air–water interfacial area measurement system.

The following sections provide brief descriptions of the experimental facility. Calibration and accuracy of the different instruments used in this study are presented and discussed in Appendix B.

The general layout of the two-phase flow test rig is schematically shown in Fig. 4.1. The working fluids were air and water at near standard atmospheric conditions. The metered single phases were fed to a mixing chamber, and the resulting two-phase mixture flowed through the test section into the separator, where the air was vented to atmosphere and the water returned by gravity to the storage tank.

The test rig was equipped with instruments to monitor air and water inlet flow conditions, such as flow rates, static pressures and temperatures. Static pressure was also measured at three different locations along the test section. The locations of the instruments and static pressure taps are shown in Fig. 4.1. The test rig was fully automated: a computer-based data acquisition system was used to monitor all parametric measurements, and a number of special subroutines were developed to reduce and analyze the raw data.

The key features of this experimental test facility include the following:

- (i) a large length to diameter ratio of the test tube of $l/d = 540$;

- (ii) use of a twin-needle fibre-optic probe to gather local and instantaneous information on the void distribution across the pipe diameter;
- (iii) the air–water interfacial area measurement system, which simultaneously videotapes the horizontal- and vertical-planes of the test section.

Descriptions of the apparatus and the measurement techniques are presented in the following sections.

4.2.2 Test Section

The length of the test section was recognized as one of the key parameters in the design of two-phase flow experimental facilities. Simpson et al. (1977) observed entrance effects on the development of flow patterns at $l/d = 128$. This highlights the need for a very long test section for flow regimes to develop fully and naturally. Forcing the mixture to take a particular distribution at the inlet on the test tube contradicts one of the main objectives of this work, which is to study the distribution of the phases, and to identify the mechanisms that govern the transition between the flow patterns. Therefore, it is important that the flow is developed under natural conditions.

In this experimental study the air–water mixture was developed in a 25.4-mm clear Plexiglas test tube with a total length of 13.65 m. This allows for a length to diameter (l/d) ratio of 540. Experimental data such as void distribution and interfacial area concentration were acquired at an axial location of about $l/d = 410$, which ensures that the two-phase mixture was fully developed at the measuring point.

The test tube was carefully manufactured, with male and female grooved flanges used to align sections of the tube that form the test tube.

As discussed in Section 2.6, some of the flow pattern transition models, such as that of Taitel and Dukler (1976a), predict large shifts in the location of the boundaries of the flow patterns because of small inclinations of the flow conduit from the horizontal. Therefore, the test section was accurately levelled. Conventional levelling instruments

were used to obtain an approximately horizontal test tube. Then, precise adjustments were made by adjusting the fine threaded floating bases of the test tube, while observing the level through a theodolite¹¹.

The phases were introduced into the test section through a T junction (mixing section). As shown in Fig. 4.1, water was introduced in the branch and air in the run of the T. The large diameter of the mixing section assisted in producing a well-mixed flow at the inlet of the test tube. The large L/d flow-development section ensures that the phases are fully developed and so specifics on the introduction of the phases into the test section are not as important.

The two phases were separated using a simple air–water separator located downstream from the test tube (Fig. 4.1). The air–water mixture was injected into the separator at a 20° inclination below the horizontal and to one side of the separator. This configuration forces the mixture to spin inside the separator with the water against the wall and the air at the centre. Separation was completed by gravity, with water flowing down the return pipe to the storage tank and air vented to the atmosphere.

4.2.3 Air Supply System

Air was provided by the University central compressed air system, which consists of two identical 50-hp compressors manufactured by Ingersoll. The system supplies satisfactory airflow conditions capable of establishing the various two-phase flow regimes discussed in Section 2.2.

Before air was introduced into the test section, it passed through a preparation unit that consists of a settling chamber, a filter and a pressure regulator (Fig. 4.1). Air was fed into the settling chamber through a 25.4-mm pipeline. The settling tank is 0.406 m in diameter and 0.965 m long. The length of the settling tank ensures the dispersion of the air jet at the inlet of the tank, while the diameter keeps the air velocity inside the tank

¹¹ Theodolite is a level survey instrument that provides accuracy better than ± 0.5 mm

under 1% of the air velocity in the supply line. The tank smoothes out any flow oscillation caused by the air supply system.

As the air exits the tank, it goes through a conditioning unit that combines the functions of a filter and a pressure regulator. The filter eliminates contaminants from the atmosphere that could be drawn into the air system at the intake of the compressors. It removes bulk liquid (e.g., oil) and solid particles down to 5 μm .

The air-settling tank is equipped with a relief valve and a pressure indicator.

4.2.4 Water Supply System

For test runs requiring low water mass flow rates, laboratory (tap) water was filtered and then fed into the test section. For all other test runs, filtered water was recirculated through a 2.0-m³ reservoir (Fig. 4.1). The water was recirculated using a 230-mm centrifugal pump driven by a 25-hp induction motor.

A by-pass system with two manually controlled valves was used to control the water flow rate through the test section. The flow rate and the inlet temperature of the water supply were monitored.

As discussed in Lamari (1990), the temperature of the recirculated water continuously increased with time, mainly because of the heat generated by the pump. A feed and bleed technique was used to stabilize the temperature of the water: cold water was continuously added to the system while heated water was continuously extracted.

4.3 Measurements of Single-Phase Flow Parameters

4.3.1 Air Mass Flow Rates

A thermal type mass flowmeter was used to measure the air mass flow rates. This device, known as an air velocity transducer (AVT), determines the air velocity by measuring the energy required to maintain a heated sensor at a constant temperature. The

AVT had a built-in temperature sensor that automatically corrects for any variations in the air free-stream temperature. The transducer was placed with sufficient upstream and downstream straight-run piping to ensure that the airflow was fully developed at the location where it was being measured.

The AVT transducer was calibrated in situ using a Pitot-static tube. The calibration method and results are presented in Appendix B.

4.3.2 Water Flow Rates

A paddlewheel sensor was used to measure the water flow rates. The sensor was placed in a 12.7-mm copper tube with sufficient upstream and downstream straight-run piping to ensure that the water flow was fully developed.

The sensor was calibrated in situ. The calibration method is presented in Appendix B.

4.3.3 Static Pressure

Static pressure was measured, with reference to atmospheric pressure, at the following four locations:

- (i) air inlet pipe, where the air mass flow rate is measured;
- (ii) inlet of test section, downstream of the air–water mixer;
- (iii) about $l/d = 420$ from the inlet, just downstream from where void fraction and interfacial area measurements were taken; and
- (iv) near the end of the test tube.

Pressure taps, made of stainless steel tubing (~1.6 mm in diameter and 15 mm long) were placed at the bottom of the pipe wall. The holes in the pipe wall were carefully machined to ensure that they were blister free and square edged.

It should be noted that attempts were made to acquire average values of the static pressure around the circumference of the test tube. This is usually achieved using a number of pressure taps placed at equal distances around the circumference of the tube. The taps are then interconnected to allow the average pressure to be measured. Unfortunately, because of flow stratification, air at the top of the tube enters the lines that connect the pressure taps to the pressure transducers. Therefore this technique was abandoned, and the static pressure was measured at the bottom of the pipe at all three locations along the test section.

Pressure drop measurements were carried out using calibrated reluctance-type electronic pressure transducers. The transducers were placed close to the pressure taps (less than 50 mm). The lines to transducers were filled with distilled water (to avoid water degassing) and carefully purged of air bubbles. The lines were continuously monitored during the experiments.

All pressure transducers employed were calibrated. The calibration method and results are given in Appendix B.

4.3.4 Temperature

Type T copper–constantan thermocouples were used to measure air inlet temperature, water inlet temperature and the temperature of the two-phase mixture downstream from the test tube (locations are shown with T in Fig. 4.1). The average temperature of the mixture was taken as the arithmetic average of the three temperatures. The difference between thermocouple readings was always less than 2°C.

The accuracy of the thermocouple readings is discussed in Appendix B.

4.3.5 Water Surface Tension

The flow pattern transition models discussed in Chapter 3, e.g., Taitel and Dukler (1976a) and Weisman et al. (1979), indicate that some of the mechanisms that govern the

transition between the flow regimes depend on the surface tension of the liquid phase. Although in this study it was not planned to vary the surface tension of the water, oil contaminants from the water circulation pump or the air compressors may pollute the air–water mixture, changing the surface tension of the liquid. A small amount of oil can significantly reduce the surface tension of the water. Therefore, it was considered beneficial to estimate the range over which the water surface tension is expected to vary in the present experiments.

A number of water samples were taken at different locations throughout the test facility and under different conditions. The sampling locations and the conditions under which the samples are taken were described in Appendix B. These conditions were expected to cover the anticipated conditions under which the experiments were to be performed.

The surface tension of the water samples was measured. Initially, a capillary tube and an *O-ring* tensiometer were tested to assess their ability to measure water surface tension. Both instruments were used to measure surface tension of distilled water at room temperature. Readings from both instruments for the same water sample were not consistent and ranged from 46 to 80 mN/m. The published value for the surface tension of water at 20°C is 72.75 mN/m (Findlay, 1973). The accuracy of both instruments was therefore considered unacceptable.

A pendant drop technique was used to measure the surface tension of 10 water samples. Each sample was measured three times and the readings ranged from 69.6 to 72.5 mN/m. Details of the pendant drop technique and the measurement results are given in Appendix B.

It should be noted that a number of water samples were collected for different test runs. The samples were stored for possible later analysis if necessary.

4.4 Measurements of Two-phase Flow Parameters

This section provides descriptions of the measurement techniques used in this study. Hewitt (1978, 1982), Delhayé (1974, 1977) and Jones (1983) published detailed reviews on the measurement techniques of various two-phase flow parameters. In this study, measurements of the local void fraction and the concentration of the air–water interfaces were obtained. In addition, other parameters, such as length and frequency of the liquid slugs and gas bubbles and velocity of the gas–liquid interfaces, could be determined from analysis of the time traces of the probes used to measure the void fraction. The probes and the time traces are discussed later in this chapter. However, analysis of the time traces is considered beyond the scope of this study. Detailed analysis of the time traces of the probes will be considered at a later date.

4.4.1 Average Void Fraction

Void fraction is perhaps the most important parameter in two-phase flow studies. Knowledge of the void fraction is required for the calculations of most of the two-phase flow parameters, e.g., pressure losses and heat transfer coefficients.

Void fraction measurements are usually divided into the following four general categories, depending on the area over which measurements are averaged:

- (i) *Channel average*: This gives the average void fraction of the full length of the test section. Quick-closing valves are commonly used for this type of measurement.
- (ii) *Cross-sectional average*: This gives the average void fraction of the channel's cross-sectional area. This method employs either neutron-scattering techniques or impedance void gauges.
- (iii) *Chordal average*: This gives the average void fraction along a diameter of the test tube. These measurements are usually achieved by radiation absorption methods, e.g. γ - and X-ray systems.

- (iv) *Local measurements*: These give local and instantaneous void fraction measurements. Some of the most commonly used techniques include electrical probes, optical probes, thermal anemometers, and radio-frequency probes.

Local measurement of the void fraction is consistent with the objectives of this study. This method provides detailed information about the distribution of the phases. In this study, an optical probe is used to measure the void fraction at nine equally-spaced radial locations. Optical probes have been successfully used to provide local and instantaneous measurements of the void fraction. Miller and Mitchie (1969) used a high-speed photography technique (5000 frames/s) to study the accuracy of a single fibre-optic probe. They concluded that by using such probes, the local void fraction can be measured to within $\pm 10\%$. Better accuracy is expected to be obtained in this study because of the improved characteristics of the instruments used, i.e., the high sampling rate and the very short response time of the light detection equipment.

4.4.2 Optical Probe

Principle of Operations

The principle of operation of optical probes is based on Snell's law of refraction (Serway, 1999). This law states that when a ray of light travelling through a transparent medium (e.g., glass fibre) encounters a boundary leading to another transparent medium (e.g., gas or liquid), part of the ray is reflected and part is refracted into the second medium. The intensity of the reflected part of the ray of light depends on among other parameters¹², the optical density of the surrounding medium, i.e., its refractive index. The lower the optical density of the surrounding medium, the more light is reflected back into the glass fibre. With gasses being less optically dense than liquids, more light would be reflected back along the glass fibre if the tip of the fibre is submerged in the gas phase. The refractive index of air is 1.0 compared with 1.33 for water.

¹² The intensity of the reflected part of the ray also depends on the angle of incidence (angle at which a light beam strikes the surface of the glass fibre), and on the conditions of the surface of the glass fibre, i.e., surface roughness.

Based on Snell's law and the difference in the refractive index between gas and liquid, a variety of optical probes was used to measure the local void fraction. The most common type of these probes consists of a bifurcated glass fibre, i.e., Y-shape. As shown in Fig. 4.2, light is transmitted from one end of the glass fibre towards the sensing tip of the probe, which is the base of the fibre. The intensity of the reflected part of the light is measured at the third end of the fibre. More light is reflected back along the glass fibre if the sensing tip is surrounded by gas than by liquid. The intensity of the returning light is recorded over a specific period of time, and the time-averaged local void fraction is determined as the fraction of the time the sensing tip of the probe was in contact with the gas phase.

Probe Details

A twin-needle fibre-optic probe was used to measure the local void fraction and to collect detailed information about the distribution of the phases. The probe consists of two identical bifurcated (Y-shaped) glass fibres manufactured by AMP Inc. Each fibre measures the local void fraction and provides information about other parameters of interest, such as frequency of liquid slugs, cutting length and passing frequency of air bubbles. However, both fibres are necessary to measure the velocities of the air–water interfaces, as will be discussed later in this section.

The probes were used in tandem (Fig. 4.2). The core of the glass fibre is less than 50 μm in diameter (this small diameter size minimizes deflection of air bubbles from the tip of the fibre). Both fibres are inserted into a protective stainless steel tube (less than 1 mm in outside diameter). The tube is bent at an angle of 90°, giving a length of ~30 mm aligned with and facing the flow. The sensing tips of the fibres are displaced from one another by approximately (δx) 2 mm in the direction of the flow and less than 1 mm in the radial direction (Fig. 4.2). The downstream fibre is placed above the upstream fibre. This arrangement allows for any bubbles deflected from the upstream sensor to be detected by the downstream sensor.

As shown in Fig. 4.2, light from a 514-nm Argon laser is split into two equal parts using a light beam splitter. Each part of the light is transmitted through one end of each of the fibres towards the sensing tip. The reflected light from each fibre is fed into a photodetector that converts the optical signal (i.e., intensity of the reflected light) into an electrical signal. The output signals of the photodetectors are recorded on a two-channel digital oscilloscope.

Figure 4.3 shows typical untreated signals obtained from the upstream and downstream sensors at a location near the top of the test tube. A high output voltage indicates that more light is reflected back along the fibre, thus the sensing tip of the fibre is immersed in air, which is optically less dense than water. The local time-averaged void fraction is therefore calculated as the fraction of time the sensing tip of the probe is immersed in air. Analysis of the recorded signals is discussed in Section 4.6.2.

Figure 4.3 also shows a time delay between the signals acquired with the upstream and downstream probes. The output signal of the downstream sensor lags that of the upstream sensor by a time δt . This time shift represents the time it takes the air–water interface to travel the distance separating the sensing tips of the probes (δx). The average velocity of the interfaces can be calculated from the average δt and δx .

Probe Calibration

Each of the probes (along with the photodetectors and related circuitry) was tested to assess its transient behaviour, i.e., how fast does the system respond to a change in light signal. More accurate measurements of the void fraction can be obtained with fast-response probes, especially at high velocities. A simple setup was used to estimate the time constant of the probe. With a stroboscope operating at a frequency of 150 kHz, the output signal of each of the photodetectors was recorded on a digital oscilloscope. This test shows that the system gives virtually instantaneous response to light signals emitted by the stroboscope. The time response of the photodetectors is estimated at less than 10^{-6} seconds, corresponding to a dynamic response of 1 MHz.

As indicated earlier, the intensity of the reflected light depends on the condition of the surface of the sensing tip of the probe. During the initial testing and calibration of the probe, it was found that the output signal of the photodetectors depends on the preparation of the sensing tip of the glass fibre. The probe was found to give a clear distinction between the air and water phases at the sensing tip if the cut of the fibre tip is clean and straight (Fig. 4.4a). If the tip of the fibre is not well prepared (Fig. 4.4-b), part of the light is scattered away at the rough surface of the sensor, resulting in a lower intensity of the reflected light and a weaker signal transmitted by the photodetector, even if the tip of the fibre is immersed in air (the gaseous phase signal approaches that of the liquid).

In addition, a rough surface of the sensing-tip may also cause the following unwanted behaviour:

- (i) air–water interfaces dragging on the rough surface of the sensing tip of the fibre, resulting in a significant increase in the response time of the sensor and reducing its overall efficiency; and
- (ii) small water droplets or air bubbles may be attached at the rough surface of the sensor, shielding the sensing tip of the probe from detecting the medium surrounding it.

In this study, the sensing tips of the probes were carefully prepared and examined with a microscope prior and following each set of experimental test runs to confirm that they met the required conditions.

4.4.3 Gas–Liquid Interfacial Area Concentration

The interfacial area concentration (or interfacial area density) is the area of the moving gas–liquid interfaces per unit volume of the mixture. Several techniques were used to measure this parameter, most common of which are the following:

- (i) light attenuation technique;

- (ii) chemical method;
- (iii) photographic studies;
- (iv) local probe techniques;

Detailed descriptions of some of these measurement techniques are provided by Landau et al. (1977).

(i) *Light Attenuation Technique*

This class of interfacial area measurement technique depends on the attenuation and scattering of γ - or X-rays and streams of atomic particles (e.g., neutrons). Radiation is transmitted differently through the gas and liquid phases. This technique provides only qualitative information about the concentration of the gas–liquid interfacial area. It does not provide information on the distribution of the phases, and the system usually requires calibration using another method, such as photographic studies.

(ii) *Chemical Method*

In this method, a reactive gas such as CO_2 is added to the air and a reactive liquid such as NaOH is added to the water before they are introduced into the channel. Chemical reaction between the added chemicals occurs at the interfacial area between the phases. This chemical reaction is well established, fast, and irreversible. The concentration of the interfacial area is obtained by chemical analysis of gas samples taken at the end of the test section. Details of this measurement technique are provided by Veteau et al. (1981).

The chemical method gives the total interfacial area in the channel but does not provide any information about the distribution of the phases. Also, this method can only be used with two-component mixtures, such as air and water, and under steady flow conditions.

Ishii (1992) indicated that the experimental set-up required for this method is cumbersome and time consuming, and depending on the experimental conditions, the errors associated with this technique can be quite significant.

(iii) *Photographic Studies*

This technique involves the determination of the interfacial area concentration from photographs of the flow taken through the transparent walls of the channel. This method provides additional information on the dispersion of the interfacial area and other parameters, such as sizes and shapes of air bubbles.

Landau et al. (1977) conducted a comparison of the above three methods for measurements of the gas–liquid interfacial area. They concluded that the chemical method is time consuming and is not suitable in cases in which the gas phase is highly dispersed in the liquid phase. For the photographic technique, Landau et al. (1977) identified the following two shortcomings:

- (i) Information obtained using this method is based on the flow conditions near the wall, which are not necessarily representative of those over the entire cross section of the apparatus.
- (ii) The photographic method involves a time-consuming data analysis.

Landau et al. (1977) recommended the use of the light-scattering technique, even though this method does not provide information about the distribution of the phases and requires calibration of the system. Ishii (1992) also indicated that the photographic technique is time consuming.

(iv) *Local Probe Techniques*

The local probe technique was developed principally by Ishii and his co-workers (Kataoka, Serizawa, Revankar, Riznic, Kocamustafaogullari). Kataoka et al. (1985) developed a theoretical formulation for measurements of the gas–liquid interfacial area

concentration using local electrical resistance probes. These probes also provide measurements of the local void fraction based on the difference in the electrical conductivity of the gas and liquid phases.

In their earlier work, Ishii and his co-worker used a two-sensor electrical resistance probe to study the interfacial area in vertical bubbly flow. This measurement technique requires assumptions about the shape of the bubbles. Revankar and Ishii (1993) proposed a theory for measurement of the local interfacial area using a four-sensor probe. The four-sensor probe technique does not require any assumptions about the shape of the interfacial surfaces. The four sensing tips of the probe are arranged such that they provide the three spatial components of the interfacial velocity.

Discussion

Recent advances in computer associated technologies helped to overcome the drawbacks associated with photographic techniques. First, horizontal- and vertical-planes of the flow can be taken simultaneously. This permits the images to be representative of the phase distribution across the entire cross section of the channel, instead of only near the wall of the tube. Second, analysis of the acquired images can be done digitally and the process can be automated. Edge detection and measurement routines are readily available, e.g., MATLAB¹³ image processing tools.

Measurement of the Interfacial Area Concentration

The image processing system used for measuring the phases interfacial area concentration is shown in Fig. 4.5. The system consists of two charge-coupled digital (CCD) cameras fitted with 25 mm/f1.2 Pentax lenses and focussed at the centre of the channel. The cameras are arranged to acquire images of the horizontal- and vertical-planes of the test tube. As shown in Fig. 4.5, a video splitter is used to combine the images so that a simultaneous view of the horizontal- and vertical-planes of the test tube

¹³ MATLAB is a sophisticated software developed by "The Math Works Inc.", which provided a flexible environment for image filtering, pixel analysis, and edge detection.

is shown on the same image. The combined images are displayed on a monitor screen and recorded on an S-VHS video recorder. The cameras are carefully positioned at the same distance from the test section (each of the cameras is mounted on an advancing mechanism).

To avoid the optical distortion caused by the curvature of the test tube, a transparent rectangular box was mounted around a portion of the test tube and filled with distilled water, thus providing a flat glass window (Fig. 4.5).

The interfacial area measurement system features high shutter speed cameras (down to 10^{-4} s), 771×492 pixels per frame, and uniform light distribution provided by two 225 mm long florescent light tubes and two layers of Mylar used as light diffusers. The lights are placed behind the test tube (opposite to the cameras). This arrangement forces the cameras to operate at a higher shutter speed, thus providing sharper images of the flow (some photographs of the flow are shown in Chapter 5).

For each test run, the flow was recorded for 1–2 min, depending on the flow regime.

4.5 Data Acquisition and Control System

The experimental test facility is fully automated. A computer-based data acquisition system was used to read the output voltage of the flow meters, the pressure transducers, and the thermocouples. The system consists of a 12-bit analog-to-digital converter and a dedicated personal computer. The radial position of the optical probe is controlled via the parallel port of the computer. Note that all instruments are wired with shielded cables to ensure a good signal-to-noise ratio.

A Quick-Basic computer program, TPDACQS.BAS, was developed for this study. It was used to read the output voltage of the instruments and to traverse the optical probes along the diameter of the test tube. The source code of this program is included in Appendix C. It should be noted that during the commissioning stage of the facility, a

number of programs were developed and used to test and to calibrate some of the instruments. These programs are similar to the subroutines of the main data acquisition program and are therefore not included in this document.

For the optical probes, the time traces were stored on a two-channel digital oscilloscope. Two different oscilloscopes were used in this study: a Nicolet oscilloscope (Model Nicolet Pro 30C – 100 MHz) was used for test runs numbered up to 64, and a LeCroy oscilloscope (Model 9310M – 300 MHz) was used for test runs numbered 65 to 106. The reason for changing oscilloscopes is that the Nicolet scope was not available during the later part of the experimental work. Both oscilloscopes have similar characteristics: both are calibrated and feature very high sampling frequency and large storage capacity. Void fraction measurements were taken for 60 experimental test runs only. Neither of the digital oscilloscopes was available to record the output signals of the optical probe for the remaining test runs (test runs numbered 107 to 136).

Data Sampling Procedure

The following are the main steps of the data sampling procedure:

- (i) Record output voltage of the flow meters, the pressure transducers, and the thermocouples. Each measurement represents the average of 5000 readings, sampled at about 1000 readings per second.
- (ii) Acquire time traces of both probes at nine equally spaced radial locations. Time traces are recorded for a period of time long enough to capture the behaviour of the flow, usually from 2 to 20 s, depending on the flow regime.
- (iii) Videotape the flow for about 1 to 2 min, depending on the flow regime. The flow is videotaped at the same time as the void fraction is being measured.
- (iv) Repeat measurements of flow rates, pressures and temperatures, i.e., repeat measurements listed in (i) above.

- (v) Collect a water sample for surface tension measurement. Note that water samples are collected for a limited number of test runs; a water sample is taken at the inlet of the test section after every 3 to 5 test runs.

Raw Data

The following data were collected for each experimental test run:

- (i) Zero readings of the air velocity transducer, the paddle-wheel sensor, and the 4 pressure transducers (first row of data in Table 4.2).
- (ii) Voltage output of the air velocity transducer, the paddle-wheel sensor, the four pressure transducers, and the three thermocouples *before* taking measurements of the local void fractions (second row of data in Table 4.2).
- (iii) Voltage output of the AVT, the paddle-wheel sensor, the four pressure transducers, and the three thermocouples *after* taking measurements of the local void fractions (last row of data in Table 4.2).
- (iv) Time traces of both probes at nine locations across the test tube diameter.
- (v) A videotape of the flow.

The raw data collected in steps (i) to (iii) for all test runs are presented in Appendix D.

Preliminary Data Analysis

The data acquisition program includes subroutines that provide preliminary reduction of the collected data. Upon completion of each test run, the program provides

the information given in Table 4.2. This information is displayed on the screen and stored in a separate file for each test run.

In Table 4.3*a*, the program provides the following:

- (i) information entered by the operator, such as lab temperature, zero readings of the instruments, and initial classification of the flow pattern;
- (ii) calculated gas and liquid superficial velocities, mass velocities and mass flow rates for both data sets. Note that data set no. 1 and data set no. 2 were taken *before* and *after* measurement of the void fraction, respectively.
- (iii) measured pressures and temperatures for both data sets; and
- (iv) difference and relative variations between the two data sets.

In Table 4.3*b*, the data acquisition program gives the measured root mean square (rms) errors and the maximum deviations. This information is obtained to ensure that there are no abnormal readings in any of the 5000 readings in the measurements taken in both data sets. Table 4.3 also includes other information about the test run, such as names of files where output signals from probes are stored.

It should also be noted that at the end of each set of test runs, the zero readings and calibration of the instruments are checked. If a discrepancy is observed, the entire set of data is discarded.

4.6 Data Reduction and Analysis

Reduction and analysis of the experimental data collected in this study are categorized as follows:

- (i) general two-phase flow parameters, e.g., mixture properties, superficial velocities of the phases and friction losses;
- (ii) void distribution and average void fraction;
- (iii) interfacial area concentration.

Details of data processing are provided in the following sections.

4.6.1 General Two-phase Flow Parameters

A number of computer programs were developed to process the experimental data. Models and equations used in the analysis are presented in Appendix A, and the source codes for some of these programs are included in Appendix C.

The main program consists of a Fortran code, TPDATRED.FOR, developed to calculate most of the two-phase flow parameters listed below. The program provides the option of processing only one experimental test run or all the test runs at the same time.

- (i) Using the calibration data given in Appendix B, the code calculates the inlet conditions of the gas and liquid phases, and temperatures, pressures and mass flow rates.
- (ii) Properties of the phases, such as density and superficial velocity.
- (iii) Average temperature, volumetric, and mass qualities of the mixture.
- (iv) Measured two-phase flow friction losses based on the static pressure measured at three locations along the test section.
- (v) Predicted friction losses using the homogeneous model, the Lockhart–Martinelli model, and the empirical correlation of Friedel.
- (vi) Predicted average void fraction using the homogeneous, the Lockhart–Martinelli and CISE correlations.

Other MATLAB routines and Excel spreadsheets were also used to calculate the predictions of the other models and correlations discussed in Chapter 3. BANDEL.M, a MATLAB routine, is used to calculate the friction losses predicted using the model of

Bandel (1973). Predictions of friction losses by other models and correlations were calculated using the TP-PRESSURE.XLS Excel spreadsheet. Predicted average void fractions using by different models and correlations were calculated using the TP-VOID.XLS Excel spreadsheet. TP-PEGIME.XLS contains calculation of parameters needed to determine the boundaries of the flow regimes.

The main results of the data processing programs are presented in Appendix E.

4.6.2 Void Fraction

The void fraction is determined using numerical processing of the output signals of the optical probes (i.e., time traces). First, the local time-averaged void fractions are determined using the classical method of histograms (Delhay, 1968, 1969). Then, the overall cross-sectional average void fraction is calculated as the area-weighted average of the measured local void fractions.

Numerical processing of the time traces is performed in three steps, with a MATLAB routine developed to carry out the tasks involved in each of these steps. The routines must be executed in the same order as presented here.

First, the SIGNALS.M routine identifies and loads the untreated signals of the upstream and the downstream probes. For each test run, 18 time traces are loaded into MATLAB workspace (two time traces for each of the nine radial positions). The routine then displays the untreated signals for the user to check for any anomalous behaviour. An example of the output of this routine is illustrated in Fig. 4.6, which shows the untreated signals of the probes for test run 68 (typical of plug flow regime). The measured voltages and the calculated preliminary results for this test run are shown in Table 4.2 and 4.3, respectively. For this test run, the output of the probes is sampled at 5 kHz for 10 s, which corresponds to 50×10^3 data points for each of the 18 time traces shown in Fig. 4.6. As indicated earlier, local void is measured at nine equally spaced locations along the diameter of the test section. The first pair of time traces, presented at the bottom of Fig. 4.6, is recorded at 2.5 mm from the bottom inner wall of the test tube. The other pairs of

time traces correspond to the other eight equally spaced (2.5 mm) locations along the diameter of the test section. The untreated signals for all test runs are stored on 3 optical disks (CDs).

Second, HISTOGRAM.M, with the time traces now loaded onto the MATLAB workspace, is used to calculate the probability density function (PDF) and draws histograms of the time traces. Figure 4.7 shows the histograms calculated for the time traces displayed in Fig. 4.6. Considering the distribution of the phases in plug flow (liquid continuously present at the bottom of the channel), the histograms shown in Fig. 4.7 clearly indicate that the amplitude of the signals is less than 0.1 V if the sensing tip of the probes is immersed in liquid (water). If the amplitude of the signal is higher than 0.1 V, the sensing-tip of the fibres is immersed in the gas phase (air). Therefore, for this test run the reference voltage (V_{ref}) used to discriminate between the presence of the gas and liquid phases around the sensing tip of the probes is taken as 0.1 V. As noted earlier, the amplitude of the signals depends on the surrounding medium (gas or liquid) and on the surface finish of the sensing tip of the probe. In addition, during the course of this experimental work, it was noted that the amplitude of the signals also depends on the distribution and concentration of the air–water interfaces. The interfaces act as light reflecting surfaces. Therefore, in this study, V_{ref} is determined for each test run using the procedure discussed above.

Third, VOID.M is a MATLAB routine that uses the reference voltage determined in the previous step to calculate the time-averaged void fraction measured by the upstream ($\alpha_{loc,u}$) and downstream ($\alpha_{loc,d}$) probes at each of the nine locations. VOID.M compares the amplitude of the output signals of the probes to V_{ref} and calculates the local void fraction as the fraction of data points with the measured voltage higher than V_{ref} . The calculated local void fractions for test run 68 are shown in Fig. 4.8. Descriptions and discussions of the distribution of the phases are given in Chapter 6. The cross-sectional average void fraction is determined by integrating the local void fraction measurements across the area of the test tube (area weighted cross-sectional void fraction). The cross-sectional area of the tube is divided into nine horizontal bands as shown in Fig. 4.2. The average void

fraction in each band is taken as the local void fraction measured within the band. For test run 68 (Fig. 4.8), the cross-sectional void fractions measured using the upstream sensor (α_u) and downstream sensor (α_d) are 0.44 and 0.47, respectively. The average cross-sectional void fraction α_{avg} is calculated as the arithmetic mean of α_u and α_d ($\alpha_{avg} = 0.455$).

As indicated earlier, the recorded output signals of the optical probes can be analyzed to obtain other useful information, such as velocities, frequencies and lengths of the liquid slugs and elongated bubbles. Analysis of the output signals of the probes for these variables will be performed in future work.

4.6.3 Interfacial Area Concentration

Measurements of the interfacial area concentration were obtained using the recorded videotapes of the flow. A frame grabber was used to load segments of the videotaped images on a PC computer (MIRO-Video DC30). As the system features a hardware image compression system, it was possible to load sequences of the images, i.e., consecutive images of the flow. The images were loaded in gray scale with 640 X 480 pixels resolution.

Detailed analysis of the interfacial area measurements will be performed in future work. The image processing will be performed using the MATLAB image processing tools. A MATLAB routine will be developed to process the large number of frames in a consistent manner and in a short period of time.

The procedure used to perform a preliminary analysis of the acquired images is as follows: (i) trace the border of the tubes; (ii) remove area outside the tubes; (iii) select area inside the tubes as area of interest; (iv) use edge detection routine to trace the interfaces.

The edge detection routine used in the preliminary analysis requires further development. The scope of the present work is limited to the analysis of the stratified flow test runs.

4.6.4 Expected Experimental Uncertainties

The uncertainties in the measured quantities are given in Table 4.1. The complete analysis is given in Appendix B.

Table 4.1: Estimates of the uncertainties in the measurements.

Quantity measured	Instrument used	Estimated Uncertainty
Water volume flowrate	Paddlewheel sensor	± 0.008 l/s
Air mass velocity	Air velocity transducer	± 1.23 kg/m ² ·s
Air inlet Pressure	Celsco VR-series (0–690 kPag)	± 1.44 kPa
Pressure; at 1 (l/d)	Omega PX26 (0–100 kPag)	± 0.17 kPa
Pressure; at 410 (l/d)	Omega PX26 (0–35 kPag)	± 0.13 kPa
Pressure; at 540 (l/d)	Omega PX136 (0–7 kPag)	± 0.011 kPa
Temperatures	Type “T” thermocouples	$\pm 0.5^{\circ}\text{C}$
Average void fraction	Twin-needle optical probe	$\sim 5\%$
Air–water interfacial area	CCD cameras (simultaneous vertical- and horizontal-planes of the test section.	0.4 mm ² (1x1 pixels).

4.7 Summary and Discussion

This chapter has provided descriptions of the test facility and the instrumentation used in this experimental study. The measurement techniques and procedure used in the analysis of the two-phase flow parameters are also presented and discussed. Methods used to calibrate the instruments and estimate the expected uncertainties associated with the measured parameters are given in Appendix B.

As discussed in Chapter 1, Collier and Thome (1994) indicated that often when a large number of experimental test runs are collected, the easily measured data points

usually tend to dominate at the expense of those for difficult test conditions. This can compromise the accuracy of the measurements. In this study, a relatively small number of data points were collected, however, great care was taken in the measurement and analysis of each test run. For example, with a manageable size data bank, V_{ref} was determined for each test run.

Zero Readings (Volts)					
AVT Trans.	PW Sensor	Selesco P. Trans.	Omega1 P. Trans.	Omega2 P. Trans.	Omega3 P. Trans.
0.0188	-0.0052	0.0199	0.0468	-0.0393	-0.1300

Measured Data (Volts)									
	AVT Trans.	PW Sensor	Selesco P. Trans.	Omega1 inlet	Omega2 Middle	Omega3 Outlet	Water Temp.	Air Temp.	Mix. temp
Set #1	1.071	3.534	4.717	0.662	0.387	0.030	22.2	16.8	16.7
Set #2	1.067	3.534	4.726	0.647	0.382	0.019	22.4	16.8	16.6

Table 4.2: Typical data collected (test run 68).


```

Test tube I.D.: 0.0254  m                               Experiment no.:    68
                                                         Exp. data part:  1
=====

Data stored on disk no  :      2    Paddle wheel zero read: -.0052 (V)
Video tape counter      :      2    Celesco zero reading  : 0.0199 (V)
Lab. Temperature(deg.C.): 22.00    OMEGA15 zero reading  : 0.0468 (V)
Water surf. tension mN/m: 73.16    OMEGA05 zero reading  : -.0393 (V)
AVT zero reading (V)    : 0.0188    OMEGA01 zero reading  : -.1300 (V)
-----

Mixture flow pattern    : Plug Flow.
-----

Air / Water phases      Set #1    Set #2    Diff.    % Var. Average
-----
Air sup. velocity (m/s) 1.939E+01 1.936E+01 0.0335  0.2  1.938E+01
Water sup. vel. (m/s)   8.558E-01 8.558E-01 0.0001  0.0  8.558E-01
Air mass vel. (kg/m2.s) 2.467E+01 2.458E+01 0.0887  0.4  2.462E+01
Water mass vel(kg/m2.s) 8.541E+02 8.542E+02 0.0659  0.0  8.542E+02
Air mass flow (kg/s)    1.250E-02 1.245E-02 0.0000  0.4  1.248E-02
Water mass flow (kg/s)  4.328E-01 4.328E-01 0.0000  0.0  4.328E-01
-----

Pressure                Set #1    Set #2    Diff.    %    Average
                        (kPa)     (kPa)     (kPa)    Var.  (kPa)
-----
Air press. @ trans.     541.471  542.495  1.025    0.2   541.983
Test tube inlet p.      12.783   12.478   0.305    2.4   12.631
Test tube mid. prs      2.991    2.956    0.035    1.2   2.973
Test tube exit prs      0.232    0.215    0.017    7.6   0.224
-----

Temperature             Set #1    Set #2    Diff.    %    Average
                        (deg.C.)  (deg.C.) (deg.C.) Var.  (deg.C.)
-----
Air inlet temp. :      22.23   22.35    0.11     0.5   22.29
Water inlet temp.:      16.80   16.79    0.01     0.1   16.80
Mixture exit temp.:    16.65   16.62    0.03     0.2   16.64
-----

COMMENTS:
-----

=====
This file: B:RUN68.DAT      Date: 11-15-1998      Time: 12:52:56
=====

```

Table 4.3a: Sample output of data acquisition program.

```

Test tube I.D.: 0.0254 m                      Experiment no.: 68
                                           Exp. data part: 2
=====

Data set no.: 1                      Number or readings taken: 5000
=====

Instrument used                      Average      rms      Max. dev.
                                   (Volts)    (Volts)   (Volts)
-----
Air velocity transducer      :      1.0708      0.0034      0.0330
Paddle wheel sensor         :      3.5337      0.0089      0.0221

Celesco: Pressure at AVT    :      4.7174      0.0014      0.0054
OMEGA15: Test tube inlet press :      0.6615      0.0108      0.0290
OMEGA05: Test tube middle press:      0.3871      0.0550      0.1281
OMEGA01: Test tube exit press :      0.0303      0.1183      0.2823
=====

Data set no.: 2                      Number or readings taken: 5000
=====

Instrument used                      Average      rms      Max. dev.
                                   (Volts)    (Volts)   (Volts)
-----
Air velocity transducer      :      1.0670      0.0023      0.0316
Paddle wheel sensor         :      3.5340      0.0036      0.0118

Celesco: Pressure at AVT    :      4.7263      0.0015      0.0039
OMEGA15: Test tube inlet press :      0.6468      0.0071      0.0217
OMEGA05: Test tube middle press:      0.3821      0.0395      0.0868
OMEGA01: Test tube exit press :      0.0185      0.1195      0.3087
=====

Local void fraction measurements; Double-sensor Fibre-optic Probe.
=====

Digital Oscilloscope sampling at: 500.00E+01 Hz, for: 10.0 sec
-----

Probe pos.   Front sensor   Back sensor
-----
1 ...      WAVE 0.WFT      WAVE 1.WFT
2 ...      WAVE 2.WFT      WAVE 3.WFT
3 ...      WAVE 4.WFT      WAVE 5.WFT
4 ...      WAVE 6.WFT      WAVE 7.WFT
5 ...      WAVE 8.WFT      WAVE 9.WFT
6 ...      WAVE 10.WFT     WAVE 11.WFT
7 ...      WAVE 12.WFT     WAVE 13.WFT
8 ...      WAVE 14.WFT     WAVE 15.WFT
9 ...      WAVE 16.WFT     WAVE 17.WFT
=====
Part 2 of file: B:RUN68.DAT      Date: 11-15-1998      Time: 12:52:57

```

Table 4. 3b: Continue (sample output of data acquisition program).

Figure 4.1: Schematic diagram of the two-phase flow experimental facility.

Figure 4.2: Experimental setup for void fraction measurement.

Figure 4.3: Typical time traces from photodetectors.

(a) Clean and straight cut.

(b) Rough cut with chipped edge.

Figure 4.4: Sensing tip of the optical probe.

Figure 4.5: Experimental setup for interfacial area measurement.

Figure 4.6: Untreated output signals of the optical probes for test run 68 (plug flow regime).

Figure 4.7: Calculated histograms for the optical signals shown in Figure 4.6 (test run 68).

Figure 4.8: Local void fraction measured using upstream and downstream sensors.

Chapter 5

Observed Flow Patterns and Pattern Transitions

5.1 Introduction

This chapter has the following objectives:

- (i) to present the methodology developed to classify the flow patterns and the criteria used to distinguish between them;
- (ii) to present and to discuss the flow patterns observed in this experimental investigation;
- (iii) to compare the experimental results with published semitheoretical and empirical flow pattern maps;
- (iv) to identify and to discuss flow pattern transition mechanisms (qualitative analysis only); and
- (v) to propose a horizontal two-phase flow regime map and identify its range of applicability;

5.2 Patterns Identification and Classification

One of the most important and challenging tasks in two-phase flow studies is proper classification and identification of the observed flow regimes. The problem is actually well defined and requires the following: (i) to determine the most relevant and convenient classification of the various types of interface distribution observed in practice; and (ii) to develop an objective method to identify the prevailing flow pattern and classify it with one of the defined groups of patterns.

In the following sections, some of the problems associated with the classification and identification of two-phase flow patterns are discussed. The approach taken to identify the prevailing flow patterns in the present investigation is presented.

5.2.1 Classification of Flow Patterns

Analogous to single-phase flow, which is divided into laminar and turbulent regimes, the various types of interface distributions in two-phase flow are grouped into a number of flow patterns. This is an essential step towards a better understanding of the two-phase flow phenomena. The literature survey presented in Section 2.3 shows that some disparity exists in the description and classification of the flow patterns suggested by different authors. This makes it difficult to compare results and to further our understanding of the two-phase flow phenomena.

To provide a relevant classification of two-phase flow, it is important to remember the main reason for dividing the flow into a number of regimes. As in single-phase flow, it is extremely difficult to develop a single model or correlation capable of describing the different conditions of the flow. The flow is divided into several regimes, and models are developed for each regime. These regime-specific models are expected to provide better predictions of the flow parameters. As discussed in Section 2.4, two-phase flow has been classified into a number of regimes based mainly on the distribution and relative velocities of the phases.

Transition Regions

A major difficulty in providing a clear and consistent classification of the flow is due to the fact that the transition from one regime to another occurs over a certain range of flow conditions. Many investigators have indicated that there is no clear-cut transition between the regimes and suggested that the boundaries be represented by a band (covering certain range of flow conditions) rather than a line. However, only Breber et al. (1980) gave clear indications of transition regions between the regimes. These regions

are analogous to the laminar-turbulent transition region considered in single-phase flow (usually when Reynolds number is in the range of 2000 to 4000).

It is important to note that showing the transition regions on the flow regime map provides important information to analysts. For example, if the transition regions are not considered, the use of different models at the transition boundaries can lead to the possibility of a discontinuity in the predicted parameters, such as heat transfer coefficients and two-phase flow friction multipliers. However, if the transition regions are considered and the prevailing regime is predicted in these regions, the discontinuity can be avoided by using a smoothing function such as the one given by the following equation.

$$\Phi_{tr} = (\Phi_1^n + \Phi_2^n)^{1/n} \quad (5.1)$$

where ϕ_1 and ϕ_2 are the values of the parameter calculated for the adjacent flow patterns, and ϕ_{tr} is the value of the weighted parameter at the transition region. Usually, good results are obtained with $n = 3$ (Rogers, 1989).

Breber et al. (1980) suggested that the two-phase flow friction multipliers and heat transfer coefficients be linearly prorated between the regimes. The flow regime map used in the CATHENA¹⁴ code considers transition regions and employs a weighted averaging procedure if the predicted regime is in the transition regions.

Intermittent (Plug and Slug) Flow Regime

For horizontal flows, the most widely used classification of the flow regimes is the one proposed by Baker (1954) and discussed in Section 2.3. Baker (1954) proposed that the flow be classified in the following regimes: stratified smooth, wavy, plug, slug, annular, and dispersed bubble flow. Taitel and Dukler (1976a) have proposed a variation of this classification. They considered the plug and slug flow as different conditions of the intermittent flow regime. Based on observations obtained in this study, as well as

¹⁴ CATHENA: State-of-the art thermalhydraulics code used in the safety analysis of CANDU reactors.

descriptions provided in the literature for these regimes, it is unrealistic to expect that the same model can be developed to describe the conditions of the plug and slug regimes. Based on present observations, the following arguments are given in support of considering plug and slug flows as separate regimes:

- (i) Plug flow occurs at relatively lower mass velocity of the gas phase, the shape of elongated bubbles is well defined, and the bubbles are transported by the liquid phase.
- (ii) Slug flow is characterized by randomly shaped and highly aerated liquid slugs being pushed along the conduit by the faster moving gas phase. As indicated earlier, slug flow is undesirable in practical applications because of its severe pressure oscillations.
- (iii) As will be discussed in Chapter 6, the ranges of measured void fraction and friction losses are different for these regimes. For plug flow, the measured friction losses are less than 1 kPa/m and $\alpha < 0.50$ for most observations, while for slug flow the frictional losses reached up to 6 kPa/m and $\alpha > 0.40$.

Stratified Smooth and Wavy Flow Regimes

In this study the stratified smooth and the wavy flow regimes are consolidated in one regime and identified as the stratified wavy flow regime. The following arguments are given in support of consolidating these regimes:

- (i) Stratified smooth flow is difficult to establish, especially in small diameter tubes (<50 mm) or in long test sections, since small perturbations can produce waves. Even small variations in the phase inlet conditions result in wave initiation. The literature indicated that with a small angle of inclination of the conduit, wavy flow was observed over a wider range of flow conditions than in horizontal flow. Based on the above arguments, it is probable that stratified smooth flow occurs only under laboratory conditions.

- (ii) The experimental studies of França and Lahey (1992) and Nitheanandan (1993) reported no observations of the stratified smooth flow conditions. In the experimental data used by Mandhane et al. (1974), stratified smooth flow represents less than 4% of the flow patterns observed in conduits less than 50 mm in diameter.
- (iii) In addition, from a modelling point of view, little is gained from this subdivision of the flow. The characteristics of the stratified smooth and wavy flows are quite similar. The only differences are that wavy flow occurs at slightly higher mass velocity of the gas phase and that because of the waves, the gas–liquid interfacial area is higher in wavy flow. This increase in the interfacial area can be modelled using a waviness coefficient, which can be calculated as a function of the average frequency and amplitude of the waves.
- (iv) Hubbard and Dukler (1966) and Pearce (1982) also indicated that little useful information would be lost and that the additional work to differentiate between these regimes would not be merited.

Summary

In summary, this study proposes that the configurations assumed by gas–liquid flows in small diameter (<50 mm) horizontal tubes be classified in the following major flow patterns: stratified wavy, plug, slug, annular and dispersed bubble flow regimes. Note that the plug and slug flows are considered as separate regimes, while stratified smooth and wavy flows are consolidated into one flow regime identified as stratified wavy flow. In addition, the flow classification should allow for transition regions between the regimes.

5.2.2 Pattern Identification

Various techniques were proposed to identify and to categorize the flow patterns. Unfortunately, as discussed in Section 2.3, these techniques have had only limited

success, as many discrepancies are still found between the proposed pattern classifications and flow regime maps.

In the present study, a methodology was developed to identify the flow patterns. This methodology is based on video techniques and uses detailed descriptions of the flow patterns, including their distinctive features and clear criteria used to distinguish between the regimes.

5.2.3 Methodology Used to Identify the Flow Patterns

The following summarizes the methodology used in this study to identify the prevailing flow pattern:

- (i) Detailed descriptions of the distinctive features of each flow pattern were set up. These features were then used to establish clear criteria that could be used to distinguish between different flow patterns. These criteria are outlined in the following subsection.
- (ii) Detailed descriptions of the flow behaviour over the entire length of the test section were reviewed. These descriptions were noted when the measurements were taken.
- (iii) Two investigators viewed the recorded videos, separately and independently. Their classifications were then compared and the regimes identified differently were viewed again in slow motion. The differentiating features of the regime were identified and the regime was classified.
- (iv) The observations were plotted on a flow regimes map, and data points located near the transition boundaries were identified. The recorded videos for these test runs were played back in slow motion (down to one frame at the time) to confirm the regime classification.

5.2.4 Criteria Used to Identify the Flow Patterns

This section provides descriptions of the distinctive features of each flow pattern along with the criteria used to distinguish it from other flow patterns. Figures 5.1 to 5.4 show typical distribution of the phases in each of the observed flow patterns.

Stratified Wavy Flow Pattern

In this regime, which includes cases of stratified smooth and wavy flows, the liquid flows along the bottom of the test tube while the gas flows on top with very few or no liquid droplets (Fig. 5.1). The liquid free surface is either smooth (stratified smooth flow) or wavy (stratified wavy flow). The waves are formed by the action of the gas flowing over the liquid film.

The stratified wavy flow regime is distinguished by (i) an almost complete separation between the phases, with the possibility of a few liquid droplets entrained in the gas, and the waves may be aerated; and (ii) waves that do not reach the top of the tube and travel at a velocity lower than the average velocity of the gas, at any point in time, the waves do not block the flow area of the gas phase.

The first condition distinguishes stratified wavy from the annular flow regime. As the waves build up, the nature of the flow becomes difficult to distinguish from the slug regime. The second condition is established to distinguish stratified wavy flow from slug flow.

Plug Flow Pattern

The plug flow is characterized by elongated bubbles flowing along the top of the tube in a continuous liquid flow. The bubbles have similar characteristics to those depicted in Fig. 5.2.

The following criteria were established to distinguish plug flow from slug flow:

- (i) elongated bubbles move at essentially the same velocity as the liquid;
- (ii) the gas–liquid interface below the bubbles is relatively stable, indicating a small difference between the velocities of the phases at the interface, i.e., elongated bubbles are transported by the liquid phase.

The distinction between plug and slug flow patterns is also based on the shape of the bubbles and the thickness and waviness of the film below the bubbles.

Slug Flow Pattern

The slug flow is characterized by turbulent liquid slugs and sprays vigorously transported down the channel by the gas flow at a higher velocity than the mean velocity of the remaining liquid. Figure 5.3 shows a typical distribution of the phases in slug flow pattern (stratified wavy like flow followed by a liquid slug).

The distinctive features of slug flow are

- (i) liquid slugs are turbulent, highly aerated, and occupy the entire cross-sectional area of the tube, i.e., blocking the flow area of the gas;
- (ii) the liquid slugs are separated by a stratified wavy-like flow pattern, i.e., a rough and irregular liquid layer below the flowing gas separates the highly aerated and turbulent liquid slugs; and
- (iii) following the passage of the liquid slugs, the liquid deposited on the top wall of the tube drains down the walls of the tube, i.e., the top wall of the tube is only intermittently wet.

The first two criteria were used to distinguish between slug and stratified wavy flow regimes, while the third criterion discriminated between slug and annular regimes.

Annular Flow Pattern

In this regime, the liquid flows as a wavy film covering the entire wall of the tube, while the faster moving gas flows in the core of the tube. The gas usually carries small

droplets of liquid dispersed in the form of a mist. Because of gravity, the liquid film at the bottom of the tube is usually thicker than at the top. Figure 5.4 shows a typical distribution of the phases in annular flow pattern.

The distinguishing features of this regime are (i) that liquid is swept up around the entire perimeter of the tube, and (ii) that the liquid phase does not block the entire cross-sectional area of the test tube.

The first criterion distinguishes between annular and stratified wavy flow patterns, while the second criterion distinguishes the annular flow pattern from the slug flow.

Dispersed Bubble Flow Pattern

This dispersed bubble flow regime is characterized by a dispersion of discrete bubbles flowing in a turbulent liquid phase. Therefore, this regime is distinguished by the random movement of the air bubbles in the radial direction of the test section. This condition is considered to be the main defining feature of this regime, since it exhibits the turbulent nature of the liquid phase. For the dispersed bubble flow regime to occur, the liquid turbulent forces must overcome the buoyancy forces keeping the air bubbles near the top of the channel. Note that no photograph of this flow pattern is presented here because, as will be discussed later in Section 5.3.2, this flow regime was not observed in this study.

5.3 Experimental Results

5.3.1 Range of Operating Conditions

A total of 136 test runs were performed, 46 of which were used to test the experimental setup or to calibrate the instruments. The range of operating conditions for the remaining 90 experimental test runs is given in Table 5.1. The experiments were conducted in a 25.4-mm horizontal test section. The total length of the test tube is about 13.7 m, which gives an l/d ratio of about 540. The large l/d ratio ensures that the flow is fully developed.

Table 5.1: Range of operating conditions of the present experimental study.

Operating conditions	Range
Air mass velocity (G_g)	0.12 – 40.9 kg/m ² ·s
Water mass velocity (G_l)	0.012 – 1.17×10 ³ kg/m ² ·s
Gas superficial velocity (J_g)	0.1 – 30 m/s
Liquid superficial velocity (J_l)	0.002 – 1.17 m/s
Operating pressure ¹⁵ ($P_{avg.}$)	101.5 – 205 kPa (abs.)
Avg. cross-sectional void fraction (α)	0.02 – 0.96
Temperature of the mixture (T_m)	19 – 23 °C
Water surface tension (σ)	69.6 – 72.7 mN/m

5.3.2 Observed Flow Patterns

The processed experimental data is presented in Table E-1 in Appendix E. This table gives the air and water superficial velocities and descriptions of the observed flow regime for each of the 90 experimental test runs.

For each test run, the prevailing flow regime was observed along the entire length of the test tube (about 13.7 m, $l/d = 540$). Detailed descriptions of these observations were noted. The flow behaviour was also videotaped at an axial location (l/d) of ~ 410 . Videos of horizontal- and vertical- planes of a 150-mm long portion of the tube were acquired simultaneously as described in Section 4.4.3.

In this study, the flow pattern identification is based on a combination of visual observations of the hydraulic features of flow and rigorous examination of the acquired videotapes. The flow regimes were classified using the methodology developed in Section 5.2.3, along with the pattern identification criteria discussed in Section 5.2.4. The observed flow patterns were identified as stratified wavy, plug, slug, or annular flow.

¹⁵ The operating pressure term refers to the absolute wall pressure measured at the inlet of the test section.

The number of observations for each flow pattern is given in Table 5.2. Note that Table 5.2 also contains the number of test runs for each flow pattern of the data collected by Mandhane et al. (1974), which will be discussed later in this section.

Table 5.2: Number of observations for each flow pattern; present study and data used by Mandhane et al. (1974).

Regime	Present Work		Mandhane et al. (1974)			
			All data		d < 50 mm	
Annular flow	18	20%	1746	29%	1512	36%
Plug flow	33	37%	758	13%	422	10%
Slug flow	31	34%	2020	34%	1431	34%
Stratified wavy flow	8	9%	783	13%	585	14%
Dispersed bubble flow	0	0%	231	4%	94	2%
Stratified smooth flow ¹⁶	-	0%	397	7%	162	4%
Total	90	100%	5935	100%	4206	100%

As indicated earlier, stratified smooth flow regime was not observed in this study. In addition, none of the flow observations was classified as dispersed bubble flow pattern. The occurrence of the dispersed bubble flow pattern in small diameter horizontal tubes is discussed later in this section.

It should be noted that simultaneous viewing of the recorded horizontal- and vertical- planes of the flow proved to be a useful tool in identifying the flow patterns as well as the pattern transition mechanisms. For example, the top view (horizontal-plane) of the tube gave a clear indication of the onset of the entrainment and deposition mechanism as the water droplets deposited on the upper wall of the tube could be viewed easily. The entrainment and deposition mechanism governs the transition from the

¹⁶ In this study, stratified smooth flow is consolidated with stratified (wavy) flow. In any case, none of the observations could have been classified as stratified smooth flow based on the description given for this regime in Section 2.2.

stratified wavy to the annular flow pattern. The transition mechanisms of the flow patterns are discussed in Section 5.5.

Statistical analysis of the time traces of both optical probes at nine equally spaced locations across the pipe diameter was also used in identifying certain flow patterns. The untreated time traces and their PDF for stratified wavy, plug, slug, and annular flow patterns are shown in Fig. 5.5 to Fig. 5.8. The upstream sensor is represented by the solid darker line and the downstream sensor is represented by the broken lighter line.

Figure 5.5 exhibits typical analog output of the two sensors for the stratified wavy flow regime. The PDF plot shows that for both sensors the output voltage was less than 0.3 V when the sensors were immersed in liquid (near the bottom of the tube). The output voltage is higher than 0.3 V when the sensors were immersed in air (near the top of the tube). The signals at the third measurement location from the bottom wall of the tube (7.5 mm) showed the probes detecting the waves. As the probe moved upward, only high amplitude waves were detected by the probes.

Figure 5.6 shows the output of the probes for the plug flow regime. Near the top of the tube, the signals are plotted for a shorter time to illustrate some details. The signals show that only water flowed at the bottom of tube, while in the upper half of the tube the flow was intermittent (showing the elongated bubbles). The PDF plot for this test run shows that the reference voltage used to distinguish between air and water was 0.3 V.

The untreated signals for slug flow regime are shown in Fig. 5.7. The signals show the liquid layer like a “carpet” at the bottom of the tube with the liquid slugs detected near the top of the tube (the liquid blocks the flow area of the air). This criterion was used to distinguish between stratified wavy and slug flow regimes. The signals near the top wall of the tube can be used to determine the average length and frequency of the liquid slugs. Note that for the test run shown in Fig. 5.7, V_{ref} is 0.15 V for the upstream sensor ~ 0.03 V for the downstream sensor.

Figure 5.8 shows the untreated signals and their PDF for the annular flow regime. For this test run, V_{ref} is 0.3 V for both optical sensors. The PDF plot shows that with the exception of the signals recorded near the bottom of the tube, the measured voltages are spread over a wider range than in the other flow regimes (Figures 5.5–5.7). Figure 5.8 also shows that more liquid is present at the bottom of the tube.

Figures 5.5–5.8 suggest that the time traces and their PDF can be used to distinguish between typical plug, slug, stratified wavy, and annular flow patterns. Unfortunately, this method could not be used with confidence to classify the observed transitional flow, i.e., test runs at or near the transition boundaries. Figure 5.9 illustrates the PDF plots for four test runs identified in the transition regions between the flow regimes.

Dispersed Bubble Flow

Based on the identifying features of flow patterns discussed in Section 5.2.4, none of the 90 experimental test runs was classified as dispersed bubble flow pattern. Attempts were made to establish these regimes by adjusting the flow rates of the phases into the test section but without success. A discussion for the probable reasons for this follows.

The literature indicates that dispersed bubble flow occurs at relatively higher liquid superficial velocities (J_l) than those measured in this study. The highest value of J_l measured in this study is 1.17 m/s, while the flow regime map of Mandhane et al. (1974) and the transition models of Weisman et al. (1979) predict the dispersed bubble flow occurs at J_l higher 4.25 m/s and 1.92 m/s, respectively.

It is also probable that dispersed bubble flow cannot exist in small diameter horizontal or inclined channels (less than 50° from the horizontal). This argument can be supported by the very small number of dispersed bubble flow observations reported in the literature. As shown in Table 5.2, less than 4% of the data collected by Mandhane et al.

(1974)¹⁷ was classified as dispersed bubble flow. For conduit sizes less than 50 mm, dispersed bubble flow represents less than 2% of observations. In addition, the experimental work of França and Lahey (1992) and Nitheanandan (1993) reported no observations of the dispersed bubble flow. França and Lahey (1992) used an air–water system in 19-mm horizontal tubes. Nitheanandan (1993) studied condensing steam flow in 13.4-mm horizontal and slightly inclined tubes (within $\pm 10^\circ$).

In the recent experimental study of Ewing et al. (1999) of an air–water adiabatic mixture in 19-mm diameter horizontal tube, less than 5% of the observations were identified as dispersed bubble flow. The definition used by Ewing et al. (1999) for this regime is not consistent with the one considered in this study. Based on the description and the photograph provided by Ewing et al. (1999) for this flow, it is probable that what was observed is annular flow with a highly aerated liquid film and concentrated wisps of liquid in the core of the tube. This type of flow resembles the homogeneous flow regime observed by Pearce (1982) at high superficial velocities of both phases. It should also be noted that Ewing et al. (1999) indicated that dispersed bubble flow was difficult to establish, and they reported observations of small bubbles coalescing to form plug flow in the flow development section of the test tube.

Bubble coalescence near the top of the tube was also observed in this study. In 3 of the 33 test runs classified as plug flow, the air bubbles were small and nearly spherical. The distribution of the phases in these test runs resembled that of bubble flow. However, a closer examination of the videotape taken for these test runs showed that the bubbles travelled near the top of the pipe in the form of “scalloped” bubbles and at the same velocity as liquid (Fig. 5.10), thus indicating that the liquid turbulent forces were not large enough to overcome the buoyancy forces keeping the air bubbles near the top wall of the conduit.

¹⁷ The data used by Mandhane et al. (1974) were collected in 13- to 150-mm pipes. Over 70% of the data were collected in 13- to 50-cm pipes, and about 20% of the data were for air–water systems.

In summary, dispersed bubble flow is more typical of vertical¹⁸ and steeply inclined channels (more than 50° from the horizontal). In horizontal and slightly inclined tubes, gravity forces cause the bubbles to rise and coalesce at the top of the pipe to form elongated bubbles, i.e., plug flow. Therefore, the dispersed bubble flow regime was not considered further in this study.

Experimental Data of França and Lahey (1992)

To further demonstrate the correctness of the classification of the flow patterns observed in this study, the present data are compared with the observations of França and Lahey (1992) in Fig. 5.11.

The data are plotted on a log-log scale with the gas superficial velocity (J_g) as the abscissa and the liquid superficial velocity (J_l) as the ordinate. These coordinates are chosen for this comparison because França and Lahey (1992) included values of the superficial velocities with their classification of the flow regimes. The experimental data of França and Lahey (1992) were for horizontal air–water flow in a 19-mm i.d. pipe at near atmospheric conditions, similar to the present conditions. The flow was classified based on visual observations of the phase distribution, observations of the pressure and pressure-drop fluctuations, and high-speed photography.

Figure 5.11 shows reasonable agreement between both sets of experimental data; similar flow patterns are located in similar areas of the map. The only notable differences are in the limits of the locations of the annular and slug flow regimes. In the present study, both annular and slug regimes were observed at slightly higher J_g . For annular flow, the discrepancy may be explained by the difference in the pipe diameter used in the studies (25.4 mm in this study compare to 19 mm used by França and Lahey, 1992). In addition, França and Lahey (1992) did not provide tabulated data for all their annular

¹⁸ Even in vertical flows, dispersed bubble flow regime is difficult to establish in small diameter pipes. Rogers (1985) provided a review on the conditions required for the existence of this regime in vertical flows.

flow observations. Their flow regime map (Fig. 3 in their publication) shows more annular flow observations, which indicate better agreement with the present data.

For plug–slug transition, the discrepancy between both data may be due to differences in the method used to identify the plug flow regimes. França and Lahey (1992) compared J_g to the gas superficial velocity of what they identified as “hypothetical homogeneous” liquid driven flow defined as $J_{gh} = \alpha J$. Data points with $(J_g - J_{gh}) < 0.05$ m/s were classified as plug flow. França and Lahey (1992) gave no justification for their choice of 0.05 m/s as the criterion to identify the plug flow regime.

5.4 Comparison with Published Flow Pattern Maps

In this section, the observed flow patterns are compared with some of the well-established flow regimes in the literature, such as Mandhane et al. (1974), Taitel and Dukler (1976a), Weisman et al. (1979), and Breber et al. (1980). A number of other horizontal flow regime maps (Baker (1954), Hoogendoorn (1974), Spedding and Nguyen (1980), Kadambi (1982), and Lin and Hanratty (1987b)) are discussed in the literature survey in Chapter 2. The locations of regime boundaries for these maps could not be accurately reproduced. Therefore, these maps are considered only for qualitative comparison in the following subsections.

5.4.1 Mandhane et al. (1974)

In Fig. 5.12, the observed flow patterns are compared with the horizontal flow regime map of Mandhane et al. (1974). The data points indicate the present data, and the solid curves show the flow regime map of Mandhane et al. (1974). Bearing in the mind that stratified smooth and wavy flow patterns were consolidated in this study, relatively good agreement can be seen for the transition from stratified wavy to plug, slug, and annular flow regimes. In addition, at low values of liquid superficial velocity ($J_l < 0.5$ m/s), the transition between slug and annular flow is well predicted by the map of Mandhane et al. (1974). However, at $J_l > 0.5$ m/s, the map of Mandhane et al. (1974)

predicts that annular flow occurs at a higher gas superficial velocity (J_g) than observed in this study.

The largest discrepancy between the observed flow patterns and the map of Mandhane et al. (1974) is in the transition between plug and slug flow regimes. In the present study, plug flow was observed at higher gas superficial velocities than predicted by the map of Mandhane et al. (1974). As could be clearly seen in Fig. 5.12, the present data indicate that at $J_l = 1$ m/s, the transition from plug to slug flow occurs at J_g equal 1.25 to 1.75 m/s. The map of Mandhane et al. (1974) predicts that this transition will occur at $J_g = 0.7$ m/s. This discrepancy can be explained by differences in the definition of plug flow. Mandhane et al. (1974) identified plug flow as bubble and elongated bubble flows. Out of the 1178 air–water data points used to produce the map of Mandhane et al. (1974) only 12 observations (about 1%) were classified as plug flow. This indicates the restricted definition used by Mandhane et al. (1974) to classify the plug flow regime.

5.4.2 Taitel and Dukler (1976a)

The present data are compared with the semitheoretical models of Taitel and Dukler (1976a) in Fig. 5.13. Again, the points represent the present data. The solid curves are the predictions of the transition models of Taitel and Dukler recalculated for an air–water system in a 25.4-mm diameter horizontal tube at near standard atmospheric conditions (20°C and 101.3 kPa pressure). As suggested by Taitel and Dukler (1976a), the interfacial friction factor (f_i) is assumed equal to the gas-wall friction factor (f_{wg}), and is calculated using Eq. 3.16. The transition models considered in this comparison are represented by curves A and B in Fig. 2.8. The data are plotted in Fig. 5.13 using the dimensionless parameters F and X as suggested by Taitel and Dukler (1976a). Derivations of the equations used to plot the transitions curves of Taitel and Dukler are given in Appendix A.

Figure 5.13 shows reasonable agreement in the transition from stratified wavy to intermittent (plug and slug) flow regimes. The stratified wavy to annular transition

proposed by Taitel and Dukler (1976a) predicts annular flow to occur at lower values of F than the present data, especially at higher values of the Lockhart–Martinelli parameter ($X > 0.1$). The present data show that the transition to annular flow occurs at $F = 1.5$ over a wide range of X . The model of Taitel and Dukler (1976a) predicts that for X values ranging from 0.01 to 1.6, the transition to annular flow occurs at F values ranging from 1.5 to 0.16. This discrepancy may be due to the fact that Taitel and Dukler (1976a) assumed that this transition is initiated by Kelvin–Helmholtz wave instabilities, while the present study shows that the transition is actually governed by entrainment and deposition mechanisms. The gas mass velocity required to initiate wave break-up and subsequent entrainment and deposition of the water droplets is expected to be higher than required for wave initiations.

Breber et al. (1980) compared the transition models of Taitel and Dukler (1976a) with condensation data from 10 different sources (all for condensing flow and covering a wide range of physical properties). They indicated that about 20% of the wavy flow data were predicted as annular flow by the Taitel and Dukler (1976a) models, especially at high values of the Lockhart–Martinelli parameter ($X > 0.1$), supporting the above observation.

Taitel and Dukler (1976a, 1980) suggested that the transition between annular and intermittent flow regimes occurs at constant values of X in the range of 0.7 – 1.6. The original models of Taitel and Dukler (1976a) predicts this transition to occur at $(h/d) = 0.5$, which corresponds to $X = 1.6$. Later, Dukler and Taitel (1986) indicated that the transition may occur at a lower $(h/d) = 0.35$, which corresponds to $X = 0.7$. Figure 5.13 shows that the observed transition between annular and slug flow regimes occurs in the range suggested by Taitel and Dukler (1976a, 1980). It is interesting to note that, in the present study the annular flow regime occurs at a higher gas superficial velocity than that predicted by Taitel and Dukler (recall that in horizontal flow $F = J_g^*$; see Section 2.4.7).

In summary, good agreement was found between the observed flow patterns and the models of Taitel and Dukler (1976a, 1980) for the transition between stratified wavy

and intermittent regimes and between annular and intermittent regimes. However, the models of Taitel and Dukler (1976*a*, 1980) predict that the transition between annular and stratified wavy flow regimes occurs at lower gas superficial velocities than those observed in the present study.

5.4.3 Weisman et al. (1979)

The empirical correlations proposed by Weisman et al. (1979) for the transitions between the flow patterns were recalculated for air–water flow in a 25.4-mm diameter horizontal tube at standard atmospheric conditions (20°C and 101.3 kPa pressure). The calculated curves are compared with the results of the present investigation in Fig. 5.14. Derivations of the equations used to plot the transition curves are given in Appendix A. Note that following the suggestion of Taitel and Dukler (1976*a*), Weisman et al. (1979) consolidated the plug and slug flow regimes.

Figure 5.14 shows good agreement between the present data and the transition models of Weisman et al. (1979) for the transition between stratified wavy and intermittent regimes, and for the transition between stratified wavy and annular regimes. However, the present work shows that slug flow occurs at a much higher gas superficial velocity than that predicted by the models of Weisman et al. (1979).

5.4.4 Breber et al. (1980)

This map is considered here for two reasons: it is simple to use, and it was based on the models of Taitel and Dukler (1976*a*). Breber et al. (1980) proposed that the transition between the regimes occurs at constant values of J_g^* and X and defined transition regions between the regimes. Although this map was developed for use in condensing flows, Ewing et al. (1999) showed that it could be used to predict the prevailing flow regimes in adiabatic flow, e.g., air–water in horizontal tubes.

Figure 5.15 shows a comparison between the present data and the flow regime map of Breber et al. (1980). In general, good agreement was found between the observed and the predicted flow regimes. At high values of J_g^* , slug flow was observed at $X < 1.0$,

whereas Breber et al. (1980) predict that $X = 1$ to 1.5 is the lower limit for the slug flow regime. The transitions between stratified wavy and plug or slug flow at $J_g^* < 0.2$, cannot be confirmed by the present data (data is sparse at $J_g^* < 0.2$ and $X > 1.5$). However, in the data used by Breber et al. (1980), stratified wavy flow was observed at $X > 1.5$. Therefore, it is probable that the transition between stratified wavy and plug or slug regimes occurs at $X > 1.5$ for $J_g^* < 0.2$. Taitel and Dukler (1976a) and Weisman et al. (1979) predict that at low values of J_g^* , the transition to plug or slug regimes occurs at X values significantly higher than 1.5 (Fig. 5.13 and Fig. 5.14).

Breber et al. (1980) gave no detailed discussions or justifications for their choice of the limits of the transition regions. The present data and the experimental data used by Breber et al. (1980), show that stratified wavy flow was observed at $J_g^* > 0.5$. Therefore, the lower limit of the transition region between stratified wavy and annular regimes should be set at a higher J_g^* .

5.5 Observed Transition Mechanisms

Various models were proposed to explain the physical basis for the pattern transition mechanisms (see Section 2.5). This section provides descriptions of the transition mechanisms observed in this experimental investigation. The descriptions are based on observations of the flow along the entire length of the test section, as well as rigorous examination of the videotaped flow conditions.

5.5.1 Transition to Annular Flow Pattern

The transition to annular flow can occur from either stratified wavy or slug flow.

Stratified Wavy-Annular Transition

Visual observations of the flow structure (test runs 52 and 53) confirmed that entrainment and deposition is the main transition mechanism between stratified wavy and annular regimes. In test run 52 ($J_g = 19.33$ m/s), the flow structure resembles more the wavy flow configuration; in test run 53 ($J_g = 23.35$ m/s), the flow structure resembles the annular flow regime configuration.

As the gas mass velocity increases, the amplitude of the waves increases until a point when the waves start to break down, resulting in splashing at the gas–liquid interface. The liquid droplets are then carried to the top of the tube. If the rate at which water is entrained to top of the tube exceeds the rate at which the liquid is drained to the bottom of the tube, annular flow is established.

Spedding and Nguyen (1980) classified the wavy–annular transitional flow as a separate flow regime, which they identified as “annular–roll waves” flow (Fig. 2.10). The horizontal- and vertical- planes acquired in this study clearly show that the transitional flow consists of a superimposition of the stratified wavy and the annular regimes (Fig. 5.16). The flow conditions alternate between both regimes until one of the regimes becomes more prominent and the transition occurs. Therefore, it is important to emphasize that the transitional flow should not be considered as a separate flow regime.

In Fig. 5.16, the left-hand-side shows a wavy-like flow regime and the right-hand-side shows annular-like flow regime.

Slug-Annular Transition

Slug flow is characterized by aerated and turbulent liquid slugs separated by gas regions above wavy liquid layers. As the liquid slug moves along the test section, it is replenished at the front and depleted at the back, from and to the wavy liquid layers separating the slugs. As the gas mass velocity increases, the liquid slugs become short, more frequent, and highly aerated. However, they continue to travel at a higher speed than the bulk liquid flow.

As discussed in Section 2.5.1, Hoogendoorn (1959) suggested that the slug–annular transition is established by the entrainment and deposition mechanism. Govier and Aziz (1972) and Taitel and Dukler (1976a) indicated that this transition is initiated by frequent liquid slugs wetting the top wall of the tube. Lin and Hanratty (1987b) indicated that depending on the liquid mass velocity, this transition is initiated by one of the above

mechanisms. The results of the present study show that in fact both mechanisms occur simultaneously (see Figs. 5.17*a* and 5.17*b*). Depending on the liquid mass velocity, one of the mechanisms may predominate; however, both transition mechanisms are present. At low liquid mass velocities, entrainment and deposition predominates, while at higher liquid mass velocities the liquid slugs are more frequent and the transition is established mainly by spreading of the liquid slugs.

Video observations of test runs 101 and 126 show that as the mass velocity of the gas phase increases, the liquid slugs become shorter and more aerated. At a certain mass velocity of the gas phase, the gas phase breaks through the liquid slugs, spreading the liquid around the entire wall of the tube and resulting in annular-like flow conditions. At the same time, the amplitude of the waves upstream and downstream of the liquid slugs increases until the entrainment and deposition mechanism is initiated.

Again, if the rate at which the liquid is raised to the top of the tube by both mechanisms exceeds the rate at which the liquid is drained to the bottom of the tube, annular flow is established.

As discussed in Section 2.3.1, Hewitt (1982) classified the slug–annular transitional flow as a separate flow regime, which they identified as a semislug flow. Lin and Hanratty (1987*b*) classified this transitional flow as pseudo-slug flow. However, observations made during this study show that the flow is in a transitional state.

5.5.2 Transition from Stratified Wavy to Plug or Slug Flow

This transition has been the subject of many theoretical and experimental studies, e.g., Wallis and Dobson (1973), Taitel and Dukler (1976*a*), Lin and Hanratty (1987-b), Barnea and Taitel (1993), and Jepson and Taylor (1993). Some of these studies were focussed on the prediction of the onset of slug flow regime. Most of the theoretical models developed to predict this transition are based on wave growth and Kelvin–Helmholtz stability criteria.

The following discussion describes the transition mechanisms observed in this experimental study. The descriptions are based on detailed examination of the flow along the entire length of the test section.

As the gas mass velocity increases, the waves build up and grow until they fill the entire channel, resulting in a sudden increase of the pressure drop along the test section and possibly leading to a discontinuity of the gas phase. Depending on the mass velocity of the gas and to a lesser extent on the design and configuration of the mixer, a transition to either slug or plug flow occurs. Slug flow was observed to occur at a higher gas mass velocity, when the liquid slugs were pushed through the channel by the faster-moving gas. At a relatively lower gas mass velocity, discontinuity of the gas flow was observed. The gas flow was halted until the pressure in the gas supply line was able to overcome the increased pressure drop along the test section. At this point, the gas flow was re-established as the gas was released in the shape of elongated bubbles.

Wallis and Dobson (1973) studied the onset of slugging in rectangular horizontal channels. They observed similar behavior of this transition mechanism and concluded that plug flow occurs at low void fractions ($\alpha < 0.5$) and slug flow occurs at higher void fractions ($\alpha > 0.5$).

Figure 5.18 (test run 73; $J_g = 0.79$ m/s and $J_l = 0.15$ m/s) shows the transition from stratified wavy to plug flow, and Fig. 5.19 (test runs 117; $J_g = 5.20$ m/s and $J_l = 0.14$ m/s) shows the transition from stratified wavy to slug flow.

5.5.3 Transition Between Plug and Slug Regimes

Little information is published on the transition between plug and slug flow regimes. This is because these regimes were considered to be subdivisions of the intermittent flow pattern, e.g., Taitel and Dukler (1976a), and most studies were concerned with the transition from stratified wavy to intermittent flow regimes.

In this study, video observations show that as the gas mass velocity increases, the liquid that separates the elongated bubbles becomes more and more aerated. At a certain point, as some of the elongated bubbles start to coalesce, the liquid separating the bubbles starts to resemble liquid slugs and the transition to slug flow occurs.

The transition between plug and slug regimes is illustrated in Fig. 5.20, which shows bubbles breaking up and coalescing with the following bubble.

Detailed examinations of the videotapes and general observations of the flow structure, along the entire length of the test section, indicate that plug flow was established as follows:

- (i) The pressure drop along the test tube increased with an increase in the liquid mass flow rate, resulting in a discontinuity of the air flow. The airflow was momentarily halted, resulting in oscillation of the air inlet pressure.
- (ii) When the air inlet pressure reached a certain level, a small amount of air was released into the channel in the form of an elongated bubble.
- (iii) Once some air was released, the air inlet pressure dropped and the airflow was discontinued again.
- (iv) This phenomenon was repeated, resulting in packets of air travelling with the liquid phase.
- (v) The liquid surface tension and the size of the conduit dictate the shape of the bubbles. The size and frequency of the bubbles depend on the inlet conditions of both phases.

This behaviour seems to be dependent on the pressure level of the gas and liquid supply lines and on the design and configuration of the mixer.

5.5.4 General Discussion about Transitional Flow

As discussed in Section 5.2.1, the transition between the flow patterns occurs over a range of flow conditions. It is important to draw the transition regions on the flow

regime map to show analysts the transitional behaviour of the flow. It should be noted that since the flow in the transition regions is in a developing state, it should not be considered as a separate flow pattern. The flow in these regions can be described as alternating between two or more flow regimes. As the inlet conditions of the flow change, e.g., an increase of either the gas or the liquid flow rates, one of the regimes predominates. An example of this phenomenon is the transition region between the stratified wavy and the annular flow regimes. It was observed in this study that an annular-like flow traveled along the pipe, followed by a stratified wavy like flow with wisps of liquid droplets in the upper half of the channel.

5.6 Proposed Flow Regime Map

The objective of this section is to recommend a flow regime map for use in horizontal adiabatic flows in small diameter pipes. The proposed map is based on the following three criteria: first, the experimental data of the present study; second, the observed transition mechanisms discussed in Section 5.5; and third, the results of the comparative study with the four most widely referenced flow regime maps discussed in Section 5.4.

5.6.1 Choice of Mapping Parameters

The coordinate systems used by some investigators to present their flow regime maps are listed in Table 2.1. As discussed in Chapter 2, there is general agreement on the most appropriate set of parameters that can be used to define the boundaries of the flow patterns. The gas and liquid superficial velocities (J_g and J_l), suggested by Mandhane et al. (1974) were used by a number of investigators, e.g., Pearce (1982), Lin and Hanratty (1987b) and Kokal and Stanislav (1989).

Taitel and Dukler (1976a) argued that the superficial velocities of the phases are not expected to correlate the various mechanisms that govern the transitions between the flow regimes. As discussed in Section 2.4.3, Taitel and Dukler (1976a) expressed the transition between the flow patterns in terms of five dimensionless parameters defined by

Eqs. 2.3 to 2.7. However, only two of these parameters are required to correlate the transitions between the flow patterns observed in this study. The two parameters are F and X , where F is the modified Froude number defined by Eq. 2.5, and X is the Lockhart–Martinelli parameter defined by Eq. 2.3. As discussed in Section 2.4.7, in horizontal flow F is equal to the dimensionless gas superficial velocity J_g^* , also known as the Wallis parameter.

The parameters J_g^* and X were selected for the flow regime map that will be presented in the following section. These parameters were successfully used by Taitel and Dukler (1976a), Breber et al. (1980) and Ewing et al. (1999). In addition, as discussed in Section 5.5, some of the transitions between the observed flow patterns occur at a given value of either J_g^* or X .

5.6.2 Boundaries of Flow Regimes

As mentioned earlier, it is recommended that the flow patterns in horizontal pipes be classified as the following four flow regimes: stratified wavy, plug, slug, annular, and dispersed bubble. Since dispersed bubble flow was not observed in this study, only the transitions between stratified wavy, plug, slug, and annular flow regimes are considered in this section. The transition boundaries proposed in this study are shown in Fig. 5.21 and discussed in the following sections.

Stratified Wavy–Annular Transition

Based on the results of the present study and the discussions presented in Section 5.5, it is proposed that the transition between stratified wavy and annular flow regimes occurs at J_g^* between 0.9 and 1.5. This proposed transition is similar to that suggested by Breber et al. (1980) except for the higher value of the lower limit of the transition region. Breber et al. (1980) suggested that this transition occurs at $J_g^* = 0.5$. In the present study, clearly identifiable stratified wavy flow was observed at a higher J_g^* .

In Fig. 5.22, the proposed transition boundaries are compared with the flow regime maps discussed in Section 5.4. As discussed earlier, the maps of Taitel and Dukler (1976a) and Breber et al. (1980) (Fig. 5.22-b and 5.22-c), were given using the same mapping parameters selected in this study (J_g^* and X). However, the maps of Mandhane et al. (1974) and Weisman et al. (1979) were given using the gas and liquid superficial velocities (J_g and J_l). Therefore, the boundaries of these maps are recalculated using the properties of air and water at standard atmospheric conditions.

Figure 5.22 shows that the proposed transition region is in reasonable agreement with the maps of Breber and al. (1980), Taitel and Dukler (1976a), and Weisman et al. (1979). The map of Mandhane et al. (1974) suggests that the transition occurs at a lower J_g^* . It is interesting to note that Wallis et al. (1978) suggested that this transition occurs at $J_g^* = 0.9$, which agrees with the lower limit of the transition range proposed here.

The proposed transition regions between the flow patterns are compared with the observations of França and Lahey (1992) in Fig. 5.23. The proposed transition regions are in reasonable agreement with the experimental data of França and Lahey (1992) with the exception of the transition between plug and slug flow regimes. As discussed at the end of Section 5.3.2, the discrepancy in plug–slug transition is probably due to the arbitrary criterion used by França and Lahey (1992) to identify the plug flow regime.

Stratified Wavy–Plug or Slug Flow Transition

As discussed earlier, the observed transition from stratified wavy to slug or plug flow is well represented by the transition boundaries of Taitel and Dukler (1976a) and Weisman et al. (1979). Breber et al. (1980) suggested that this transition region occurs at $X = 1$ to 1.5. However, as shown in Fig. 5.22, the flow regime maps of Mandhane et al. (1974), Taitel and Dukler (1976a), and Weisman et al. (1979) indicate that as J_g^* increases, the transition occurs at lower values of X .

The proposed transition between stratified wavy and plug or slug flow (shown in Fig. 5.21) approximates the transitions of Taitel and Dukler (1976a) and Weisman et al.

(1979) (Fig. 5.22). Again, the map of Mandhane et al. (1974) indicates that the transition between stratified wavy and slug flow occurs at lower J_g^* and X . This may be due to the well-defined definition of slug flow considered in the present study, i.e., slug flow is identified when complete blockage of the gas flow area occurs (i.e., not when the waves momentarily touch the top wall of the channel).

Slug–Annular Flow Transition

The proposed transition between slug and annular flow regimes occurs at a constant X in the range between $X = 1$ – 1.5 (Fig. 5.21). Figure 5.22 shows that this transition is in reasonable agreement with the flow regime maps of Taitel and Dukler (1976a) and Breber et al. (1980). Figure 5.22c shows that the proposed transition agrees with the transition of Weisman et al. (1979) for $0.01 < X < 0.1$.

Plug–Slug Flow Transition

As indicated in Chapter 2, little information is provided in the literature on the transition between plug and slug flow regimes. The present data show that this transition occurs at a nearly constant value of J_g^* in the range of 0.09 to 0.15 m/s at high values of X . As indicated earlier, Taitel and Dukler (1976a), Weisman et al. (1979), and Breber et al. (1980) amalgamated plug and slug flow regimes. Therefore, their flow regime maps show no transition between plug and slug flow regimes.

The flow regime map of Mandhane et al. (1979) indicates that this transition occurs at J_g^* in the range of 0.01 to 0.02 m/s. Tandon et al. (1982) proposed a flow regime map for condensing flows in horizontal tubes. They suggested that the transition between plug and slug regimes occurs at $J_g^* = 0.01$. This discrepancy may be explained by the restricted definition of the plug flow regime of Mandhane et al. (1974) and Tandon et al. (1982), as discussed in Section 5.4.1.

Transition Regions

The limits of the proposed transition regions between flow patterns can be reproduced on (X, J_g^*) coordinates using the data points listed in the Table 5.3.

Table 5.3: Proposed limits of the transition regions between the observed flow patterns.

Transition region	Data points for (X, J_g^*) coordinates
• Transition between stratified wavy and annular flow regimes	(0.001 , 0.9) and (0.5 , 0.9)
	(0.001 , 1.5) and (1 , 1.5)
• Transition between stratified wavy and plug or slug flow regimes	(0.5 , 0.9) and (8 , 0.01)
	(1.5 , 0.4) and (15 , 0.01)
• Transition between plug and slug flow regimes	(4 , 0.9) and (100 , 0.09)
	(3 , 0.15) and (100 , 0.15)
• Transition between slug and annular flow regimes	(1 , 1.5) and (1 , 10)
	(1.5 , 0.4) and (1.5 , 10)

Measurements Uncertainties

The complete analysis of the expected uncertainties in the measured and calculated parameters is presented in Appendix B. The 95% confidence level uncertainties calculated for the liquid and gas superficial velocities are presented in Table B.3 and B.4, respectively. Figure 5.24 shows the expected uncertainties in the variations of the location of the data points on the flow regime. Note that, for most of the data points, the size of the error bars is smaller than the size of the symbols. Figure 5.24 shows that, with a 95% confidence level, all data points fall within the boundaries of their assigned flow regime.

5.6.3 Range of Applicability

The range of operating conditions of the present data is given in Table 5.1. The proposed flow regime map is proposed mainly for adiabatic horizontal flows in small diameter pipes. The present data were taken in a 25.4-mm diameter pipe. However, as

discussed in Section 2.6.2, slightly larger diameter pipes (up to 50 mm) are expected to have only moderate effects on the boundaries of the flow regimes.

Effects of variations in the fluid properties and inclination of the test section from the horizontal on the boundaries of the flow patterns are discussed in Chapter 2. The following is intended to provide general discussions on the applicability of the proposed flow regime map to boiling and condensing flows.

General Consideration for Boiling and Condensing Flows

Most of the two-phase flow maps were concerned with adiabatic gas–liquid flows. Therefore, these maps do not consider momentum exchange due to mass transfer during phase change. Some investigators suggested that flow regime maps developed for adiabatic flows should not be used for boiling and condensing flows (Hewitt, 1978 and Nitheanandan, 1993). It is argued that the mechanisms for the transition between flow patterns in boiling and condensing flow are different from those observed in adiabatic flows. Mass and momentum transfer between the phases is expected to generate additional forces. In the case of boiling flows, the vapour phase is generated at nucleation sites over the entire wall of the conduit or from the surface of the wall film in annular boiling flows. This affects the distribution of the phases and the mechanisms that govern the transition between the regimes. In addition, the mass flow rates of the phases vary along the conduit and the flow regimes are not fully developed. However, the adiabatic flow regime maps discussed in this study were used to predict the flow regimes in boiling and condensing flows, e.g., CATHENA uses some of the models of Weisman et al. (1979).

Dukler and Taitel (1986) studied the effect of mass transfer between the phases on the transition boundaries predicted by their original models, which they developed for adiabatic flows (Taitel and Dukler, 1976). Results of this study show that the effects are small and are expected to be within the uncertainties in the regime boundaries.

In summary, based on the above discussion, it is reasonable to assume that effects caused by mass transfer between the phases are expected to be within the overall uncertainties associated with the locations of the transitions curves. However, it is considered advisable to test the proposed flow regime against boiling and condensing flow data.

Figure 5.1: Photograph of stratified wavy flow (test run 48: $J_g = 3.04$ m/s and $J_l = 0.05$ m/s).

Figure 5.2: Photograph of plug flow (test run 64: $J_g = 0.1$ m/s and $J_l = 0.93$ m/s).

Figure 5.3: Photograph of slug flow (test run 99: $J_g = 6.61 \text{ m/s}$ and $J_l = 0.73 \text{ m/s}$).

Figure 5.4: Photograph of annular flow (test run 55: $J_g = 21.26 \text{ m/s}$ and $J_l = 0.20 \text{ m/s}$).

Figure 5.5: Time traces and their PDF for stratified wavy flow (test run 48: $J_g = 3.04$ m/s and $J_l = 0.05$ m/s).

Figure 5.6: Time traces and their PDF for plug flow (test run 64: $J_g = 0.1$ m/s and $J_l = 0.93$ m/s).

Figure 5.7: Time traces and their PDF for slug flow (test run 99: $J_g = 6.61\text{m/s}$ and $J_l = 0.73\text{ m/s}$).

Figure 5.8: Time traces and their PDF for annular flow (test run 55: $J_g = 21.26$ m/s and $J_l = 0.20$ m/s).

(a) Stratified wavy – Annular transition
(Test run 52; $J_g = 19.33$ m/s and $J_l = 0.02$ m/s).

(b) Slug – Annular transition
(Test run 101; $J_g = 13.64$ m/s and $J_l = 0.81$ m/s).

(c) Stratified wavy– Plug transition
(Test run 73; $J_g = 0.79$ m/s and $J_l = 0.15$ m/s).

(d) Plug – Slug transition
(Test run 91; $J_g = 1.29$ m/s and $J_l = 0.95$ m/s).

Figure 5.9: PDF of the time traces from the optical probe for test runs in the transition regions.

Figure 5.10: Photograph showing scalloped bubbles in plug flow regime (test run 89; $J_g = 0.181$ m/s and $J_l = 1.076$ m/s).

Figure 5.11: Comparison of the present data with data of Franca and Lahey (1992).

Figure 5.12: Comparison of observed flow regimes with flow regime map of Mandhane et al. (1974).

Figure 5.13: Comparison of observed flow regimes with transition models of Taitel and Dukler (1976a).

Figure 5.14: Comparison of observed flow regimes with flow regime map of Weisman et al. (1979).

Figure 5.15: Comparison of observed flow regimes with flow regime map of Breber et al. (1980).

Figure 5.16: Photograph showing superimposition of stratified wavy and annular flow regimes.

Figure 5.17: Transition between slug and annular flow regimes; both entrainment and deposition (a) and wave break-up and spreading (b) are present (test run 101, $J_1 = 0.811$ m/s and $J_g = 13.464$ m/s).

Figure 5.18: Transition between stratified wavy and plug flow regimes (test run 73; $J_g = 0.79$ m/s and $J_l = 0.15$ m/s).

Figure 5.19: Transition between stratified wavy and slug flow regimes (test run 117; $J_g = 5.20$ m/s and $J_l = 0.14$ m/s).

Figure 5.20: Transition between plug and slug flow regimes - onset of bubble coalescence, (test run 91; $J_g = 1.566$ m/s and $J_l = 0.809$ m/s).

Figure 5.21: Proposed flow regime map for adiabatic gas-liquid flow in small diameter horizontal pipes.

Figure 5.22: Comparison of the proposed pattern boundaries with other flow regime maps.

Figure 5.23: Comparison of the proposed transition regions with the experimental data of França and Lahey (1992).

Figure 5.24: Expected (95% confidence level) variations in the location of the data points in the flow regime map.

Measured Friction Losses and Void Fraction

6.1 Introduction

In Chapter 3, the models and correlations used in the analysis of two-phase flow systems were categorized into two groups: (i) *Generalized* models and correlations, which do not require prior knowledge of the prevailing flow pattern, and (ii) *Pattern-dependent* models and correlations, which do require prior identification of the flow pattern.

The following are the main objectives of this chapter:

- (i) to present and to discuss measurements of friction losses and local and average void fractions;
- (ii) to compare measurements with predictions of the generalized and specific models and correlations presented in the previous Chapters;
- (iii) to discuss the applicability of these models in predicting the friction losses and average void fractions for each of the flow regimes observed in this study;
- (iv) to compare the predictions of the two-fluid models developed in Chapter 3 for the prediction of the friction losses in the observed flow regimes; and
- (v) to present measurements of interfacial area concentration for the stratified wavy flow regime.

It should be noted that analysis of the interfacial area measurements is limited to the stratified wavy flow regime.

6.2 Void Fraction

6.2.1 Measured Local Void Fraction

The void fraction was measured using the double-sensor probe described in Section 4.4.2. Each sensor measures the time-averaged void fraction at nine equally spaced locations along the pipe diameter. Typical profiles of the local void fraction (α_{loc}) for stratified wavy, plug, slug, and annular flow patterns, are shown in Fig. 6.1. The radial locations are presented as (r/R) , where -1 indicates the bottom of the tube and $+1$ indicates the top of the tube. The general trends of the void profile are discussed in the following sections.

The recorded signal and PDF for test runs shown in Fig. 6.1 are given in Chapter 5 (Fig. 5.5 to Fig. 5.8). The measured local void fractions for all test runs are presented in Appendix E.

- (i) *Stratified Wavy Flow*: The radial distribution of the void fraction for the stratified wavy flow is illustrated in Fig. 6.1a, which shows the liquid flowing along the bottom of the tube ($\alpha_{loc} = 0$) while the gas occupies the upper part of tube ($\alpha_{loc} = 1$). As discussed earlier, because of flow stratification, the downstream sensor, which is located ~ 1 mm above the upstream sensor, measures slightly higher void fractions than the upstream sensor.

It should be noted that in all test runs classified as stratified wavy flow, the local void fraction measure near the top of the test tube is almost equal to one. This confirms that the waves do not reach the top of the tube and block the flow area of the gas phase. This is consistent with the criterion used to distinguish between stratified wavy and slug flow regimes.

- (ii) *Plug Flow*: Figure 6.1b shows a typical distribution of the local void fraction for the plug flow regime. Liquid flows at the bottom of the channel while air intermittently flows at the upper part of the channel. Again, the average void

fraction measured by the downstream sensor is slightly higher than that measured by the upstream sensor for the reasons stated above. Differences between the measurements are relatively higher near the air–water interface ($r/R = -0.2$ at the bottom of the elongated bubbles).

(iii) *Slug Flow*: Figure 6.1c shows a typical distribution of the local void fraction for slug flow regime. The distribution is similar to that of the plug flow (Fig. 6.1b), except for the following:

- higher local void fractions measured near the top of the tube (0.9 for slug flow compared to 0.7 for plug flow);
- thinner film at the bottom of the channel (air–water interface is detected at $(r/R) < -0.6$ in slug flow, while in plug flow the interface is detected at $(r/R) > -0.4$); and
- more waves are observed at the gas–liquid interface in slug flow than in plug flow. The waves are evident in the variations of the measured local void fraction between the sensors.

(iv) *Annular Flow*: The distribution of the local void fraction for annular flow regime is shown in Fig. 6.1d. The measured α_{loc} is relatively high at all nine locations. Note that because of gravity, the thickness of the liquid film is higher at the bottom of the tube than at the top. The average local void fraction measured at 2.5 mm from the bottom wall of the tube is ~ 0.7 . The average local void fraction measured at 2.5 mm from the top wall of the tube is about 1.0, which indicates that the thickness of the liquid film at the top of the tube is less than 2.5 mm.

6.2.2 Measured Average Void Fractions

As discussed in Section 4.6.2, the average void fraction represents the area-weighted average of the local void fraction measurements. Figure 6.2 compares the average void fraction measured using the upstream and downstream sensors for all the

test runs. This figure shows that the measurements are within 10%. The relative difference between the measurements is less than 5% for about half of the test runs and less than 10% for 80% of the test runs. The relative difference between the measurements is higher than 15% in only 5 of the 60 test runs²⁰.

Figure 6.2 also shows that the void measured using the downstream sensor is consistently slightly higher than that measured using the upstream sensor. The reason for this variation is that the downstream sensor is positioned at ~1 mm above the upstream sensor, as shown in Fig. 4.3. Because of flow stratification, the downstream sensor is expected to measure higher time-averaged local void fractions, in wavy, plug, and slug flow regimes.

Comparison with Experimental data of França and Lahey (1992)

To check for validity and correctness, the measurements of the void fraction are compared with the experimental data of França and Lahey (1992) (Fig. 6.3). The data of França and Lahey (1992) are for air–water flow in a 19-mm diameter horizontal tube. The average void fraction was measured using quick-closing valves, and the total length of the test section was ~315 (l/d).

The experimental data are presented in terms of the measured void fractions (α) plotted against the ratios of the gas to liquid superficial velocities (J_g/J_l). These variables were chosen for direct comparison with the data of França and Lahey (1992), which are exhibited in these coordinates and also because they included the values of J_g and J_l with their published results. Note that Elkow and Rezkallah (1997) used (J_g/J_l) to present their measurements of the void fraction and referred to this ratio as the pseudo void fraction ratio.

Figure 6.3 shows that at low (J_g/J_l) ratios (less than 10), França and Lahey (1992) reported higher readings of the void fraction than measured in the present study. At high

²⁰ As stated in Chapter 4, void fraction measurements were taken for 60 experimental test runs only.

ratios ($J_g/J_l > 50$), their readings show a large scatter in the data, with the upper part of the scatter range in good agreement with the present data. Note that low values of α are for plug flow and high values of α are for annular and stratified wavy flow.

França and Lahey (1992) used quick-closing valves to measure the global void fraction. They did not give many details of the system, except that each measurement represents the average of at least three readings, and that the uncertainties associated with their measurements are expected to be within $\pm 5\%$. Accurate measurements of the void fraction using quick-closing valves require good synchronization between the valves and simultaneous diversion of the flow. If average readings are taken, after each measurement the original conditions of the flow must be reestablished and the flow must be brought back to the same steady state conditions before the readings are taken again. This might be difficult to achieve, especially under highly intermittent flow conditions such as slug flow.

6.2.3 Comparison with Predictions of Generalized Models

As discussed in Chapter 3, the most widely accepted generalized models for predicting the time-averaged void fraction are the homogeneous model, the Lockhart–Martinelli correlation, the separate-cylinders model, and the CISE correlation. Predictions based on these models are compared with the measured average cross-sectional void fractions in Fig. 6.4 and Fig. 6.5. In Fig. 6.4, the measured and predicted void fractions are presented as a function of the Lockhart–Martinelli parameter (X); in Fig. 6.5, predictions of the models are compared individually with the measured void fractions.

The observed scatter in the predicted values shown in Fig. 6.4 is due to the fact that the predictions of the models are based on the measured properties of the phases. For example, density and dynamic viscosity of the phases are calculated based on the measured pressure and temperature.

Homogeneous Model

The homogeneous void fraction is calculated using Eq. 3.22. Figure 6.5a shows that the homogeneous model overpredicts the experimental data significantly, which agrees with the general consensus. It is well established in the literature that the homogeneous model overestimates the average void fraction (Rogers, 1985, Whalley, 1987 and Collier and Thome, 1994). The homogeneous void fraction (Eq. 3.22) is based on the assumption that the slip ratio “ S ” is equal to one, i.e., $\alpha_h = \beta$. This assumption does not agree with the results of most experimental studies, including the present study. As discussed later in this chapter, for most flow regimes (annular, slug, and stratified wavy) the gas phase travels at a higher velocity than the liquid phase, i.e., $S > 1$.

Lockhart–Martinelli Model

Figure 6.5b shows that the Lockhart–Martinelli model provides reasonable predictions of the measured void fractions at $\alpha > 0.4$. At low void fractions ($\alpha < 0.4$), the Lockhart–Martinelli model overpredicts the measured void fractions significantly. The low values of the void fraction correspond mainly to plug flow. At higher values of the void fraction ($\alpha > 0.4$) the predicted values are well within $\pm 20\%$, with the correlations slightly underpredicting the measured average void fraction.

Separate-cylinders Model

Predictions of the separate-cylinders model are compared with the measured void fractions in Fig. 6.4c. In general, this model considerably underpredicts the experimental data, especially for $0.3 < \alpha < 0.7$. The separate-cylinders model gives relatively better predictions at high values of α , mainly for annular flow. This is not surprising since the model was developed based on an idealized configuration of annular flow regime.

CISE Correlation

Figure 6.4d shows that the CISE correlation provides reasonably good agreement with the measured void fractions. At low void fraction ($\alpha < 0.4$) the CISE correlation

slightly overpredicts the measurements; however, at higher void fractions, the predictions are well within $\pm 10\%$.

Summary

The homogeneous model consistently overpredicted the measured time-average void fraction. The Lockhart–Martinelli model provides reasonable predictions of the void fraction if $\alpha > 0.4$ and overpredicts the measurements if $\alpha < 0.4$. The separate-cylinders model underpredicts the void fraction measurements. The CISE correlation provides the best predictions of the time-average void fractions.

6.2.4 Variations of Averaged Void Fraction with Flow Regimes

In this section, predictions of regime-specific correlations are compared with the measured average void fraction.

Observed variation of Averaged Void Fraction with Flow Regimes

The ranges of the measured average void fraction for the flow patterns observed in this study are given Table 6.1. Figure 6.6 shows variations of the void fraction for the observed flow patterns as a function of the Lockhart–Martinelli parameter (X_{tt}) calculated for turbulent gas and liquid flows (see Section 3.3.2). Figure 6.6 shows a trend of the measured void fraction with the observed flow regime.

Table 6.1: Range of measured average cross-sectional void fraction for different flow patterns.

Flow regime	No. of test runs	Range of measured average cross-sectional void fraction
Annular flow	7	0.77 – 0.97
Plug flow	28	0.02 – 0.67
Slug flow	17	0.40 – 0.88
Stratified wavy flow	8	0.65 – 0.94
All test runs	60	0.02 – 0.97

Stratified wavy and annular flow are observed at high void fractions ($\alpha > 0.65$). Plug flow occurs at low void fractions; with the exception of three test runs, the void fraction measured in plug flow is less than 0.5. For slug flow, the measured void fraction is higher than 0.5. Void fraction is often used to distinguish between plug and slug flow patterns, e.g., Taitel and Dukler (1976a).

6.2.5 Drift-flux Model for Void Fraction

In Fig. 6.7, the experimental data are plotted in terms of the standard drift-flux variables (J_g/α) and (J) for the annular, plug, slug, and stratified wavy flow regimes. Note that the averaging symbol $\langle \rangle$ was omitted for convenience of presentation. For slug and stratified wavy flow patterns, Fig. 6.7a and Fig. 6.7b show a clear linear relationship between the drift-flux variables. For annular flow, Fig. 6.7c, the relationship between the variables is less noticeable because of the relatively small number of experimental data points (seven data points). Figure 6.7d shows a large scatter in the experimental data for the plug flow regime. The reason for this scatter is discussed later in this section. The size of the error bars shown in Fig. 6.7 is equal to $\pm$ the 95% confidence level uncertainties calculated for J_g , see Appendix B.

The drift-flux parameters C_o and V_{gj} were determined experimentally using linear regression analysis of the experimental data plotted in terms of the drift-flux variables and are presented and discussed for each of the observed flow regimes.

Stratified Wavy Flow

Figure 6.8 shows a striking linear dependency between the drift-flux variables for the stratified wavy flow pattern. The solid line represents the best linear fit of the data. The coefficient of determination of the fit (R^2) is higher than 0.999, which indicates a strong linear relationship between the drift-flux variables.

The measured drift-flux parameters C_o and V_{gj} , are 1.04 and 0.466 m/s, respectively. The distribution parameter (C_o) is slightly higher than 1, which implies that

the void profile is not completely in phase with volumetric flux profile. The positive value of V_{gj} implies that the gas phase travels at a higher velocity than the liquid phase. This agrees with the widely accepted characteristics of the stratified wavy flow regime. As discussed in Chapter 5, the waves are generated by the action of the faster moving gas phase at the free surface of the liquid phase.

In Chapter 3, the stratified wavy flow data of França and Lahey (1992) are correlated using the standard drift-flux variables (J_g/α) and (J), see Fig. 3.7d. The correlation shows a strong linear relationship between the drift-flux variables ($R^2 = 0.995$). The calculated values of C_o and V_{gj} are 1.01 and 0.757 m/s, respectively (see Table 3.3). Reasonable agreement was obtained with the results of the present work for C_o . However, the data of França and Lahey (1992) gave a higher value for V_{gj} than the calculated value based on the data of the present work.

Slug flow

Correlation of the slug flow data is presented Fig. 6.9. A strong dependency between the drift-flux variables is also observed for the slug flow regime ($R^2 = 0.991$). The calculated values for the drift-flux parameters C_o and V_{gj} , are 1.06 and 0.991, respectively. Similar to stratified wavy flow regime, C_o is slightly greater than 1, which implies that the void profile and the volumetric flux profile are not in phase. The positive value of V_{gj} indicates the gas phase moves at a higher velocity than the liquid phase, which agrees with the generally accepted definition of slug flow, i.e., liquid slugs are transported by the faster moving gas phase. It is also interesting to note that, because slug flow is characterized by larger differences between the gas and liquid velocities than stratified wavy flow, V_{gj} for slug flow is higher than that of stratified wavy flow.

The values for C_o and V_{gj} are distinctly different from those calculated for the data of França and Lahey (1992) ($C_o = 1.22$ and $V_{gj} = -0.218$ m/s, see Table 3.3). The negative value of V_{gj} given by França and Lahey (1992) suggests that the gas phase moves at a lower velocity than the liquid phase. This contradicts the reported observations of the

characteristics of slug flow. As indicated earlier, it is widely reported that the liquid slugs are pushed through the test section by the faster moving gas. Also, the wavy interface of liquid film, downstream of the liquid slugs, clearly implies that the gas moves at a higher velocity than the liquid, thus implying a positive value of V_{gj} .

This discrepancy in the calculated value of V_{gj} from the slug flow data of França and Lahey (1992) may be due to the slightly higher void fraction reported for the intermittent flow regime (see Section 6.2.2). As discussed in Chapter 5, França and Lahey (1992) considered that the transition from slug flow to annular flow occurs at a lower gas superficial velocity than observed in this study. Figure 3.7 shows that slug flow occurs at a volumetric flux less than 3.5 m/s, compared with ~15 m/s in the present study. This small range of J for slug flow may have contributed to the discrepancy in the calculated value of V_{gj} . To confirm this proposition, some of the test runs classified by França and Lahey (1992) as annular flow, i.e., those with $J_g < 20$ m/s, are combined with their slug flow data and correlated using the standard drift-flux variables. The results of this correlation are shown in Fig. 6.10, with the added test runs are represented by slightly larger symbols. The calculated values for C_0 and V_{gj} are 1.12 and 0.05 m/s, respectively. A positive value for V_{gj} is calculated if the flow patterns are classified in a manner consistent with the present study (see Chapter 5).

Annular Flow

As indicated earlier, the void fraction was measured for only 7 of the 18 annular flow test runs. Because of this small number of test runs, the present data could not be correlated with the accuracy required for this study. Therefore, the experimental data of França and Lahey (1992) for annular flow data points with $J_g > 20$ m/s were combined with the annular flow data of the present study and correlated using the standard drift-flux variables. The results of this correlation are shown in Fig. 6.11. Note that the present data are represented with slightly larger symbols than those representing the data of França and Lahey (1992).

The calculated values for C_o and V_{gj} are 0.92 and 3.688 m/s, respectively. The coefficient of determination (R^2) is 0.832, which is lower than the values calculated for the stratified wavy and slug flow regimes. This can be expected since in this case the two sets were taken in pipes of different diameters using different measurement techniques and were amalgamated and correlated together. In annular flow C_o is less than 1, and the profiles of α and J peak at the centre of the conduit. As discussed in Chapter 5, most of the liquid flows along the wall of the channel.

The positive value for V_{gj} indicates that the gas moves at a higher velocity than the liquid phase, which agrees with the definition of annular flow regime. The characteristics of the annular flow and mechanisms that govern the transition to this regime are discussed in Chapter 5. Only a fraction of the liquid phase travels at the same velocity as the gas phase, in the form of liquid droplets in the core of the conduit. The bulk of the liquid is dragged along the wall of the conduit in an annular form with a wavy interface. Therefore, on average, the gas moves at a much higher velocity than the liquid phase.

Note that best linear-fit of the annular flow data of França and Lahey (1992) yields $C_o = 1.01$ and $V_{gj} = 1.790$ m/s with $R^2 = 0.994$ (see Fig. 3.7a).

Plug flow

Figure 6.7d, using the data for plug flow, shows that for $J < 1.5$ m/s, a number of test runs show significantly high values of J_g/α . The measured void fraction for these data points (test runs 70, 85, 86, 87, 88 and 89) is less than 0.07. Therefore, the variations in the calculated values of J_g/α were initially attributed to errors associated with measurements of very low void fractions. However, the void measurements were checked and all indications show that the error associated with the measurements for these test runs is the same as that of other test runs (less than 5%). Uncertainties in the measurement of other parameters for these test runs, such as flow rates and static pressure, do not account for the significantly high values of J_g/α . Estimates of the uncertainties in the calculated values of J_g/α are shown in Fig. 6.7d.

Close examination of these test runs revealed that the distribution of the phases is significantly different from the remaining test runs of the plug flow regime. The air flows along the top of the test section in the form of scalloped bubbles that are similar to those depicted in Fig. 5.10. As mentioned in Chapter 5, other investigators identified such flow as a bubble flow regime. However, in this study, since the turbulent forces in the liquid phase are not strong enough to overcome the buoyancy forces that keep the gas bubbles near the top of the conduit, these test runs are considered to be different conditions of the plug flow regime.

In stratified wavy and annular flow regimes, the distribution of the phases along the test section is almost uniform. This is not the case of the plug and slug flow regimes. As described in Chapters 2 and 5, these regimes are characterized by intermittence in the distribution of the phases, hence the term intermittent flow that is used to identify these regimes. In slug flow, the intermittence of the gas phase is less pronounced since the liquid slugs are highly aerated and both phases are present along the test section. However, in plug flow the intermittence in the gas phase is more pronounced. As discussed in Chapter 5, plug flow is established by discontinuity in the flow of the gas phase. Therefore, plug flow can be described as alternations in time and space between single-phase liquid flow and two-phase gas–liquid flow. Figure 3.10 shows an idealized diagram of the distribution of the phases in the plug flow regime. In Fig. 3.10, L_B represents the average length of the elongated or scalloped bubbles, and L_S is the average length of the liquid slugs that separate the bubbles.

In Fig. 6.7*d*, the drift-flux variable J_g/α is calculated using the time-average cross-sectional void fraction α . A new approach was developed in this study in which the plug flow data was correlated based on the void fraction averaged over the two-phase flow region L_B . The time-average void fraction α_p can be defined as

$$\alpha_p = \frac{\alpha}{\left(\frac{L_B}{L_B + L_S} \right)} \quad (6.1)$$

The ratio of $L_B/(L_B+L_S)$ represents the fraction of the test section length that is simultaneously occupied by both phases. This ratio is equivalent to the time-averaged local void fraction measured near the upper wall of the tube. Therefore, the time-averaged void fraction in the two-phase flow region is given by

$$\alpha_p = \frac{\alpha}{\alpha_{loc,9}} \quad (6.2)$$

where $\alpha_{loc,9}$ is the average local void fraction measured using both sensors at the uppermost location along the diameter of the test tube (location 9). Figure 6.12 shows the calculated values of α_p as a function of α . The data is fitted using a power-fit and the best-fit equation is

$$\alpha_p = 0.986 \times \alpha^{0.533} \quad (6.3)$$

The coefficient of determination of this power-fit (R^2) is ~ 0.90 .

The fractions of the two-phase flow regions ($L_B/(L_B+L_S)$) were also determined from the recorded videos of the flow. The videos were played back at a slower speed and the times when the beginning and the end of the elongated or scalloped bubbles cross over a marked line on the screen were recorded using an interactive MATLAB program. The fractions of the two-phase flow regions determined using this approach are compared to those calculated as the local void fraction near the top wall of the channel in Fig. 6.13. The solid line represents the best linear-fit between the results of the two approaches. Reasonable agreement was observed between the results of both approaches. Results obtained from video measurements are, on average, about 4% higher than results obtained using the optical probes.

Both approaches are based on the assumption that the liquid and the air bubbles travel at the same velocity, since it is in fact the time fractions that are measured by the optical probes and the video method. This is an acceptable assumption since, as discussed in Chapter 5, in plug flow the air bubbles are transported by the liquid phase. Also, since the experiments were performed at near atmospheric pressure, the average length of the bubbles is not expected to change as the bubbles travel along the test section.

The data of plug flow were correlated using the variables J_g/α_p and J , with α_p calculated using Eq. 6.3. The results of this correlation are shown in Fig. 6.14. The calculated drift-flux parameters C_o and V_{gj} are 0.98 and +0.068 m/s, respectively. The coefficient of determination (R^2) of this correlation is 0.727.

The drift-flux distribution parameter C_o is almost equal to 1, and average drift velocity (V_{gj}) is almost equal to 0. This indicates that phases travel at the nearly the same velocity. This agrees with the observed characteristics of the plug flow regime. As discussed in Chapter 5, in plug flow the elongated bubbles are entrained along the channel by the liquid phase.

The experimental data of França and Lahey (1992) for plug flow were correlated using the variables J_g/α_p and J , with α_p calculated using Eq. 6.3. The plug flow data of França and Lahey (1992) were extended to include some of their slug flow data, using test runs with $J_g < 1.5$ m/s. As discussed in Chapter 5, França and Lahey (1992) considered the transition to slug flow to occur at a lower J_g than observed in the present study.

The results of this correlation are shown in Fig. 6.15. The calculated values for C_o and V_{gj} are 0.736 and -0.066 m/s, respectively. The distribution parameter is considerably less than that calculated for the present data because França and Lahey (1992) measured higher void fractions for the plug flow than in the present study (see Section 6.2.2). The calculated drift velocity is almost equal to 0, which agrees with the

present study. The calculated values of V_{gj} for the present data and for the data of França and Lahey (1992) are statistically insignificant.

Summary and Discussion

Results of the present study show that the drift-flux model can be used to correlate the average cross-sectional void fraction in horizontal flows. This study shows a strong linear relationship between the standard drift-flux variables J_g/α and J for slug, stratified wavy, and annular flow regimes. For plug flow, the drift-flux variable (J_g/α) is calculated based on the average void fraction at the level of the elongated bubbles (α_p). The following are the results of the drift-flux model correlations:

$$\text{stratified wavy flow} \quad J_g/\alpha = 1.04 \ J + 0.466 \quad (6.4a)$$

$$\text{slug flow} \quad J_g/\alpha = 1.06 \ J + 0.991 \quad (6.4b)$$

$$\text{annular flow} \quad J_g/\alpha = 0.92 \ J + 3.688 \quad (6.4c)$$

$$\text{plug flow} \quad J_g/\alpha_p = 0.98 \ J + 0.068 \quad (6.4d)$$

Knowing the gas and liquid superficial velocities and the prevailing flow pattern, these equations can be used to calculate the time-average void fraction. For plug flow, Eq. 6.4d gives the void fraction in the two-phase flow regions α_p . In this case, Eq. 6.3 can be used to calculate the overall time-average void fraction α .

The above values of the drift-flux parameters agree reasonably well with those calculated based on the experimental data of França and Lahey (1992), once the data of França and Lahey (1992) are classified in a manner consistent with the present study (see Chapter 5).

6.3 Two-phase Flow Friction Losses

6.3.1 Measured Friction Losses

As discussed in Section 4.3.3, wall pressure was measured at three locations²¹ along the test tube (see Fig. 4.1):

- (i) $P1$ is near the inlet ($l/d = 1$) downstream from where the conical diffuser joins the mixer to the test tube;
- (ii) $P2$ is at ~ 410 (l/d), just downstream of where void fraction and the interfacial area measurements were taken; and
- (iii) $P3$ is near the exit of the test section at $l/d = 540$.

Three frictional pressure gradients were calculated from the measured wall pressures $P1$, $P2$ and $P3$, as follows:

- $(dP/dx)_{\text{exp},1}$: based on pressure measured at locations 1 and 3;
- $(dP/dx)_{\text{exp},2}$: based on pressure measured at locations 1 and 2;
- $(dP/dx)_{\text{exp},3}$: based on pressure measured at locations 2 and 3;

For most of the experimental test runs, the pressure gradient was calculated as the arithmetic average of the three measured pressure gradients. In 8 of the 90 test runs (test runs 45, 46, 47, 48, 73, 81, 94, and 107), the measured wall pressures at location 3 were in the lower 10% range of the pressure transducer. These readings were not considered accurate. Therefore, for these test runs, the pressure gradient was taken as $(dP/dx)_{\text{exp},2}$, which is calculated from pressure readings taken at locations 1 and 2.

In this study, there was no change in elevation (horizontal test tube) and no change in momentum along the test tube (adiabatic flow in constant area test tube).

²¹ Static pressure was also measured at the air supply line. This pressure was used to correct for variations of the AVT readings with the system pressure.

Therefore, the measured pressure gradients represent the frictional component of the total pressure loss (see Section 3.3).

The measured frictional pressure gradients are shown in Fig. 6.16. The data are presented in terms of the measured pressure gradients plotted against the average pressure gradient for each test run. Figure 6.16 shows that most of the measured pressure gradients fall within $\pm 15\%$ of the average pressure gradient. These variations are generally within the expected uncertainties associated with measurements of the two-phase flow pressure (Collier and Thom, 1994).

The calculated uncertainties in the measured pressure gradients are presented in Appendix B, Table B.5. The size of the error bars is smaller than the size of the symbols used to represent the data points in Fig. 6.16.

Since pressure tap 1 is located only $3\ l/d$ from the entrance of the test tube, entrance length effects are measured in $(dP/dx)_{exp,1}$ and $(dP/dx)_{exp,2}$. However, because of the overall length over which the pressure gradients are calculated, i.e., $540\ l/d$ for $(dP/dx)_{exp,1}$ and $410\ l/d$ for $(dP/dx)_{exp,2}$, the effects are expected to be negligible.

6.3.2 Comparison with Predictions of Generalized Models

The measured frictional pressure gradients are compared with predictions of the following generalized models: (i) homogeneous flow model; (ii) Lockhart–Martinelli model; (iii) empirical correlation of Friedel (1979).

These models are presented and discussed in Chapter 3.

Method used to Assess Accuracy of Models in Predicting the Measurements

In this study, a linear regression analysis method was used to evaluate the accuracy of the models by comparing predicted to measured values. The predicted values were fitted with a straight line that intercepts the origin. The following are the parameters of the regression analysis:

- (i) The regression coefficient (slope of the best-fit line) indicates the average deviation of the predicted values from the measurements. The closer the regression coefficient is to 1 (45° line) the better the predictions are.
- (ii) The coefficient of determination (R^2) indicates the strength of the linear relation between the predicted and the measured parameters.
- (iii) The standard deviation of the residuals is a measure of the spread around the best-fit line.

This method provides a good measure of the model accuracy over the entire range of the measured data and is not affected by the concentration of the data in one range or another.

Homogeneous Model

Predictions of the friction losses using the homogeneous model depend on the formulation used to estimate the mean viscosity of the mixture. As discussed in Section 3.3, five formulations were proposed to estimate the mean viscosity of the mixture. Figure 6.17 compares the measured friction losses with the predictions of the homogeneous model using the five definitions of the mean viscosity of the mixture given by Eq. 3.23a to Eq. 3.23e. The solid line represents the regression line, and the broken line indicates a regression coefficient equal to 1 (the 45° angle), i.e., perfect prediction.

From the homogeneous model equation, an increase of the mixture viscosity leads to a higher friction factor, thus higher frictional pressure loss. The simple definition proposed by Owens et al. (1962), which assumes that the mean mixture viscosity is equal to the viscosity of the liquid, results in overpredictions of the friction losses. Figure 6.17c shows that on average, the formulation of Owens et al. (1962) results in overpredictions of the friction losses by over 40%. Similarly, the formulation of Chicchitti et al. (1960), which is simply a variation of the formulation of Owen et al. (1962) (see Section 3.3), also results in overpredictions of the friction losses by about 40%, see Fig. 6.17b. As

expected, at high gas mass velocities, the formulation of Cicchitti et al. (1960) resulted in slightly better predictions than that of Owens et al. (1962). This is simply because Cicchitti et al. (1960) assumed that at high mass quality, the mean viscosity of the mixture approaches that of the gas phase (see Fig. 3.1).

Figure 6.17*d* shows that the formulations proposed by Dukler et al. (1964) results in underpredictions of the friction losses by about 19%. Dukler et al. (1964) suggested that the mean viscosity of the mixture be prorated with the volumetric fractions of the gas and liquid phases ($\mu_m = \beta \mu_g + (1-\beta) \mu_l$). This definition ignores the fact that the liquid phase tends to cover the wall of the tube. In annular flow, the liquid phase covers the entire wall of the tube. This may explain why the formulation proposed by Dukler et al. (1964) results in underpredictions of the friction losses, especially at high mass velocities of the gas phase (conditions typical of the annular flow regime).

The formulations proposed by McAdams et al. (1942) and Beattie and Whalley (1982) resulted in reasonable predictions of the friction losses (see Fig. 6.17*a* and Eq. 6.17*e*). The formulation of McAdams et al. (1942) resulted in overpredictions of the friction losses by about 11%, while on average, the formulation of Beattie and Whalley (1982) resulted in accurate predictions of the friction losses (less than 2% deviation). Ali et al. (1993) examined the same formulations of mixture viscosity considered in this study and reached similar conclusions. They compared predictions of the homogeneous model with experimental data for adiabatic air–water flow in a narrow channel and found that the formulation of Beattie and Whalley (1982) gave accurate predictions of the friction losses.

The results of the regression analysis are summarized in Table 6.2.

Table 6.2: Results of the linear regression analysis.

Flow regime	Regression coefficient	Coefficient of determination (R^2)	Residual standard deviation (kPa/m)
McAdams et al. (1942)	1.11	0.97	0.378
Cicchitti et al. (1960)	1.40	0.97	0.474
Owens et al. (1962)	1.42	0.97	0.453
Dukler et al. (1964)	0.81	0.97	0.264
Beattie and Whalley (1982)	0.99	0.96	0.351

The coefficient of determination (R^2) of all the models is ~ 0.97 , which indicates a strong linear relation between the predicted and the measured parameters. The standard deviation of the residuals ranged from 0.264 to 0.474 kPa/m.

In summary, the above comparison indicates that out the five definitions of the mean mixture viscosity considered in this study, the definitions proposed by McAdams et al. (1942) and Beattie and Whalley (1982) result in reasonable agreement with the measured frictional pressure losses. Therefore, only these formulations are considered for further comparison with other models in the following sections.

Lockhart–Martinelli Correlation

The empirical correlation proposed by Lockhart and Martinelli (1949) for the prediction of the frictional pressure loss is given in Section 3.4. In this study, the representation suggested by Chisholm et al. (1967) for the Lockhart–Martinelli correlation, Eq. 3.26, is used to calculate predictions of the friction losses.

Figure 6.18 compares the measured average frictional pressure gradients with predictions of the Lockhart–Martinelli correlation. The predicted frictional pressure gradients are, on average, about 9% higher than the measured values. Again here, the coefficient of determination is ~ 0.97 , and the residual standard error is ~ 0.380 kPa/m. This is discussed further later in this section.

Friedel (1979) Correlation

Figure 6.19 compares the measured average frictional pressure gradients with predictions of the empirical correlation proposed by Friedel (1979), and shows that Friedel's correlation overpredicts the friction losses in over 90% of experimental test runs. On average, Friedel's (1979) correlation predicts frictional pressure gradients about 34% higher than the measured values. The coefficient of determination is ~ 0.94 , and the residual standard error is ~ 0.610 kPa/m. This is discussed further later in this section.

Summary and Discussion

Table 6.3 summarizes the results of the linear regression analysis discussed above. The regression coefficient, the coefficient of determination, and the standard error of the residuals are given for the homogeneous model, the Lockhart–Martinelli model, and the empirical correlation of Friedel (1979). The homogeneous model is considered only with the formulations of the mixture viscosity proposed by McAdams et al. (1942) and Beattie and Whalley (1982), as discussed previously.

Table 6.3: Results of the linear regression analysis of the friction losses.

Model or correlation	Regression coefficient	Coefficient of determination (R^2)	Std. Error of residuals (kPa/m)
Homogeneous (McAdams et al., 1942)	1.11	0.97	0.378
Homogeneous (Beattie and Whalley, 1982)	0.99	0.96	0.351
Lockhart–Martinelli (1949)	1.09	0.97	0.380
Friedel (1979)	1.34	0.94	0.610

Considering the uncertainties associated with measurement and prediction of two-phase flow parameters, Table 6.3 shows that the homogeneous and Lockhart–Martinelli models provide reasonable agreement with the measured friction losses.

The good agreement obtained between the Lockhart–Martinelli model and the experimental data is possibly due to the fact that the Lockhart–Martinelli model was based on experimental data taken under conditions similar to those of the present study (adiabatic systems in horizontal tubes and at near-standard atmospheric conditions).

For the homogeneous model, the literature seems to indicate that it has been used with notable success to predict the pressure drop, including the friction losses. Giot (1981) also indicated that the homogeneous model was successfully used to predict the friction losses in high-pressure boiling flows. As mentioned earlier, Ali et al. (1993) concluded that using the formulation proposed by Beattie and Whalley (1982), the homogeneous model gave accurate predictions of their measurements of friction losses in a narrow horizontal channel. Whitcutt and Chojnowski (1973) conducted an experimental investigation, in a boiling water facility that covered a wide range of mixture mass velocity (400–3500 kg/m²·s), system pressure (110–205 bars), and exit quality (up to 80%). They indicated that by setting the dynamic viscosity of the mixture equal to that of water, the $\mu_h = \mu_f$ formulation suggested by Owens et al. (1962), their pressure drop measurements were reasonably well correlated by the homogeneous model.

Some investigators attributed the notable success of the homogeneous model to compensating errors in the model. Whalley (1987) suggested that the success of the homogeneous model is fortuitous. Nevertheless, the success of the homogeneous model in predicting the friction losses is substantiated by a number of experimental investigations, including the present one. This success cannot be considered haphazard and may be the result of the following two factors: (i) the model is based on the well-established and well-tested formulations of single-phase flow, and (ii) the definition of the viscosity of the mixture is used to adjust for the effects of any variations that may be caused by, among other factors, the flow patterns.

Some investigators, such as Whalley (1986) and Collier and Thome (1994), suggested that Friedel's (1979) correlation provides the best estimates of the friction losses. As discussed above, this is not the conclusion of this study. Friedel's (1979)

correlations, in over 90% of the test runs, almost consistently overpredict the measured frictional pressure gradients.

This different conclusion may be explained by the fact that Friedel's (1979) correlation was developed based on experimental data taken in vertical test sections for boiling flow under high system pressures. Under such boiling flow conditions, vapour bubbles are created at the heating surface (nucleation sites), which is usually the wall of the conduit. Before these bubbles detach from the heating surface, they result in an increase of the wall surface or apparent roughness of the surface, thus increasing friction losses. It should be noted that high pressure and turbulent conditions encourage the bubbles to detach from the heating wall. Under annular flow boiling conditions, the waves at the liquid film and the mass transfer at the interface (evaporation or break-up of waves and entrainment) are expected to increase the interfacial drag forces.

In addition, when pressure measurements are taken in vertical flow channels, the gravitational component of the pressure drop (and in the case of boiling and condensing flow, the momentum component of the pressure drop) has to be subtracted from the measured total pressure drop. Therefore, Friedel (1979) may not have completely removed the gravity and momentum losses from the experimental data used in developing his model. This introduces additional errors and will certainly increase the scatter. Hewitt (1982) indicated that Friedel (1979) compared the predictions of his correlation with a large data bank (over 25000 test points). For single-component flow, Friedel (1979) calculated a standard deviation of ~30%, and 40–50% for two-component mixtures. The highest scatter was observed for horizontal flow and especially two-component horizontal flow, as shown in Fig. 2.2.6 in Hewitt (1982).

6.3.3 Comparison with Predictions of Regime-specific Models

In this section, predictions of regime-specific correlations are compared with the measured friction losses.

Observed Variations of Friction Losses with Flow Regimes

The range of the measured frictional pressure gradients for the observed flow patterns is given in Table 6.4, along with the number of observations for each flow pattern. Table 6.4 shows the dependency of the measured frictional pressure gradients on the flow patterns. This dependency is also illustrated in Fig. 6.20, which shows the two-phase flow friction multiplier ϕ as a function of the Lockhart–Martinelli parameter X . The friction multipliers are calculated from the measured pressure gradients. Recognizing that transition regions exist (as discussed in Chapter 5), ϕ varies with the flow patterns (Fig. 6.20). High values of ϕ were calculated for the stratified wavy and annular flow regimes (up to about 200) for X less than 1.5. The lowest values of ϕ were calculated for the plug flow regime ($\phi < 2$ for $X > 5$). For slug flow, values of ϕ ranged from ~ 3 to 5 for X in the range of 0.8 to 20.

Table 6.4: Range of measured average frictional pressure gradients for different flow patterns.

Flow regime	No. of test runs	Range of measured friction pressure gradients (kPa/m)
Annular flow	18	0.941 – 7.556
Plug flow	33	0.039 – 1.145
Slug flow	31	0.140 – 4.197
Stratified wavy flow	8	0.012 – 0.663
All test runs	90	0.012 – 7.556

Applicability of Generalized Models to Predict the Friction Losses for Each Flow Regime

Since it was shown that friction losses are regime-dependent, predictions of the homogeneous, Lockhart–Martinelli and Friedel’s (1979) models were compared to the measured pressure losses for each flow regime individually. The results of this comparison are shown in Fig. 6.21 for the annular, plug, slug, and stratified wavy flow patterns. Results of the regression analysis are given in Table 6.5. Two test runs were not considered in the calculated values of the regression parameters given in Table 6.5 for the

stratified wavy flow regime. The two data points (test runs 51 and 52 shown in Fig. 6.21*d* with measured pressure gradients higher than 0.4 kPa/m) are considered to be outliers. These two test runs are classified as transitional flow between stratified wavy and annular flow regimes.

Table 6.5: Variations of the regression coefficient and the coefficient of determination of models predictions with the flow patterns.

Model	Regression coefficient (Coefficient of determination, R^2)			
	Annular flow	Plug flow	Slug flow	St. wavy flow
Homogeneous (McAdams et al.)	1.10 (0.95)	1.04 (0.90)	1.18 (0.89)	1.38 (0.95)
Homogeneous (Beattie et al.)	1.10 (0.95)	1.07 (0.91)	1.09 (0.92)	1.16 (0.95)
Lockhart–Martinelli (1949)	0.96 (0.94)	0.89 (0.90)	1.07 (0.89)	0.91 (0.95)
Friedel (1979)	1.25 (0.95)	1.76 (0.84)	1.67 (0.90)	1.38 (0.95)

Figure 6.21*a* shows that for annular flow, the Lockhart–Martinelli model provide reasonable predictions of the friction losses. Friedel’s (1979) correlation overpredicts the measurements by about 25%. This supports the argument presented previously that suggests that Friedel’s (1979) correlation is more appropriate for use at high gas and liquid mass velocities. In annular flow, the waves at the gas–liquid interface are expected to increase the interfacial drag forces.

For plug and slug flow regimes, Friedel’s (1979) correlation significantly overpredicts the measured frictional pressure gradients, especially plug flow regimes (more than twice the measurements for some of the data points). For plug flow, the best predictions of the friction losses were obtained using the homogeneous model and the formulation of the mixture viscosity of McAdams et al. (1942) or Beattie and Whalley (1982). For slug and annular flow regimes, the best predictions were obtained using the Lockhart–Martinelli correlation. For stratified wavy flow, the best predictions were obtained using the Lockhart–Martinelli correlation and the homogeneous model using the definition of the Beattie and Whalley (1982) for the viscosity of the mixture.

Correlations of Baker (1961)

The flow regime-dependent correlations of Baker (1961) are given in Section 3.4.2. The predictions of these correlations are compared with the results of the present study in Fig. 6.22. The results are presented in terms of the two-phase flow friction multiplier ϕ_g as a function of X .

Figure 6.22a shows that Baker's (1961) correlation provides good predictions of the friction multiplier for the annular flow regime as calculated using Eq. 3.33e. Baker's (1961) correlations for dispersed bubble (Eq. 3.33c) and plug (Eq. 3.33b) flow regimes are compared with the plug flow data of the present study in Fig. 6.22b. Both of the dispersed bubbly and plug flow correlations of Baker (1961) overpredict the measured friction losses. As shown in Fig. 6.22b, the bubbly and plug flow correlations of Baker (1961) predict similar friction losses. This supports the argument presented in Chapter 5, which suggests that dispersed bubbly flow may not exist in horizontal channels. What was classified as dispersed bubbly flow may in fact be what has been described in this document as scalloped plug flow.

For stratified wavy and slug flow regimes, Baker (1961) suggested the use of the Lockhart–Martinelli correlation if the pipe diameter is 25 mm or smaller. Figures 6.22c and 6.22d show the predictions of the Baker (1961) correlations for slug and stratified wavy flow regimes as well as the predictions of the Lockhart–Martinelli correlation. Baker's (1961) correlations do not distinguish between stratified smooth and stratified wavy flow regimes. Figure 6.22c shows that Baker's (1961) correlation for slug flow (Eq. 3.33d) significantly underpredicts the measured friction losses. The calculated ϕ_g based on the Lockhart–Martinelli model shows better agreement with the present data. Similar results are obtained for the stratified wavy flow data; Baker's (1961) correlation for stratified flow (Eq. 3.33a) significantly underpredicts the measurements, while better agreements were observed between the present data and the predictions of the Lockhart–Martinelli correlation. This is not surprising since Baker's (1961) correlations were

developed for large-diameter pipes, and Baker (1961) also recommended using the Lockhart–Martinelli correlation over his correlations for slug and stratified flow regimes if the diameter of the pipe is 25 mm or less.

Correlations of Beattie (1973)

Beattie (1973) developed expressions for the friction multipliers as a function of the mass quality of the mixture and the density and viscosity of the phases. As discussed in Section 3.4.2, the correlations are based on simplifying assumptions about the distribution of the phases. Because the flow pattern classification of Beattie (1973) is not consistent with the classification used in this study, his four correlations are compared with the data for all the regimes in Fig. 6.23. This comparison shows that only Beattie's (1973) correlation proposed for annular flow and the stratified wavy gas–liquid interface shows reasonable agreement with the experimental data of the present work. Figure 6.23 also shows that the correlation proposed by Baker (1961) for post burnout conditions, which represents all vapor (or gas) flow, significantly underpredicts the two-phase flow friction multipliers, as expected.

Correlations of Bandel (1973)

As discussed in Section 3.4.2, the empirical correlation of Bandel (1973) classifies the flow into a stratified, annular, or transitional regime based on the calculated friction losses. The correlation compares the calculated frictional pressure gradient with a lower limiting pressure gradient for annular flow and an upper limiting pressure gradient for stratified flow (Eq. 3.35a and 3.35b). Bandel's (1973) classifications of the present data are compared with the classifications of the present study in Fig. 6.24.

Bandel's (1973) classifications of the annular flow are in good agreement with the present data. The pressure gradient calculated using Bandel's (1973) correlation for all the test runs identified in this study as annular flow are higher than the lower limiting frictional pressure gradient for annular flow represented by the solid line in Fig. 6.24. Figure 6.24 also shows that only two test runs are classified by Bandel's (1973)

correlation as stratified flow, i.e., the calculated pressure gradient for these test runs is lower than the upper limiting pressure gradient for stratified flow represented by the broken line. The test runs are classified in this study as stratified wavy flow. As discussed in Chapter 5, none of the test runs were classified as stratified smooth.

The remaining test runs are classified by Bandel's (1973) correlation as transitional flow. The pressure gradients for these test runs are calculated by interpolating between pressure gradients calculated for annular and stratified flow (see Appendix A).

Figure 6.25 compares the predictions of Bandel's (1973) model with the measured pressure gradients for all test runs. The solid line represents the best-fit line. The predicted pressure gradients are, on average, about 32% lower than the measured values. The coefficient of determination (R^2) is ~ 0.87 .

Figure 6.26 compares the predicted and measured pressure gradients for different flow regimes. Better agreements between measured and predicted frictional pressure gradients are obtained for the annular and stratified wavy flow regimes than for plug and slug flow regimes. For annular and stratified wavy flow, the model underpredicts the measurements by 24 and 14%, respectively, and underpredicts the pressure gradients by over 57% for slug flow and $\sim 80\%$ for plug flow. As discussed earlier, the model considers these regimes as transitional flow and interpolates between the pressure gradients calculated for the stratified and annular flow patterns.

6.4 Two-fluid Models for Friction Losses

This section discusses results for the two-fluid models presented in Section 3.5.

6.4.1 Frictional Pressure Gradient

An expression for the frictional pressure gradient was obtained from the basic momentum equations by eliminating the interfacial friction terms (Eq. 3.40). The contact perimeters and the hydraulic diameters were calculated based on the disposition of the

phases for each flow regime and using the measured void fractions. The shear stresses and the friction factors were calculated using the conventional single-phase flow relationships.

The pressure gradients calculated using the two-fluid models are compared with the measurements in Fig. 6.27 for the stratified wavy, slug, annular, and plug flow regimes. The horizontal bars in Fig. 6.27 shows the calculated 95% Confidence level uncertainties in the measured pressure gradients. Note that for the slug, annular, and plug flow data points, the size of the error bars is smaller than the size of the symbols.

(a) Stratified Wavy Flow Regime

For stratified wavy flow, Fig. 6.27a, the two-fluid model underpredicts the measured pressure gradients. Most of the observed discrepancies are due to the effects of the conditions of the gas–liquid interface on the velocity profile in the phases. The relative velocity between the phases at the interface affects the velocity profiles in the phases such that the conventional single-phase flow equations for the wall friction factors are invalidated. The conventional equations for wall friction factors in single-phase flow were developed for full pipe flow under fully developed turbulent conditions.

The calculated average drift velocities for the observed flow regime were presented and discussed in Section 6.2.5 and are conveniently list in Table 6.6.

Newton and Behnia (1998) also observed similar effects of the conditions of the interfaces on the liquid–wall shear stresses. Based on measurements of the gas–wall shear stress, the pressure drop and the thickness of the liquid film, they calculated the liquid–wall and the interfacial shear stresses. The results of their study showed the conditions of the liquid film were strongly influenced by the interfacial shear stress. Newton and Behnia (1998) found a strong relationship between the interfacial and the liquid–wall friction factors and concluded that expressions developed for full pipe flow were not applicable for the liquid region in stratified two-phase flow.

As indicated earlier, most of the observed discrepancies between the measured and the predicted pressure gradients are due to the effects of the conditions of the gas–liquid interface on the velocity profile in the phases. However, as shown in Fig. 6.27a, the measured pressure gradients are significantly underpredicted for test runs 51 and 52. Both test runs are characterized by higher relative velocity between the phases at the interface than other stratified wavy flow test runs. The recorded videos for these test runs show that the relatively faster moving air causes the waves to breakup and some water droplets were entrained and deposited on the top wall of the channel. These test runs are near the transition region between the stratified wavy and the annular flow regimes.

Ignoring test runs 51 and 52, Fig. 6.27a shows that the two-fluid model underpredicts the measured pressure gradients by $\sim 45\%$. The coefficient of determination of the fit (R^2) is ~ 0.98 .

Table 6.6: Calculated average drift velocities for the flow patterns observed in this study.

Flow Regime	Average drift velocity (V_{gi}) (m/s)
Stratified wavy flow	0.466
Slug flow	0.991
Plug flow	0.068
Annular flow	3.688

(b) Slug Flow Regime

For slug flow, Fig. 6.27b, the two-fluid model also underpredicts the measured pressure gradients. Similarly to stratified wavy flow, the relative velocity between the phases at the interface affects the velocity profiles in the phases such that the conventional single-phase flow definition of the wall friction factors are invalidated.

Figure 6.27b shows that the predicted pressure gradient for test run 101 differs significantly from the pressure gradients predicted for all other slug flow test runs. Test run 101 is near the transition region between slug and annular flow regimes. In accordance with the *Chauvenet's criterion* (Young, 1962), test run 101 can be rejected because of its large deviation from the mean. Chauvenet's criterion recommends that an observation should be disregarded if its deviation from the mean is so large that the probability of occurrence of a deviation that large is less than 1/20 (or 5%). This criterion can be used only once on a set of data and the number of observations that can be eliminated is limited and depends on the size of the set of data. The standard deviation (S), calculated based on all the slug flow data points from the best-fit line, is ~ 0.34 kPa/m. Test run 101 deviates from this best-fit line by ~ 0.89 kPa/m, which is larger than $2S$ (0.68 kPa/m). Therefore, in accordance to Chauvenet's criterion test run 101 was discarded.

Correlation of predicted pressure for the slug test runs was repeated without test run 101. Figure 6.27b, shows that the two-fluid model underpredicts the measured pressure gradients by $\sim 41\%$. The coefficient of determination of the fit (R^2) is ~ 0.89 .

It should be noted that, although V_{gi} for slug flow is higher than that calculated for stratified wavy flow, on average, the two-fluid model gave slightly better predictions of the pressure gradients in slug flow than in stratified wavy flow, i.e., underpredicted the measurements by $\sim 41\%$ for slug flow compare to $\sim 45\%$ for stratified wavy flow. This may be explained by the fact that the two-fluid model ignores the fraction of the liquid phase entrained in the gas, which is more significant in slug flow than in stratified wavy flow. This results in higher liquid velocity at the wall and thus higher pressure gradients for the slug flow test runs.

(c) Annular Flow Regime

The two-fluid model used to calculate the frictional pressure gradients in annular flow assumes all the liquid flows in an evenly distributed film around the wall of the pipe.

The film thickness was calculated using the average void fraction measured using the optical probes.

Figure 6.27c shows that the two-fluid model significantly overpredicts the measured pressure gradients for test runs 55, 58 and 59, while it underpredicts the measurements for test run 57. Overpredictions of the frictional pressure gradient are most likely due the fact that the two-fluid model ignores the fraction of the liquid entrained in the gas. Assuming all the liquid flows in the film around the wall of the pipe results in a higher liquid velocity and, thus, higher frictional pressure gradients.

The fraction of the liquid entrained in the gas, for each of the annular flow test runs, was estimated using the expression for the critical Reynolds number ($Re_{f,crit}$) proposed by Govan et al. (1988). As discussed in Section 3.5.1(a), Govan et al. (1988) suggested that entrainment occurs when the film's Reynolds number (Re_f) reaches 429 (calculated using Eq. 3.43). Table 6.7 summarizes the results of an analysis performed to estimate the entrained fraction of the liquid film for each of the test runs. Note that, in Table 6.7, δ_{exp} is the average thickness of the liquid film calculated using the measured void fraction, and $\dot{m}_l$ is the measured mass flow rate of the liquid. The film Reynolds number (Re_f) was calculated as

$$Re_f = \frac{d J_l \rho_l}{\mu} \quad (6.5)$$

As discussed in Collier and Thome (1994), the above expression for Re_f is equivalent²² to

$$Re_f = \frac{4 \delta_{exp} u_l \rho_l}{\mu} \quad (6.6)$$

²² Eq. 6.5 and Eq. 6.6 gave similar values for Re_f .

Table 6.7 also lists the critical film thickness (δ_{crit}) calculated using Eq. 6.6 and the critical Reynolds number suggested by Govan et al. (1988), i.e., $Re_{f,crit} = 429$. As shown in Table 6.7, with the exception of test run 53, liquid entrainment was predicted to occur in all test runs ($\delta_{exp} > \delta_{crit}$). The mass flow rate of the liquid entrained in the gas (if $\delta_{exp} > \delta_{crit}$) was estimated as

$$\dot{m}_e = \dot{m}_t \left(1 - \frac{\delta_{crit}}{\delta_{exp}} \right) \quad (6.7)$$

In summary, this analysis show that a significant fraction of the liquid phase was entrained along the test section in the gas, which agrees with the observations of the recorded videos of the flow. Therefore, assuming all the liquid flows in the film is expected to result in overpredictions of the frictional pressure gradients, which is the case of most of the annular flow test runs, particularly for test runs 55, 58, and 59 (see Fig. 6.27c). Note that it is not clear why the model underpredicts the pressure gradient for test run 57.

Table 6.7: Estimates of the fraction of liquid entrained in the gas for the annular flow test runs.

Test run	δ_{exp}	$\dot{m}_t$	Re	δ_{crit}	$\delta_{exp} - \delta_{crit}$	$\dot{m}_e$	$\frac{\dot{m}_e}{\dot{m}_t}$
No.	(m)	(kg/s)	(δ_{exp})	(m)	(m)	kg/s	
53	1.95E-04	0.008	387	2.15E-04	-1.94E-05	0.0000	0.0%
54	1.95E-04	0.012	556	1.49E-04	4.59E-05	0.0027	23.5%
55	4.56E-04	0.102	4885	3.93E-05	4.17E-04	0.0933	91.4%
56	1.05E-03	0.141	6723	6.41E-05	9.85E-04	0.1319	93.9%
57	1.53E-03	0.218	10424	5.90E-05	1.47E-03	0.2094	96.1%
58	9.62E-04	0.303	14487	2.74E-05	9.34E-04	0.2941	97.2%
59	1.29E-03	0.415	19881	2.64E-05	1.26E-03	0.4070	98.0%

As discussed in Section 3.5.1, Wallis's (1969) correlation suggests that, for $Re_f > 3000$, the onset of liquid entrainment occurs at $J_g = 28.3$ m/s. The calculated values of J_g for the annular flow test runs are less than 24 m/s. Therefore, predictions of the

correlation proposed by Wallis (1969) do not agree with the results of the above analysis and observations from the present study.

Whalley (1987) presented a correlation for predicting the fraction of the liquid phase entrained in the gas. This correlation predicts lower fractions of the entrained liquid than those listed in Table 6.7.

(d) Plug Flow Regime

For the plug flow test runs, the frictional pressure gradients were calculated using the two-fluid model developed in Section 3.5.1(c). This model assumes that the phases travel at the same velocity. As discussed in Chapter 5, this assumption is validated by the detailed analysis of the recorded videotapes of the plug flow regime (the elongated gas bubbles were entrained along the test section by the liquid phase). The calculated average drift velocity for the plug flow test runs is less than 0.07 m/s (see Table 6.6). The plug flow model divides the flow along the test section into single-phase flow and two-phase flow regions (see Fig. 3.10). The average pressure gradient along the test section was then calculated as the weighted-average of the pressure gradients in the single-phase and two-phase regions (see Section 3.5.1(c)). The average length of the single-phase and two-phase regions, given by the ratios $L_B/(L_B+L_S)$ and $L_S/(L_B+L_S)$, were determined experimentally from the output of the optical probes as described in Section 6.2.5.

In the single-phase regions, the frictional pressure gradient was calculated using the conventional single-phase flow relationships (Eq. 3.47) with the velocity of the liquid phase calculated based on the measured liquid flow rate and the overall void fraction measured using the optical probes (α).

In the two-phase flow regions, the pressure gradients were calculated in a similar manner as those of the stratified flow regime with the hydraulic diameters, the contact perimeters, and the average velocities of the phases based on the average void fraction within the two-phase flow regions (α_p). As discussed in Section 6.2.5, α_p was determined

experimentally from signals of the optical probes. An expression for α_p as a function of α was given in Eq. 6.3. It is interesting to note that, as shown in Fig. 6.28, the calculated values of α_p for plug flow test runs are in good agreement with the homogeneous void fraction (α_h), which assumes that there is no relative velocity between the phases ($S = 1$). The velocities of the phases calculated using α_p agree well with the experimental observations, which showed that the gas and liquid phases move at relatively the same velocity.

The pressure gradients predicted using the plug flow model are compared with the measurements in Fig. 6.27d. The plug flow model provides good predictions of the frictional pressure gradients. Results of the linear regression analysis gave a regression coefficient of ~ 1 , with a coefficient of determination (R^2) of ~ 0.95 , and a 95% confidence-level standard deviation equal to 0.074 kPa/m.

In plug flow, the interfacial interactions between the phases are small since the phases travel at the nearly the same velocity. The velocity profiles in the phases were not affected by the conditions of the interfaces. Therefore, the conventional single-phase flow expression for the wall friction factor can be used to calculate the liquid- and gas-wall friction factors in plug flow regime.

6.4.2 Correlation of the Interfacial Friction Factor

The approach used in the present study to correlate the interfacial friction factors, for the different flow patterns, based on measurements of the frictional pressure gradients and the average void fractions, is described in Section 3.5.2. In this approach, it was recognized that the conventional single-phase flow expressions for the wall friction factors, which were developed for full pipe flow, are not applicable for two-phase flow except for plug flow regime. Therefore, the interfacial friction factor was correlated in terms of a waviness coefficient C_w , which was introduced to account for; (i) the effects of the relative velocity of the gas-liquid interfacial conditions on the velocity profiles in the phases, and (ii) the increased friction factor as a result of the waves.

The following sections describe the correlation results of interfacial friction factors for the stratified wavy, annular, and slug flow regimes. For plug flow, since the phases travel at relatively the same velocity, the interfacial shear stress is negligible.

(a) Stratified Wavy Flow Regime

The calculated values of C_w for the stratified wavy flow test runs are presented in Fig. 6.29 as function of the absolute differences between the average velocities of the gas and liquid phases ($|u_g - u_l|$). Figure 6.30 compares the calculated values of C_w with the experimental results of Andritsos and Hanratty (1987), and Andritsos (1992). The results are presented as a function of the gas superficial velocity (J_g). The data of Andritsos and Hanratty (1987) are for a 95.3-mm pipe diameter and a liquid viscosity of 0.001 Pa·s (1cP), and the data of Andritsos (1992) are for a 25.4-mm pipe diameter and a liquid viscosity of 0.0045 Pa·s (4.5cP). Considering that either the pipe diameter or the liquid viscosity is different from that of the present study, Fig. 6.30 shows that the calculated values of C_w are in reasonable agreement with those of Andritsos and Hanratty (1987), and Andritsos (1992).

The calculated values of the equivalent interfacial friction factor ($f_{i,eq} = C_w f_i$) based on the experimental data of the present study are compared with the predictions of the correlations of Newton and Behnia (1998) and Vlachos et al. (1997) in Figs. 6.31 and 6.32. The results are presented as a function of the interfacial Reynolds number Re_i , calculated based on the relative velocity between the phases, the properties of the gas phase, and the hydraulic diameter of the gas flow area (d_{eg}).

Figure 6.31 shows that the correlation of Newton and Behnia (1998) for the 50-mm pipe (Eq. 3.51a) underpredicts the present data for $Re_i < 5,000$, and overpredicts the present data for $Re_i > 5,000$. However, the correlation of Newton and Behnia (1998) for 80-mm pipe (Eq. 3.51b) consistently underpredicts the calculated values of $f_{i,eq}$ of the present data.

Figure 6.32 shows that the correlation of Vlachos et al. (1997), Eq. 3.50, provides good agreement with calculated values of $f_{i,eq}$ of the present data. The experimental work of Vlachos et al. (1997) was presented in Section 3.5.2. The solid line in Fig. 6.32 represents the power-fit of the calculated $f_{i,eq}$ from the present data for $Re_i > 5000$. The best-fit equation is

$$f_{i,eq} = 6.88 Re_i^{-0.5345} \quad (6.8)$$

The coefficient of determination for this power-fit (R^2) is ~ 0.89 . It should be noted that Re_i is as defined as

$$Re_i = \frac{\rho_g |u_g - u_l| d_{eg}}{\mu_g} \quad (6.9)$$

(b) Annular Flow Regime

The calculated C_w for the annular flow test runs are presented in Fig. 6.33 as a function of $|u_g - u_l|$. Higher values of C_w are calculated for the annular flow regime than the stratified wavy flow regime. This reflects the higher drift velocities associated with the annular flow regime (see Table 6.6). The highest values of C_w were calculated for test runs 58 and 59 (see Fig. 6.33). The recorded videos of the flow show that both of these test runs have significant droplet flow, i.e., a larger fraction of the liquid phase is entrained in the gas phase. Note that the calculation of C_w and $f_{i,eq}$ assume that all the liquid flows in the film around the wall of the pipe. This assumption is expected to result in higher values of $f_{i,eq}$. The film thickness was calculated based on the measured void fractions.

The calculated values of $f_{i,eq}$ are compared with the predictions of different correlations in Fig. 6.34. Note that data for test runs 58 and 59 are not included in Fig. 6.34. The correlation of Hart et al. (1989) predicts lower values of $f_{i,eq}$ than the present data. Predictions of the correlations of Wallis (1969) and Vlachos et al. (1997) are in reasonable agreement with the present data, especially at high values of Re_i .

The present data (without test runs 58 and 59) are best fitted with an exponential-fit. The equation of the best-fit line is

$$f_{i,eq} = 501.21 \exp(3 \times 10^{-4} Re_i) \quad (6.10)$$

The coefficient of determination for this exponential-fit (R^2) is ~ 0.995 .

(c) Slug Flow Regime

The calculated values of C_w for the slug flow test runs are presented as a function of $|u_g - u_l|$ in Fig. 6.35. For low values of the gas and liquid relative velocity ($|u_g - u_l| < 4 \text{ m/s}$), the calculated values of C_w decrease with an increase in $|u_g - u_l|$. For higher values of $|u_g - u_l|$, the calculated values of C_w are almost constant. This trend is also observed in Fig. 6.36, which shows variations of the calculated values of $f_{i,eq}$ as a function Re_i . The data are correlated as two separate sets. For $Re_i < 4000$, the data are best fitted with a linear-fit, and for $Re_i > 4000$ the data are best fitted with a logarithmic-fit.

$$\text{For } Re_i < 4000; \quad f_{i,eq} = -6.5 \times 10^{-3} Re_i + 18.244 \quad (6.11)$$

$$\text{For } Re_i > 4000; \quad f_{i,eq} = -1.45 \ln(Re_i) + 14.364 \quad (6.12)$$

The coefficient of determination (R^2) is ~ 0.746 for the linear-fit and ~ 0.38 for the logarithmic-fit. The lower values for R^2 is a result of the observed highly transient nature of the slug flow regime.

6.4.3 Summary

Analysis of the two-fluid model results clearly show that the interactions between the phases at the interface have considerable effects on the shear stresses between the phases and the wall of the channel. The two-fluid model, which was based on a simplified disposition of the phases and assumes no liquid entrainment at the interfaces,

predicts lower values of the frictional pressure gradients in the stratified wavy and slug flow regimes. For annular flow, the model overpredicts the pressure gradients for test runs that were characterized with significant entrainment of the liquid phase; particularly those near the transition region to slug flow regime. The interactions between the phases at the interfaces invalidate the use of the conventional single-phase flow equations for wall friction factors.

The model developed for plug flow gave accurate predictions of the frictional pressure gradients. In plug flow, the interfaces were smooth and there were no significant interactions between the phases. The conventional single-phase flow equation for the wall friction factors is applicable for the plug flow regime.

6.5 Interfacial Area Concentration in Stratified Wavy Flow Regime

The average thickness of the liquid layer (h) was measured from a number of the digitized videotaped frames for each of the stratified wavy flow test runs. The concentration of the gas–liquid interfacial area (a_i) was then calculated using Eqs. 3.56 and 3.57. For the range of stratified wavy flow conditions covered in the present experimental study, a_i varied between ~ 35 – $50 \text{ m}^2/\text{m}^3$. For an average liquid film thickness to diameter ratio (h/d) > 0.25 , the calculated values of a_i are greater than $45 \text{ m}^2/\text{m}^3$.

The calculated values of a_i are compared with those calculated from the measured void fractions in Fig. 6.37. With the exception of one data point, the image analysis of the digitized videotaped frames predicts higher values of a_i than measurements obtained using the average void fractions, especially for (h/d) < 0.25 (or $a_i < 45 \text{ m}^2/\text{m}^3$). At low values of h/d , the air–water interfaces in the pipe are usually concave because water is a wetting liquid, i.e., contact angle between water and an Acrylic surface is less than 90° . Also, low values of h/d are usually associated with higher velocities of the gas phase, which tends to spread the liquid film along the wall of the pipe. Figure 3.9 shows a concave gas–liquid interface in a pipe. Therefore, the values of a_i calculated using the

average void fractions measured along the centerline of the pipe and assuming that the gas–liquid interfaces are flat (i.e., horizontal), are expected to underpredict the true values of a_i .

Figure 6.37 shows better agreements for $(h/d) > 0.25$, for which the observed air–water interfaces were relatively flat.

The measured average thickness of the liquid film obtained from the digitized images was also used to calculate the average void fraction for the stratified wavy flow test runs. Figure 6.38 compares the calculated average void fractions obtained from the image analysis with measurements obtained from the optical probes. The measured void fractions are within $\pm 5\%$, with the optical probes providing slightly higher void fractions, especially at high values of α . Again here, this due to the fact that, at high values of α (i.e., lower values of h/d), the gas–liquid interfaces are concave.

The resolution of the image analysis technique is estimated in Appendix B at 0.4 mm^2 (equivalent to the area of 1 pixel). However, the accuracy of the measurements depends to a certain extent on the ability to clearly identify the interfaces in both the horizontal- and vertical-planes of the test section. The recorded videos show that the clarity on the images also depends on the prevailing flow regime. Clear video-images were obtained for stratified wavy and plug flow regimes. The videos of the annular and slug flow regimes require elaborate image analysis and edge detection techniques.

In summary, the videotapes of the horizontal- and vertical-planes of the flow inside the test section can be used to provide accurate measurements of the interfacial area concentration, and the local void fractions for the stratified wavy flow regime. The recorded videotapes for the plug flow regime can also provide accurate measurements of the interfacial area concentration. For slug and annular flow regimes, the recorded videos may require elaborate image analysis techniques and edge detection routines in order to obtain reasonable estimates of the interfacial area concentration.

Figure 6.1: Typical profiles of the local void fraction for a) Stratified wavy flow; b) Plug flow; c) Slug flow; and d) Annular flow.

Figure 6.2: A comparison between measured average void fraction using upstream and downstream sensors.

Figure 6.3: Comparison of measured average void fraction with experimental data of Franca and Lahey (1992).

Figure 6.4: Variation of measured and predicted void fractions with the Lockhart–Martinelli parameter (X).

Figure 6.5: Comparison between measured and predicted average void fractions.

Figure 6.6: Variation of the average void fraction with flow regimes.

Figure 6.7: Correlation of average void fraction with the drift-flux model.

Figure 6.8: Correlation of measured void fraction for stratified flow regime.

Figure 6.9: Correlation of measured void fraction for slug flow regime.

Figure 6.10: Correlation of experimental data of França and Lahey (1992) for slug flow and annular flow test runs with $J_g < 20$ m/s.

Figure 6.11: Correlation of measured void fraction for annular flow regime (present data and data of França and Lahey, 1992).

Figure 6.12: Time-averaged void fraction in the two-phase flow regions of plug flow as a function of the overall average void fraction.

Figure 6.13: Average fraction of two-phase flow segments in plug flow regime.

Figure 6.14: Correlation of measured void fraction for plug flow regime with modified drift-flux model variables.

Figure 6. 15: Correlation of plug flow data of Franca and Lahey (1992) with the modified variables of the drift-flux model.

Figure 6.16: Variations in the measured frictional pressure gradients.

Figure 6.17: Homogeneous model predictions of the friction losses for different formulations of the mixture viscosity.

Figure 6.18: Friction losses: comparison between experimental data and Lockhart-Martinelli model.

Figure 6.19: Friction losses: comparison between experimental data and Friedel correlation.

Figure 6.20: Variations of the two-phase flow friction multiplier with the observed flow patterns.

Figure 6.21: Comparison between measured and predicted friction losses for different flow regimes.

Figure 6.22: Comparison of measured two-phase flow friction multiplier with prediction of Baker (1961).

Figure 6.23: Comparison of measured two-phase flow friction multiplier with prediction of Beattie (1973).

Figure 6.24: Bandel's classifications of the flow patterns.

Figure 6.25: Comparison of Bandel's predictions with the measured frictional pressure gradients.

Figure 6.26: Comparison of the predictions of Bandel's model with measured pressure gradients for different flow regimes.

Figure 6.27: Pressure gradients predicted using the simplified equations of the two-fluid model.

Figure 6.28: Comparison of the homogeneous void fraction with the measured void fraction within the two-phase flow regions of plug flow regime.

Figure 6.29: Calculated waviness coefficient (C_w) for the stratified wavy flow test runs.

▲ Present Work ● Andritsos (1992) $d=9.53$ cm; 1cP. ● Andritsos and Hanratty (1987) $d=2.52$ cm; 4.5 cP.

Figure 6.30: Comparison of C_w with the experimental data of Andritsos et al. (1987, 1992).

Figure 6.31: Interfacial friction factors for the stratified flow test runs.

Figure 6.32: Comparison of the interfacial friction factors with the results of the correlation of Vlachos et al. (1997).

Figure 6.33: Calculated waviness coefficient (C_w) for annular flow test runs.

Figure 6.34: Interfacial friction factor for annular flow regime.

Figure 6.35: Calculated waviness coefficient for slug flow test runs.

Figure 6.36: Interfacial friction factor for slug flow regime.

Figure 6.37: Gas-liquid interfacial area concentration of stratified flow test runs.

Figure 6.38: Average cross-sectional void fractions calculated using image processing.

Conclusions and Recommendations

7.1 Main Conclusions

The main conclusions of the present experimental study are summarized in the following subsections.

7.1.1 Flow Patterns and Pattern Transitions

Conclusions reached in this study on two-phase flow patterns and pattern transitions are:

Identification of Flow Patterns

- Earlier identifications of the flow patterns were based on visual observations of the flow and were therefore considered subjective to the particular interpretation of the investigator. Techniques developed in the past few decades for objective identifications of the flow patterns are judged to have had only limited success.
- In the present study, a methodology was developed to identify the flow patterns. This methodology was based on video techniques and used detailed descriptions of the flow patterns, including their distinctive features and clear criteria used to distinguish between the regimes.

Observed Flow Patterns

- The observed flow patterns were classified in four major flow patterns: stratified wavy, plug, slug, and annular flow regimes. In general, these regimes are consistent with the most widely accepted classification of horizontal flow regimes, with the

exception of the dispersed bubble and the stratified smooth flow patterns, which could not be established in this experimental study.

- This study proposes that plug and slug flows be considered as separate regimes because they exhibit different phase distribution and flow characteristics and occur under different flow conditions.
- Based on the results of the present study and on experimental data from other sources, stratified smooth and dispersed bubble regimes may not exist in small diameter horizontal tubes ($d < 50$ mm).
- Stratified smooth and wavy flows were consolidated into one regime identified as stratified wavy flow. Stratified smooth flow is expected to be unstable except under almost stagnant conditions. In addition, from a modelling point of view, little is gained from this subdivision of the flow as both regimes have similar flow characteristics. The effects of the waves at the interfaces can be accounted for using a waviness coefficient.

Transition Regions

- The transitions between the flow regimes were observed to occur over a range of flow conditions (transition regions).
- Transition regions should be identified on the flow regime map. Predicting that the flow is in transition provides analysts with important information about the behaviour of the flow.
- These transition regions should not be viewed as separate flow regimes. The flow in these regions alternates between two or more flow regimes. Small variations in the flow conditions may let one of the regimes predominate and the transition occurs.

Observed Pattern Transition Mechanisms

The following transition mechanisms were observed in this study:

- stratified wavy to annular: this transition is initiated by entrainment and deposition mechanisms;
- stratified wavy to slug: this transition is initiated by wave growth until waves reach top wall of the tube and completely block the gas flow area;
- slug to annular: this transition was governed by two mechanisms; liquid slugs reaching the top wall of the tube and entrainment and deposition mechanisms;
- plug to slug: this transition is initiated by bubble coalescence.

Proposed Flow Regime Map

Based on the experimental data of the present work and on comparison with other flow regime maps, a flow regime map, based on (X, J_g^*) coordinates, is proposed for predicting the prevailing flow regimes in adiabatic horizontal flow in small diameter pipes ($d < 50$ mm). Since the transitions between the regimes were observed to occur over a certain range of flow conditions, the boundaries of the regimes were represented by transition bands.

Based on the results of Dukler and Taitel (1986), the range of applicability of the proposed flow regime map may be extended to boiling and condensing flows in horizontal tubes. Variations of the boundaries of the flow regimes as a result of mass transfer between the phases (cases of boiling and condensing flows) are expected to be within the proposed ranges for the transitions regions.

The proposed flow regime map including the proposed transition regions between the flow patterns is shown in Figure 5.21. The transition regions can be reproduced on (X, J_g^*) coordinates using the points given in Table 5.3.

7.1.2 Void Fraction

The homogeneous, Lockhart–Martinelli (1949), separate-cylinders models, and the CISE correlations were used to predict the average void fraction for all the test runs. Predictions of these models were compared with the average void fraction measured using the optical probes. Results of this comparison are:

- The homogeneous model significantly overpredicts the average void fractions.
- The Lockhart–Martinelli model overpredicts measurements for $\alpha < 0.4$, and provides reasonable predictions for $\alpha > 0.4$.
- The separate-cylinders model generally underpredicts measurements of the void fractions. Slightly better agreements were obtained for $\alpha > 0.7$ (range typical of annular flow regime).
- The CISE correlation provided the best predictions of the average void fractions (~70% of the measurements were predicted within $\pm 10\%$).

The measured void fractions were found to be regime-dependent, and the measurements for the different regimes were well correlated with the drift-flux model.

- For stratified wavy, slug, and annular flow regimes, the measured void fractions were correlated using the standard variables of the drift-flux model (J_g/α and J). The calculated drift-flux parameters are

Flow Regime	Distribution parameter C_o	Weighted average drift velocity V_{gj} (m/s)
Stratified wavy flow	1.04	0.466
Slug flow	1.06	0.991
Annular flow	0.92	3.688

- For plug flow, the measured void fractions were correlated based on the average void fraction within the two-phase flow segments α_p . The modified drift-flux variables are $(J_g/\alpha_p$ and $J)$, and the calculated drift-flux parameters for plug flow are $C_o = 0.98$, and $V_{gj} = 0.068$ m/s. The overall average void fraction can then be calculated using $\alpha = 1.014 \times \alpha_p^{1.87}$.
- The above drift-flux models were validated using the experimental data of França and Lahey (1992), and are expected to provide better predictions of the average void fractions than the four generalized models discussed earlier.

7.1.3 Friction Losses

The measured frictional pressure gradients were compared with predictions of the homogeneous model, Lockhart–Martinelli model, and Friedel’s correlation. For the homogeneous model, the predicted pressure gradients were calculated using five different formulations for the average viscosity of the mixture. The definitions proposed by McAdams et al. (1942) and Beattie and Whalley (1982) gave the best predictions of the measured frictional pressure losses. The main results of this comparison are:

- The homogeneous and Lockhart–Martinelli models provided reasonable agreement with the measured friction losses, while Friedel’s correlations significantly overpredicted the measurements.
- The reasonable agreement obtained between the Lockhart–Martinelli model and the experimental data is possibly due to the fact that the Lockhart–Martinelli model was

based on experimental data taken under conditions similar to those of the present study (adiabatic flow in horizontal tubes at near-standard atmospheric conditions).

- For the homogeneous model, some investigators, such as Whalley (1987) indicated that its success is fortuitous and may be due to compensating errors in the model. However, this success is substantiated by a number of experimental investigations, including the present one, and cannot be considered haphazard. The following are given as possible reasons for the reasonable predictions of the friction losses using the homogeneous model:
 - (i) the model is based on the well-established and well-tested formulations of single-phase flow; and
 - (ii) the definition of the mean viscosity of the mixture is used to adjust for the effects of any variations that may be caused by, among other factors, the flow patterns.
- Whalley (1986) and Collier and Thome (1994) indicated that Friedel's (1979) correlation provides the best estimates of the friction losses. However, the results of the present study show that Friedel's (1979) correlations significantly overpredicted the measured frictional pressure gradients, particularly in the case of plug and slug flow regimes. A similar conclusion was reached in the recent comparative study conducted by Tribbe and Müller-Steinhagen (2000). This different conclusion may be explained by the fact that Friedel's (1979) correlation was developed based mainly on experimental data taken in vertical test sections for boiling flows under high system pressures.

Flow Pattern-dependent models

The measured frictional pressure gradients were found to be flow pattern-dependent. Predictions of the homogeneous, Lockhart–Martinelli models, and Friedel's

correlations were compared to the measured friction losses for each of the observed flow regimes. The following results were obtained:

- For plug flow, the best predictions of the friction losses were obtained using the homogeneous model using the McAdams et al. (1942) or Beattie and Whalley (1982) definitions to calculate the mean viscosity of the mixture.
- For slug and annular flow regimes, the best predictions were obtained using the Lockhart–Martinelli correlation.
- For stratified wavy flow, both the homogeneous and the Lockhart–Martinelli models provided reasonable predictions of the friction losses.

The measured friction losses were also compared with the predictions of the regime-dependent correlations of Baker (1961), Beattie (1973), and Bandel (1973).

- Baker's (1961) correlations overpredicted the measurements of the annular and plug flow regimes and underpredicted the measurements of the slug and stratified wavy flow regimes. In general, the Lockhart–Martinelli model gave better predictions of the friction losses. This may be due to the fact that Baker's (1961) correlations were developed based mainly on experimental data taken in large diameter pipes.
- The classifications of the flow patterns used by Beattie (1973) are not consistent with those accepted in the literature, including the present study. Therefore, the predictions of all the correlations of Beattie (1973) were compared with the measurements for all the flow regimes. Only the annular and the wavy gas–liquid interfaces correlations showed reasonable agreement with the present data.
- The criteria used by Bandel (1973) to predict the occurrence of the annular flow regime agreed very well with the flow regime identification of the present study. However, the Bandel's (1973) correlations for the frictional pressure gradients

generally underpredicted the measurements. Better predictions were obtained for annular and stratified wavy flow test runs than for the slug and plug flow regimes. Bandel's (1973) correlations significantly underpredicted the measurements of the plug and slug flow regimes.

Two-Fluid Models for Friction Losses

- For stratified wavy and slug flow regimes, the two-fluid model underpredicts the measurements. This was attributed to the effects of the relative velocity between the phases at the interfaces on the velocity profiles in the phases. These effects invalidate the use of the conventional single-phase flow equations for the wall friction factors, which were developed for fully turbulent full pipe flow.
- For plug flow, the phases move at the same velocity. Therefore, the effect of the conditions of the gas–liquid interfaces on the velocity profiles in the phases is small and the conventional single-phase flow equation can be used to predict the wall friction factors. Results of the simple model developed for plug flow regime agreed very well with the measurements.
- For annular flow, the two-fluid model used assumed that all the liquid flows in the film. This assumption resulted in higher velocity of the liquid phase and, thus, higher frictional pressure gradients. Note that the acquired videos of the flow showed that a significant fraction of the liquid was entrained in the gas.
- Expressions were developed for the gas–liquid interfacial friction factors using the measured pressure gradients and a waviness coefficient to correct for the effects of the conditions of the interfaces on the velocity profiles in the phases, and for the increase in the interfacial area of the phases. The proposed expressions were compared with existing correlations and correlations are proposed for certain regimes.

7.1.4 Interfacial Area Concentration

Preliminary analysis of the interfacial area concentration for the stratified wavy flow test runs showed that the recorded videos of the horizontal- and vertical-planes of

the flow can be used to obtain accurate measurements of the concentration of the gas–liquid interfacial area and the local void fractions.

7.2 Contributions of the Present Work

The contributions of this experimental investigation include the following:

- (i) Identification and understanding of mechanisms that govern the transitions between the flow patterns.
- (ii) A two-phase flow regime map including allowance for transition regions, which can be used to identify the prevailing flow patterns in adiabatic flow in small diameter horizontal pipes at near atmospheric conditions. Based on information from the literature (see Section 5.6.3), the applicability of the proposed flow regime map can be extended to boiling and condensing flows in small diameter horizontal tubes.
- (iii) Review and assessment of the adequacy of models used to predict the frictional pressure gradients and average cross-sectional void fractions and recommendations for general and regime-specific predictions.
- (iv) Validated drift-flux models to predict the average void fractions in stratified wavy, slug, plug, and annular flow regimes.
- (v) Conventional expressions of the wall friction factors for fully developed full pipe flow are inappropriate for two-fluid models because of the effects of the relative velocity between the phases at the interface on the velocity profiles in the phases for all the observed flow regimes except for plug flow.
- (vi) A simple two-fluid model for predicting the frictional pressure gradients in plug flow regime.
- (vii) Experimental data that could be used to provide accurate measurements of the concentration of the gas–liquid interfaces and other important parameters of the

slug and plug flow regimes, such as shapes of the air bubbles and frequency of the liquid slugs.

7.3 Suggestions for Future Research

During the course of this investigation, a number of aspects, which require further work, have been identified. These aspects are discussed in this section.

(i) *Limits of the transition regions*

In most two-phase flow experiments, including the present study, the flow patterns were established by setting a certain liquid mass flow rate and consistently increasing or decreasing the air mass flow rate, while observing and recording changes in the disposition of the phases (or flow patterns).

Barajas and Panton (1993) studied the effects of contact angle on two-phase flow in capillary tubes. They noted that, for a certain mass flow rate of the liquid, the boundary of the flow pattern shifts depending on whether the flow regime was established by increasing or decreasing the air mass flow rate. Barajas and Panton (1993) attributed this shift to experimental hysteresis, i.e., hysteresis associated with the instruments. In reality, the observed shifts in the boundary of the flow regime may be due to physical hysteresis associated with the mechanisms that govern the transition between the regimes. The forces governing the transition from flow pattern A to B may not be the same as those governing the transition from B to A. As an example, video observations from the present study showed that the transition between stratified wavy and annular flow regimes is governed by entrainment and deposition mechanisms. This transition occurs when the gas turbulent energy is sufficient to break the liquid waves, and lift and suspend liquid droplets. While transition from annular to stratified wavy flow occurs when the gas turbulent forces are no longer able to suspend the liquid droplets in the gas phase, i.e., the gravity forces overcome the turbulence forces and the suspended droplets drop back on the liquid film. This seems to suggest that the transition

from stratified wavy to annular flow occurs at a higher gas superficial velocity than the transition from annular to stratified wavy, hence the proposed transition regions.

To investigate this hypothesis, data points in the transition regions should be investigated to determine if the prevailing flow pattern was established by increasing or decreasing the gas or liquid mass flow rate. The results of this investigation may establish a trend that can be used to test this hypothesis. It should be noted that it is possible that the number of data points in the transition regions may not be sufficient to establish a definitive trend to test this hypothesis. In this case, it is recommended that well documented experimental data from the literature, which include information on whether the flow patterns were established by increasing or decreasing the flow rate of each of the phases, should be identified and used to test this hypothesis.

(ii) *Flow patterns identification and classification*

The literature review performed in this study showed that there are different classifications of the flow patterns, and different methods have been used to predict the prevailing flow regime. The lack of a standard classification of the flow regimes, and a common methodology to predict these regimes makes it difficult to develop, validate or assess the flow patterns specific models and correlations.

Therefore, it is recommended to post the flow videos acquired in the present study on a web site and to invite the two-phase flow community to view these videos via the Internet, indicate and discuss their identification of the regimes using a clear set of pattern-specific identifying features. The results of these efforts may help achieve not only an objective but also a standard methodology that can be used to identify the prevailing flow regimes.

(iii) *Applicability of the proposed flow regime map to boiling and condensing flows*

Based on information from the literature (see Section 5.6.3), the applicability of the flow regime map proposed in this study can be extended to boiling and condensing flows in small diameter horizontal tubes. However, some investigators (e.g., Hewitt,

1978 and Nitheanandan, 1993) argued that, since the mechanisms that govern the transitions may be different in boiling and condensing flows than in adiabatic flow, flow regime maps developed based on adiabatic flow data should not be used for boiling and condensing flows. Therefore, it is advisable to test the validity of the proposed flow regime map using boiling and condensing flow data.

(iv) *Wall shear stresses*

This study showed that the conventional single-phase flow expressions for the wall friction factors cannot be used in two-phase flow systems because of the effects of the conditions of the gas–liquid interfaces on the velocity profiles in the phases. Measurements of the interfacial shear stress, which can be used to obtain correlations for the wall frictions factors, are scarce. Therefore, it is recommended that measurements of both the liquid- and gas-wall shear stresses be taken for different flow regimes. The data can then be used to develop more reliable two-fluid models for the calculations of the frictional pressure gradients. Accurate prediction of the fraction of the liquid phase entrained in the gas is also important for the annular flow regime.

(v) *Concentration of the Interfacial Area*

As shown for the stratified wavy flow regime, the recorded videotapes of the horizontal- and vertical-planes of the flow can be used to provide good measurements of the concentration of the gas–liquid interfaces (a_i). It is recommended that the recorded videotapes be analyzed to obtain expressions for a_i for all of the observed flow regimes. Results of this analysis are of prime importance for the development of detailed two-fluid models.

(vi) *Other Two-phase Flow Parameters*

The recorded signals of the optical probes should be analyzed to obtain information on other two-phase flow parameters, such as the length and frequency of the liquid slugs and the elongated gas bubbles.

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APPENDICES

Appendix A

Models and Derivation of Equations

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Models and Derivation of Equations

This appendix presents some of the models discussed in Chapter 3, and shows derivations of some of the equations used to plot the flow regime maps discussed in Chapters 2 and 5.

A.1 Friedel's Correlation for Friction Losses

A.1.1 Friedel's Correlation for Friction Losses

Friedel (1979) developed the following expression for the two-phase flow friction multiplier based on a large database of two-phase pressure drop measurements:

$$\phi_{lo}^2 = E + \frac{3.24 F H}{Fr^{0.045} We^{0.035}} \quad (A.1a)$$

where the variables E , F , H , Fr and We are defined using the following equations:

$$E = (1 - \chi)^2 + \chi^2 \frac{\rho_l}{\rho_g} \frac{C_{go}}{C_{lo}} \quad (A.1b)$$

$$F = \chi^{0.78} (1 - \chi)^{0.224} \quad (A.1c)$$

$$H = \left(\frac{\rho_l}{\rho_g} \right)^{0.91} \left(\frac{\mu_g}{\mu_l} \right)^{0.19} \left(1 - \frac{\mu_g}{\mu_l} \right)^{0.7} \quad (A.1d)$$

where Fr and We are the Froude and Weber numbers defined as follows:

$$Fr = \frac{G^2 d}{g d \rho_h} \quad (A.1e)$$

$$We = \frac{G^2 d}{\sigma \rho_h} \quad (A.1f)$$

where C_{go} and C_{lo} are the friction factors calculated based on the total mass velocity of the mixture flowing with the gas and liquid properties, respectively.

Friedel's (1979) correlation is applicable for vertical up-flow as well as horizontal flow. A separate correlation was provided for vertical downward flows.

A.1.2 CISE Correlation for Average Void Fraction

Premoli et al. (1970) suggested a correlation for the slip ratio S and thus the average cross-sectional void fraction. This correlation, known as the CISE correlation, is given in Whalley (1980). The slip ratio is given by

$$S = 1 + E_1 \left(\frac{y}{1 + y E_2} - y E_2 \right)^{0.5} \quad (\text{A.2a})$$

where y , E_1 and E_2 are defined as follows:

$$y = \frac{\beta}{1 - \beta} \quad (\text{A.2b})$$

$$\beta = \frac{\rho_l \chi}{\rho_l \chi + \rho_g (1 - \chi)} \quad (\text{A.2c})$$

$$E_1 = 1.578 Re^{-0.19} \left(\frac{\rho_l}{\rho_g} \right)^{0.22} \quad (\text{A.2d})$$

$$E_2 = 0.0273 We Re^{-0.51} \left(\frac{\rho_l}{\rho_g} \right)^{-0.088} \quad (\text{A.2e})$$

$$Re = \frac{G d}{\mu} \quad (\text{A.2f})$$

$$We = \frac{G^2 d}{\sigma \rho_l} \quad (\text{A.2g})$$

Re and We are Reynolds and Weber numbers for the total mass velocity flowing with the liquid properties. Note that the definition of We (Eq. A.2g), is different from that used in the correlation suggested by Friedel (1979) for the friction losses (Eq. A.1f).

A.1.3 Correlations of Bandel (1973)

As discussed in Section 3.4.2(b), Bandel (1973) developed empirical correlations for friction losses in horizontal two-phase flow systems. Bandel's (1973) correlations do not require prior knowledge of the prevailing flow regime. The flow patterns were divided into 3 categories; (i) annular flow, (ii) stratified smooth, and (iii) transitional flow. The transitional flow category includes all flow patterns other than annular and stratified smooth flow regimes.

The empirical method of Bandel (1973) calculates the frictional pressure gradient as follows:

- (1) Calculate the lower limiting pressure gradient for annular flow $\left(\frac{dP}{dx}\right)_{A,min}$, using Eq. A.3a.
- (2) Calculate the upper limiting pressure gradient for stratified flow $\left(\frac{dP}{dx}\right)_{St,max}$, using Eq. A.3b.
- (3) Calculate the pressure gradient using the annular flow correlation, Eqs. A.8a–d.
- (4) If the calculated pressure gradient in step (3) is *higher* than $\left(\frac{dP}{dx}\right)_{A,min}$, then the flow is annular and the pressure gradient is equal to that calculated in step (3), otherwise proceed to the following step.
- (5) Calculate the pressure gradient using the stratified smooth flow correlation, Eqs. A.9a–e.
- (6) If the calculated pressure gradient in step (5) is *less* than $\left(\frac{dP}{dx}\right)_{St,max}$, then the flow is stratified smooth and the pressure gradient is equal to that calculated in step (5), otherwise the flow is transitional.
- (7) For transitional flow, the pressure gradient is calculated by interpolating between the stratified smooth and the annular regimes (Eqs. A.10a–d).

The lower limiting pressure gradient for annular flow and the upper limiting pressure gradient for stratified smooth flow are calculated using Eqs. A.3a and A.3b, respectively.

$$\left(\frac{dP}{dx}\right)_{A,min} = \frac{g \rho_l}{10 \chi^{0.3}} \quad (\text{A.3a})$$

$$\left(\frac{dP}{dx}\right)_{St,max} = \frac{g \rho_l \chi}{15 \left(\frac{\mu}{\mu_g}\right)^{0.2}} \quad (\text{A.3b})$$

Note that the limiting pressure gradients for an air–water system, at standard atmospheric temperature and pressure, are plotted as a function of mass quality of the mixture (χ) in Fig.3.8.

(i) Single-phase flow parameters

The single-phase pressure gradients are calculated using the Blasius equation. Therefore, the gas and liquid Reynolds numbers are calculated as

$$Re_g = \frac{G \chi d}{\mu_g} \quad (\text{A.4a})$$

$$Re_l = \frac{G (1 - \chi) d}{\mu} \quad (\text{A.4b})$$

The gas– and liquid–wall friction factors are calculated as

$$f_g = \frac{0.3164}{Re_g^{0.25}} \quad (\text{A.5a})$$

$$f_l = \frac{0.3164}{Re_l^{0.25}} \quad (\text{A.5b})$$

The gas and liquid single-phase pressure drop are calculated as

$$\left(\frac{dP}{dx}\right)_{g,fric} = \frac{f_g G^2 \chi^2}{2 \rho_g d} \quad (\text{A.6a})$$

$$\left(\frac{dP}{dx}\right)_{l,fric} = \frac{f_l G^2 (1 - \chi)^2}{2 \rho_l d} \quad (\text{A.6b})$$

$$F_L = -0.31 \chi^{0.1} \quad (\text{A.7a})$$

$$\text{for } \chi \leq 0.5 \quad F_G = 0.15 \left(\frac{1-\chi}{\chi} \right)^{0.5} \left(\frac{\mu_l}{\mu_g} \right)^{0.3} \quad (\text{A.7b})$$

$$\text{for } \chi > 0.5 \quad F_G = 0.16 (1-\chi)^{0.1} \left(\frac{\mu_l}{\mu_g} \right)^{0.3} \quad (\text{A.7c})$$

Note that the single-phase pressure gradients are calculated assuming the phases are turbulent, since initially the flow is assumed to be annular.

(ii) Annular Flow

The pressure gradient for annular flow was calculated as follows:

$$F_{l,A} = \frac{(dP/dx)_{g,fric}}{(dP/dx)_{l,fric}} \frac{0.3164 + F_L}{0.3164 + F_G} \quad (\text{A.8a})$$

Then, iteration sequences are used to solve for the diameters and areas of the phases in annular flow. Increment $d_{g,A}$ and calculate $F_{2,A}$ using

$$F_{2,A} = \left(\frac{d_{g,A}}{d_{l,A}} \right)^{1.25} \left(\frac{A_{g,A}}{A_{l,A}} \right)^{1.75} \quad (\text{A.8b})$$

$$\text{where } d_{l,A} = d - d_{g,A}, \quad A_{g,A} = \frac{\pi d_{g,A}^2}{4}, \quad A_{l,A} = A - A_{g,A}.$$

For each value of $d_{g,A}$, $F_{l,A}$ and $F_{2,A}$ are compared and iteration was stopped when

$$\left| \frac{F_{2,A} - F_{l,A}}{F_{l,A}} \right| \leq \frac{1}{100}. \quad \text{Then the pressure gradients for annular flow are calculated as follows:}$$

$$\phi_{g,A}^2 = \left(1 + \frac{F_G}{0.3164} \right) \left(\frac{d}{d_{g,A}} \right)^{1.25} \left(\frac{A}{A_{g,A}} \right)^{1.75} \quad (\text{A.8c})$$

$$\left(\frac{dP}{dx} \right)_{TP,A} = \left(\frac{dP}{dx} \right)_{g,fric} \phi_{g,A}^2 \quad (\text{A.8d})$$

Now, if $\left(\frac{dP}{dx}\right)_{TP,A} \geq \left(\frac{dP}{dx}\right)_{A,min}$ then $\left(\frac{dP}{dx}\right)_{TP,fric} = \left(\frac{dP}{dx}\right)_{TP,A}$, otherwise

proceed with the stratified flow calculations.

(iii) Stratified Smooth Flow

In stratified smooth flow, the liquid phase may be under laminar flow conditions, therefore, if $Re_l < 2400$;

$$\left(\frac{dP}{dx}\right)_{l,fric} = \frac{64}{Re_l} \frac{G^2 (1-\chi)^2}{2 \rho_l d} \quad (A.9a)$$

$$F_{1,St} = \frac{(dP/dx)_{g,fric}}{(dP/dx)_{l,fric}} \frac{0.3164 + F_L}{0.3164 + F_G} \quad (A.9b)$$

Again here, iteration sequences are used to solve for equivalent diameters and areas of the phases for stratified flow. Increment the angle formed by the gas-liquid interface and the centre of the pipe ϕ .

$$A_{l,St} = \frac{d^2(\phi - \sin\phi)}{8}$$

$$A_{g,St} = A - A_{l,St}$$

$$P_{l,St} = \frac{d\phi}{2} + d \sin \frac{\phi}{2}$$

$$P_{g,St} = d(\pi - \phi + \sin \frac{\phi}{2})$$

$$d_{g,St} = \frac{4A_{g,St}}{P_{g,St}}$$

$$d_{l,St} = \frac{4A_{l,St}}{P_{l,St}}$$

$$F_{2,St} = \left(\frac{d_{g,St}}{d_{l,St}}\right)^{1.25} \left(\frac{A_{g,St}}{A_{l,St}}\right)^{1.75} \quad (A.9c)$$

For each value of ϕ , $F_{1,St}$ and $F_{2,St}$ are compared and iteration was stopped when

$$\left| \frac{F_{2,St} - F_{1,St}}{F_{1,St}} \right| \leq \frac{1}{100}.$$

Then the pressure gradients for stratified smooth flow are calculated as follows:

$$\phi_{g,St}^2 = \left(1 + \frac{F_G}{0.3164}\right) \left(\frac{d}{d_{g,St}}\right)^{1.25} \left(\frac{A}{A_{g,St}}\right)^{1.75} \quad (\text{A.9d})$$

$$\left(\frac{dP}{dx}\right)_{TP,St} = \left(\frac{dP}{dx}\right)_{g,fric} \phi_{g,St}^2 \quad (\text{A.9e})$$

Now, if $\left(\frac{dP}{dx}\right)_{TP,St} \leq \left(\frac{dP}{dx}\right)_{St,max}$ then $\left(\frac{dP}{dx}\right)_{TP,fric} = \left(\frac{dP}{dx}\right)_{TP,St}$, otherwise

the flow is transitional.

(iv) Transitional Flow

The frictional pressure gradient is calculated using linear interpolation of critical stratified smooth and annular flow pressure gradients, as follows:

$$F_T = \frac{0.3164 \chi^{1.75} \mu_g^{0.25}}{2 \rho_l d^{1.25}} \quad (\text{A.10a})$$

$$\dot{m}_{A,min} = \left(\frac{(dP/dx)_{A,min}}{\phi_{g,A}^2 F_T} \right)^{1/1.75} \quad (\text{A.10b})$$

$$\dot{m}_{St,max} = \left(\frac{(dP/dx)_{St,max}}{\phi_{g,St}^2 F_T} \right)^{1/1.75} \quad (\text{A.10c})$$

$$\left(\frac{dP}{dx}\right)_{TP,fric} = \left(\frac{dP}{dx}\right)_{St,max} \exp \left(\frac{\ln \left(\frac{\dot{m}}{\dot{m}_{St,max}} \right) \ln \left(\frac{(dP/dx)_{A,min}}{(dP/dx)_{St,max}} \right)}{\ln \left(\frac{\dot{m}_{A,min}}{\dot{m}_{St,max}} \right)} \right) \quad (\text{A.10d})$$

A.2 Derivation of Equations

The flow regime maps of Taitel and Dukler (1976) and Weisman et (1979) were presented using different mapping parameters, which are also different from those used to present the flow regime map of the present study. This section presents some of the derivation of equations used to plot the proposed transition curves in terms of the dimensionless superficial velocity of the gas (J_g^*) and the Lockhart-Martinelli (1948) parameter (X).

A.2.1 Taitel and Dukler (1976a)

The mapping parameters used by Taitel and Dukler (1976a) are presented in Section 2.4.4, and their map is discussed in Section 5.4.2. The flow regime map of Taitel and Dukler (1976a) is presented as Fig. 2.9. The equations used to plot Curve A and Curve B are discussed in the following sections.

(i) Transition from Stratified Wavy to Annular or Intermittent (Curve A)

Taitel and Dukler (1976a) indicated that the transition from stratified wavy to annular or intermittent flow regimes (Curve A in Figure 2.9) occurs when

$$F^2 \frac{1}{C_2^2} \frac{\tilde{u}_g}{\tilde{A}_g} \left(\frac{d\tilde{A}_L}{d\tilde{h}} \right) \geq 1 \quad (\text{A.11})$$

where F is the modified Froude number defined in Eq. 2.5, and $C_2 = 1 - \tilde{h}_L$.

Other parameters in Eq. A.11 are defined as follow

$$\tilde{u}_g = \frac{u_g}{J_g} \quad \tilde{A}_g = \frac{A_g}{d^2} \quad \tilde{A}_l = \frac{A_l}{d^2} \quad \tilde{h}_L = \frac{h_L}{d}$$

The above terms can all be defined as a function of $\tilde{h}_L$. Note that the symbol ($\sim$) is used to indicate that the parameters are dimensionless.

Taitel and Dukler (1976a) used the momentum balance equations of the gas and liquid phases for a simplified configuration of the stratified smooth flow regime to obtain an expression of X as a function of $\tilde{h}_L$. Then, substituting this expression in Eq. A.11 to

obtain an equation of F as a function of X , which was used to plot Curve A in Fig. 5.13 (for a horizontal 25.4-mm diameter pipe and using gas and liquid properties at standard atmospheric conditions, i.e., 20°C and 101.3 kPa pressure).

Note that Curve A was plotted by setting the *L.H.S.* of Eq. A.11 equal to 1.

(ii) Transition from Dispersed Bubbly or Intermittent to Annular (Curve B)

Taitel and Dukler (1976a) considered that the transition from dispersed bubbly or intermittent flow regimes to annular flow occurs at $\tilde{h}_L$ values in the range of 0.35–0.5, which corresponds to values of X in the range of 0.7–1.6.

A.2.2 Weisman et al. (1979)

The correlations and the flow regime map proposed by Weisman et al. (1979) were discussed in Sections 2.4.5 and 5.4.3. Weisman et al. (1979) adopted the flow patterns classification of Mandhane et al. (1974) and studied the effects of fluid properties and pipe diameter on the boundaries of the flow regimes in a 6.1-m long horizontal test section. The pipe diameters considered were 12 mm, 25 mm and 54 mm.

The flow regime map of Weisman et al. (1979) is presented as Fig. 2.5. The equations used to plot this flow regime map in Fig. 5.14 are presented in the following sections.

(i) Stratified to intermittent transition

Weisman et al. (1979) proposed the following expression for the transition between the stratified and intermittent flow regimes:

$$\frac{J_g}{(g d)^{1/2}} = 0.25 \left(\frac{J_g}{J_l} \right)^{1.1} \quad (\text{A.12a})$$

The following describes the method used to define this transition curve in terms of J_g^* and X .

Assuming turbulent flow, the Lockhart-Martinelli (1948) parameter X can be expressed as a function of J_g and J_l .

$$X = \left(\frac{\rho_l}{\rho_g} \right)^{0.4} \left(\frac{J_l}{J_g} \right)^{0.9} \left(\frac{\mu}{\mu_g} \right)^{0.1} \quad (\text{A.12b})$$

$$\left(\frac{J_g}{J_l} \right) = X^{-1/0.9} \left(\frac{\rho_l}{\rho_g} \right)^{0.4/0.9} \left(\frac{\mu}{\mu_g} \right)^{0.1/0.9} \quad (\text{A.12c})$$

Substituting the above equation in Eq. A.12a gives

$$J_g = 0.25 (g d)^{1/2} X^{-1/0.9} \left(\frac{\rho_l}{\rho_g} \right)^{0.44/0.9} \left(\frac{\mu}{\mu_g} \right)^{0.11/0.9} \quad (\text{A.12d})$$

Using the gas dimensionless velocity (J_g^*) defined in Eq. 2.13 (Wallis's, 1969)

$$J_g^* = 0.25 X^{-1/0.9} \left(\frac{\rho_g}{\rho_l - \rho_g} \right)^{1/2} \left(\frac{\rho_l}{\rho_g} \right)^{0.44/0.9} \left(\frac{\mu}{\mu_g} \right)^{0.11/0.9} \quad (\text{A.12e})$$

The above equation was used to plot the transition curve separating the stratified and the intermittent flow regimes in Fig. 5.14.

(ii) Transition to annular flow

$$1.9 \left(\frac{J_g}{J_l} \right)^{1/8} = \left[\frac{J_g \rho_g^{1/2}}{(g (\rho_l - \rho_g) \sigma)^{1/4}} \right]^{0.2} \left(\frac{J_g^2}{g d} \right)^{0.18} \quad (\text{A.13a})$$

Re-arranging the above equation and substituting for $\left(\frac{J_g}{J_l} \right)$ as a function of X , gives the following equation:

$$J_k^* = \left[\frac{\rho_k}{g d (\rho - \rho_k)} \right]^{1/2} \left\{ 1.9 \left[X^{-1/0.9} \left(\frac{\rho_k}{\rho} \right)^{0.4/0.9} \left(\frac{\mu_k}{\mu} \right)^{0.4/0.9} \right]^{1/8} \left[\frac{\rho_k^{1/2}}{(g (\rho - \rho_k) \sigma)^{1/4}} \right]^{-0.2} \left(\frac{1}{g d} \right)^{-0.18} \right\}^{1/0.56} \quad (\text{A.13b})$$

(iii) Stratified smooth to stratified wavy transition

$$\left[\frac{\sigma}{(\rho - \rho_k) g d^2} \right]^{0.2} \left(\frac{d J_g \rho_g}{\mu_g} \right)^{0.45} = 8 \left(\frac{J_g}{J_l} \right)^{0.16} \quad (\text{A.14a})$$

Again here, re-arranging the Eq. A.14a, and substituting for $\left(\frac{J_g}{J_l} \right)$ Eq., gives the following expression for J_g^* as a function of X :

$$J_k^* = \left[\frac{\rho_k}{g d (\rho - \rho_k)} \right]^{1/2} \left\{ 8 \left[\frac{\sigma}{(\rho - \rho_k) g d^2} \right]^{-0.2} \left[\frac{d \rho_k}{\mu_g} \right]^{-0.45} \left[X^{-1/0.9} \left(\frac{\rho_k}{\rho} \right)^{-0.4/0.9} \left(\frac{\mu_k}{\mu} \right)^{-0.4/0.9} \right]^{0.16} \right\}^{1/0.45} \quad (\text{A.14b})$$

◆ *Summary*

The empirical correlations proposed by Weisman et al. (1979) for the transitions between the flow patterns were recalculated using Eqs. A.12e, A.13b, and A.14b for air–water flow in a 25.4-mm diameter horizontal tube at standard atmospheric conditions (20°C and 101.3 kPa pressure). The calculated curves are compared with the results of the present study in Fig. 5.14.

Appendix B

Instrument Calibration and Experimental Uncertainties

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Instrument Calibration and Experimental Uncertainties

This appendix describes the methods used to calibrate the instruments and to estimate the uncertainties associated with their measurements.

B.1 Calibration Method and Results

All instruments were calibrated prior to each set of experimental test runs (typically 10–20 test runs). Calibrations of the instruments were also checked following each set of experiments to confirm that the instruments operated properly for all the test runs. Typical calibration results of the instruments are presented in the following subsections.

B.1.1 Pressure Transducers

The following pressure transducers were used in the present experimental study:

- (i) Celesco VR-series differential pressure transducer with a 0–690 kPag pressure range. This transducer was used to measure the wall pressure of the air supply line, near the air velocity transducer. The transducer was calibrated using a dead-weight tester, the output voltage of the transducer was recorded as weights were added or removed. The calibration data are shown in Fig. B.1*a*. The slope of the least-squares fitted line is 115.39 kPa/V. The coefficient of determination (R^2) = 1, and the standard deviation is 0.720 kPa.

The standard deviation (S) for all the instruments was calculated as

$$S = \sqrt{\frac{\sum_{i=1}^N (Y_i - Y_{ic})^2}{n - 1}} \quad (\text{B.1})$$

where Y_i is the measured quantity, Y_{ic} is the value calculated using the best-fit line (calibration curve), and n is the number of readings. In Eq. B.1, $(n-1)$ was used to obtain unbiased estimates of S . This is important when n is small (<30).

- (ii) Omega PX136-series (0–7 kPag), used to measure the static pressure near the end of the test section. This transducer was calibrated using a vertical water manometer. The calibration data are shown in Fig. B.1*b*. The slope of the least-squares fitted line is 1.467 kPa/V. The coefficient of determination (R^2) = 1, and the standard deviation is 0.0055 kPa.
- (iii) Omega PX26-series (0–35 kPag), used to measure the wall pressure midway through the test section, near where local void measurements were taken. In the lower 30% of the pressure range, the transducer was calibrated using a vertical water manometer (Fig. B.1*c*). At higher pressures, the transducer was calibrated using another calibrated transducer. Since higher uncertainties are usually observed in the lower end of the pressure range of the transducer, calculations of the uncertainties are based on the calibration data of this range. However, calibration of the transducer was checked over its full pressure range. Fig. B.1*c* shows that the slope of the least-squares fitted line is 6.980 kPa/V. The coefficient of determination (R^2) is ~ 1 , and the standard deviation is 0.066 kPa.
- (iv) Omega PX136-series (0–100 kPag), used to measure the wall pressure at the inlet of the test section. In the lower 30% of its pressure range, this transducer was calibrated using Omega (0–35 kPag) pressure transducer. The calibration data are shown in Fig. B.1*d*. The slope of the least-squares fitted line is 20.565 kPa/V. The coefficient of determination (R^2) is ~ 1 , and the standard deviation is 0.087 kPa.
- (v) Baratron pressure transducer was used in the calibration of the air velocity transducer. This transducer was calibrated using a vertical water manometer. The calibration data are shown in Fig. B.1*e*. The slope of the least-squares fitted line is 1.331 kPa/V. The coefficient of determination (R^2) is ~ 1 , and the standard deviation is 0.044 kPa.

Figure B.1: Calibrations results of the pressure transducers used in this study.

B.1.2 Thermocouples

Three type “T” copper-constantan thermocouples were to measure (i) air inlet temperature, (ii) water inlet temperature, and (iii) temperature of the two-phase mixture downstream of the test tube. The properties were based on the average of the three temperatures. The locations at which the temperature measurements were taken are shown in Fig. 4.1.

Linearization of the output of the thermocouples, and allowance for cold junction compensation (CJC) were performed using a subroutine called TEMPERATURE. The accuracy of the thermocouples was checked by comparison with a mercury thermometer. Readings were compared at three different temperatures; (i) ice bath; (ii) boiling water; and (iii) controlled ambient temperature of the laboratory. The recorded temperatures are shown in Fig. B.2. The solid lines represent the average readings of the mercury thermometer.

The average standard deviation of the three thermocouples calculated using Eq. B.1 for the ambient temperature measurements was $\sim 0.25^{\circ}\text{C}$.

The effects of measuring the temperature of moving fluid with a stationary thermocouple were assessed and found to be well within the estimated uncertainty in the measurements.

Figure B.2: Assessing the accuracy of the thermocouples.

B.1.3 Air Velocity Transducer

The air mass flow rate meter was calibrated *in situ* using a pitot tube (1.6 mm o.d. and 0.8 mm i.d.). The pitot tube was placed in the test section at an $(l/d) = 410$ (location where the optical probe was used to measure the local void fraction). The tube was properly aligned with the flow and was connected to a Baratron pressure transducer. The wall pressure was measured using the Omega PX136 pressure transducer. It should be noted that the measurements of the wall pressures were corrected for errors introduced because the total and wall pressure were not measured at the same l/d location.

The pitot tube was traversed across the flow using the traversing mechanism of the optical probe. For each air flow rate, the total pressure was measured at 9 locations along the pipe diameter. The dynamic pressure was calculated as the difference between the total and the corrected static pressure. The mass velocity of the air was then determined using the velocity profile and the calculated air density based on the measured static pressure.

Figure B.3 shows good correlation between the mass velocity measured using the AVT and that measured using the stagnation tube (standard deviation is $0.617 \text{ kg/m}^2\cdot\text{s}$.)

Figure B.3: Calibration results of the air velocity transducer.

B.1.4 Paddlewheel Sensor

An Omega paddlewheel sensor (FP5300) was used to measure the flow rate of the water. The sensor was placed in a 12.7-mm pipefitting with sufficient upstream and downstream straight-run piping to ensure that the flow is fully developed.

The paddlewheel sensor was calibrated *in situ*. Lab water was passed through the test section and collected in the 2-m³ reservoir (Fig. 4.1). The accumulation rate of the water in the reservoir was determined using the surface area of the tank and the rate of increase of the water level in the tank. Measurements were taken at 6 different flow rates (i.e., valve settings). The calibration data is shown in Fig. B.4. The slope of the least-squares fitted line is 0.118 l/s·V and the coefficient of determination (R^2) ~ 1 . The standard deviation is 0.004 l/s (calculated using Eq. B.1).

Figure B.4: Calibration results of the paddlewheel sensor.

B.1.5 Optical Probes

As discussed in Section 4.4.2, during the setup and testing of the instruments, it was found that the response time of the optical probes depended to a great extent on the preparation of the sensing tip of the fibre. Therefore, the probes were carefully prepared, examined, and their response time was assessed. A stroboscope was set to emit light pulses at 10 kHz while the output signals of the photodetectors were recorded on a digital oscilloscope. This test showed that the 100% fall and rise time of the photodetectors and related circuitry is $< 50 \mu\text{s}$.

Miller and Mitchie (1969) used a high-speed photography technique (5000 frames/s) to study the accuracy of a single fibre-optic probe. They concluded that the optical probe the measurements were within $\pm 10\%$. Better accuracy is expected in this study because of: (i) improved characteristics of the instruments used, i.e., the very short response time of the photodetectors; (ii) small diameter probes ($\sim 50 \mu\text{m}$); (iii) high data sampling rate (300 MHz digital oscilloscope); and (iv) average measurement of the two probes. Measurements of the void fraction in this study are expected to be within $\pm 5\%$.

B.1.6 Data Acquisition System

The performance of the analog-to-digital (AD) converters of the data acquisition system was tested using signals of known amplitude and frequency. A calibrated multifunction wave generator was set to produce a sine waveform of $5 \pm \text{V}$ at 75 Hz. The waveform was fed into each input channel of the data acquisition system. Figure B.6 shows a typical time trace of the waveform. The input signals were correctly measured with respect to phase and amplitude.

Figure B.6: Measured voltage of a sine waveform of $\pm 5 \text{ V}$ at 75 Hz.

B.1.7 Liquid Surface Tension Measurements

As discussed in Chapter 4.3.5, variations of water surface tension were found to effect the transition between the flow regimes (Taitel and Dukler, 1976*a* and Weisman et al., 1979). Therefore, it was considered beneficial to estimate the range over which the water surface tension is expected to vary in the present experimental study.

A number of water samples were taken at different locations throughout the test facility and under different conditions, which cover those under which the experiments were to be performed (see Table B.1). A pendant drop measurement system was used to measure the surface tension of these water samples. The system was set-up at the Dept. of Chemical Eng. at the University of Ottawa. The system was calibrated and used by a Ph.D. student to measure the surface tension of different liquid solutions.

The pendent drop technique consists of suspending a liquid drop at the end of a small diameter gage tube. The drop is videotaped while its size is increased very slowly. The liquid drop is attached to the tube by surface tension forces. At a certain size the liquid drop detaches from the needle. At this point, the surface tension force is equal to the weight of the drop. Therefore, the last frame recorded before the drop detaches is used to estimate the size of the drop and thus its weight. The surface tension force is equal to the weight divided by the perimeter of the gage tube.

As shown in Fig. B.7*a*, the surface tension of each water sample was measured three times. The average readings ranged from 69.6 to 72.5 mN/m, except for water sample no. 10. As indicated in Table B.1, a drop of soap (sodium dodecyl sulphate) was added to this sample to lower its surface tension. This was done to ensure that the measurement system responds to a change in surface tension.

Measurements of the surface tension of the samples were performed at ambient room temperature (about 18–20°C).

The surface tension of the liquid also changes with temperature. Figure B.7b shows the variation of the water surface tension with temperature. The data were taken from Findlay (1973). The expression of the least-squares fitted line shown in Fig. B.7b was used to calculate the surface tension in this study.

$$\sigma = -0.155 T_m + 75.774 \quad (\text{B.2})$$

where σ is the surface tension in (mN/m) and T_m is the average temperature of the mixture.

It should be noted that a number of water samples were also collected for different test runs. The samples were stored for possible later analysis if necessary.

Table B.1: Descriptions of the water samples tested used surface tension measurements.

Sample #	Water sample description
1	Distilled water sample.
2	Sample collected at the inlet of the test section after water was circulated through the test section for ~5 min.
3	Sample collected at the inlet of the test section after was circulated through the test section for ~20 min.
4	Sample collected from the water reservoir (from the top surface).
5	Water left in the reservoir for over 24 hrs, a water sample was then collected.
6	Water left in the reservoir for over 24 hrs. Circulated the water through the test section and collected a sample at the inlet of the mixer.
7	Water sample collected form the reservoir before starting the pump.
8	Water sample collected at inlet of the test tube, water one in reservoir and test rig running for over 15 min
9	Water was left in the reservoir for over 2 days, then a sample was collected from the reservoir (top surface).
10	Distilled water with a small quantity (1 drop) of "sodium dodecyl sulphate". This sample was considered to ensure that the system detects a known decrease in the surface tension of the liquid.

Figure B.7a: Surface tension of the water samples listed in Table B.1.

Figure B.7b: Variation of the surface tension of water with temperature.

B.1.8 Interfacial Area Concentration

The CCD cameras and the optical lenses were checked for optical distortions, which cause variations in the aspect ratio of the pixels near the edges of the photographs. Optical distortions can cause either barrel or pincushion effects. The optical distortion of the system was estimated using a photograph of a grid on a metal plate (Fig. B.8). This photograph was digitized and analyzed to estimate the distortion of the grid near the edges of the photographs. Horizontal and vertical lines (one pixel width) were drawn around the photograph as shown in Fig. B.8.

Figure B.8 shows a small distortion of the grid near the corners of the photograph (barrel effects). The largest distortion is less than half the width of the grid line (estimated at 0.5 mm). Therefore, the optical distortion of the system is less than 0.25 mm, which represents $\sim 1\%$ of the pipe diameter. The resolution of videotaped images is 5 pixels per mm (or 25 pixels per mm^2). The linear resolution of the images is $\sim \pm 0.2$ mm. Considering that the smallest identifiable size from an image is 1 pixel by 1 pixel, the resolution of the images is 0.4 mm^2 . However, the uncertainties associated with actual measurements of the interfacial area depend on other parameters, such as the clarity of the images and ability of the analyst to clearly identify the interfaces. The recorded videos show that, because plug and stratified wavy flow occur at relatively lower mass-velocities than slug and annular flow, more accurate measurements of the interfacial area can be obtained for the plug and stratified wavy flow regimes than for the slug and annular flow regimes.

In summary, the uncertainties in measurement of the concentration of the interfacial area of the phases depend on the flow regime on the method used to identify the interfaces.

Figure B.8: Photograph used to estimate the optical distortion of the interfacial area measurement system.

B.2 Estimates of Uncertainties

B.2.1 Measured Quantities

The uncertainty in a measured quantity Y_i was estimated from the calibration data of the instruments using

$$\delta Y_i = \pm 2 S_i \quad (\text{B.3})$$

where S_i is the standard deviation of the calibration data. If the calibration data can be approximated by a Gaussian distribution, Eq. B.3 estimates the uncertainty in the measured quantity Y_i at a 95 % confidence level. In other words, the true value of the measured quantity is expected to be within $\pm \delta Y_i$ interval of the measured value 95 times out of 100, or the odds are 20:1 against the uncertainty in a measurement Y_i being larger than $\pm \delta Y_i$.

The uncertainties in the measured quantities are estimated using Eq. B.3 and are summarized in Table B.2.

Table B.2: Estimates of uncertainties in the measured quantities.

Measuring Instrument	Standard deviation (S_i)	Estimated Uncertainty @ 95% confidence level
Celesco VR-series (0–690 kPag)	0.720 kPa	± 1.44 kPa
Omega PX136 (0–7 kPag)	0.0055 kPa	± 0.011 kPa
Omega PX26 (0–35 kPag)	0.066 kPa	± 0.132 kPa
Omega PX136 (0–100 kPag)	0.087 kPa	± 0.174 kPa
Baratron pressure transducer	0.044 kPa	± 0.088 kPa.
Air velocity transducer	0.617 kg/m ² ·s	± 1.23 kg/m ² ·s
Paddlewheel sensor	0.004 l/s	± 0.008 l/s
Type “T” thermocouples	(see Section B.1.2)	$\pm 0.5^\circ\text{C}$
Twin-needle optical probe	(see Section B.1.4)	$\sim 5\%$
CCD cameras (simultaneous vertical- and horizontal-planes of the test section.	(see Section B.1.8)	0.4 mm ² (1x1 pixels)*.

* uncertainty depends on the flow regime and the method used to identify the gas–liquid interfaces.

B.2.2 Calculated Quantities

The uncertainties of the calculated quantities, such as pressure gradients and superficial velocities, were estimated using the uncertainties of the measured quantities. If N is a known function of n independent variables ($x_1, x_2, \dots, x_n$), where x_1 to x_n are measured quantities with δx_1 to δx_n individual errors. The uncertainty in the calculated value of N is given by

$$\delta N = \sqrt{\sum_{i=1}^n \left(\delta x_i \frac{\partial N}{\partial x_i} \right)^2} \quad (\text{B.4})$$

where $\frac{\partial N}{\partial x_i}$ is the partial derivative of N with respect to x_i . If all uncertainties (δx_i) in Eq. B.4 are estimated at 95% confidence level, δN is also estimated at a 95% confidence level. The relative probable error is calculated as $\delta N/N$.

The uncertainties in the calculated values of the superficial velocities of the gas and liquid phases and the pressure gradients are estimated as outlined in the following sections.

(a) Liquid Superficial Velocity

The superficial velocity of the liquid (J_l) is calculated from the measured flow rate of the water (Q) and the cross-sectional area of the test section (A).

$$J_l = \frac{Q}{A} \quad (\text{B.5})$$

The area A is calculated as $A = \frac{\pi d^2}{4}$. Therefore, J_l can be expressed as a function of Q and d as follows:

$$J_l = \frac{4Q}{\pi d^2} \quad (\text{B.6})$$

The partial derivatives of J_l with respect to Q and d are

$$\left(\frac{\partial J_l}{\partial Q}\right) = \frac{4}{\pi d^2} \quad (\text{B.7})$$

$$\left(\frac{\partial J_l}{\partial d}\right) = \frac{-8 Q}{\pi d^3} \quad (\text{B.8})$$

The uncertainties in the calculated values of J_l are calculated as

$$\delta J_l = \sqrt{\left(\delta Q \frac{4}{\pi d^2}\right)^2 + \left(\delta d \frac{-8 Q}{\pi d^3}\right)^2} \quad (\text{B.9})$$

where δd is the uncertainty in the diameter of the test tube and δQ is the uncertainty in the measured liquid flow rate. Based on information provided by the manufacturer of the test tube, δd is estimated at ± 0.5 mm, and δQ is estimated in Section B.2.1 at ± 0.008 l/s (or $\delta Q = 0.008 \cdot 10^{-3} \text{ m}^3/\text{s}$). Using these estimates of δd and δQ , the uncertainties in the calculated values of J_l are calculated using Eq. B.9 and are given in Table B.3.

(b) Gas Superficial Velocity

The superficial velocity of the gas (J_g) is calculated from the measured mass velocity of the air (G), the density of the air (ρ_g).

$$J_g = \frac{G}{\rho_g} \quad (\text{B.10})$$

The air density was calculated as

$$\rho_g = \frac{P}{R T} \quad (\text{B.11})$$

where $P = P_{atm} + \left(\frac{P_1 + P_3}{2} \right)$ is the average pressure in the test section, P_{atm} is the standard atmospheric pressure (101.3 kPa), P_1 and P_3 are the static gauge pressures measured at the inlet and the end of the test section, respectively. R is the gas constant for air (287 J/kg·K) and T is the measured temperature of the air–water mixture. J_g is expressed as a function of G , R , T and P ;

$$J_g = \frac{G R T}{P} \quad (B.12)$$

The partial derivatives of J_g with respect to G , T and P are

$$\left(\frac{\partial J_g}{\partial G} \right) = \frac{R T}{P} \quad (B.13)$$

$$\left(\frac{\partial J_g}{\partial T} \right) = \frac{G R}{P} \quad (B.14)$$

$$\left(\frac{\partial J_g}{\partial P} \right) = -\frac{G R T}{P^2} \quad (B.15)$$

The uncertainties in the calculated values of J_g are calculated as

$$\delta J_g = \sqrt{\left(\delta G \frac{R T}{P} \right)^2 + \left(\delta T \frac{G R}{P} \right)^2 + \left(\delta P \frac{-G R T}{P^2} \right)^2} \quad (B.16)$$

The uncertainties in the measured quantities δG and δT are estimated in Table B.2 at $\pm 1.234 \text{ kg/s.m}^2$ and $\pm 0.5^\circ\text{C}$, respectively. The uncertainty in the calculated values of the average pressure in the test section is estimated as $\delta P = (\delta P_1^2 + \delta P_3^2)^{1/2}$, where δP_1 and δP_3 are the uncertainties in the static gauge pressures measured at the inlet and the end of the test section, respectively. Therefore, $\delta P = (174^2 + 11^2)^{1/2} = 174 \text{ Pa}$.

The calculated uncertainties δJ_g are given in Table B.4.

(c) Measured Pressure Gradient

The average measured pressure gradient is calculated as

$$\left(\frac{dP}{dx}\right) = \frac{l}{3} \left(\frac{P_1 - P_3}{L_1} + \frac{P_1 - P_2}{L_2} + \frac{P_2 - P_3}{L_3} \right) \quad (\text{B.17})$$

where P_1 , P_2 , and P_3 are the wall pressures measured at $l/d = 1, 410$, and 540 , respectively, and L_1 , L_2 , and L_3 are the distances separating the pressure ports. The uncertainties in the measured pressures P_1 , P_2 and P_3 are given in Table B.2 the uncertainty in measuring L_1 , L_2 , and L_3 is estimated at ± 5 mm.

The overall uncertainties in $\left(\frac{dP}{dx}\right)$ are given by

$$\delta\left(\frac{dP}{dx}\right) = \left[\left(\delta P_1 \left(\partial \left(\frac{dP}{dx} \right) / \partial P_1 \right) \right)^2 + \left(\delta P_2 \left(\partial \left(\frac{dP}{dx} \right) / \partial P_2 \right) \right)^2 + \left(\delta P_3 \left(\partial \left(\frac{dP}{dx} \right) / \partial P_3 \right) \right)^2 + \left(\delta L_1 \left(\partial \left(\frac{dP}{dx} \right) / \partial L_1 \right) \right)^2 + \left(\delta L_2 \left(\partial \left(\frac{dP}{dx} \right) / \partial L_2 \right) \right)^2 + \left(\delta L_3 \left(\partial \left(\frac{dP}{dx} \right) / \partial L_3 \right) \right)^2 \right]^{1/2} \quad (\text{B.18})$$

The partial derivatives of $\left(\frac{dP}{dx}\right)$ with respect to the measured pressures and distances separating the pressure ports are

$$\left(\partial \left(\frac{dP}{dx} \right) / \partial P_1 \right) = \frac{l}{3} \left(\frac{1}{L_1} + \frac{1}{L_2} \right) \quad (\text{B.19})$$

$$\left(\partial \left(\frac{dP}{dx} \right) / \partial P_2 \right) = \frac{l}{3} \left(\frac{-1}{L_2} + \frac{1}{L_3} \right) \quad (\text{B.20})$$

$$\left(\partial \left(\frac{dP}{dx} \right) / \partial P_3 \right) = \frac{l}{3} \left(\frac{-1}{L_1} + \frac{-1}{L_3} \right) \quad (\text{B.21})$$

$$\left(\partial \left(\frac{dP}{dx} \right) / \partial L_1 \right) = \frac{l}{3} \left(-\frac{P_1 - P_3}{L_1^2} \right) \quad (\text{B.22})$$

$$\left(\partial \left(\frac{dP}{dx} \right) / \partial L_2 \right) = \frac{1}{3} \left(-\frac{P_1 - P_2}{L_2^2} \right) \quad (\text{B.23})$$

$$\left(\partial \left(\frac{dP}{dx} \right) / \partial L_3 \right) = \frac{1}{3} \left(-\frac{P_2 - P_3}{L_3^2} \right) \quad (\text{B.24})$$

As discussed in Section 6.3.1, for 8 test runs (45, 46, 47, 48, 73, 81, 94 and 107) the pressure measured using the pressure transducer at the end of the test section were discarded. The overall pressure gradients for these test runs were taken as

$$\left(\frac{dP}{dx} \right) = \left(\frac{P_1 - P_2}{L_2} \right) \quad (\text{B.25})$$

Therefore, the uncertainties in $\left(\frac{dP}{dx} \right)$ for these test runs were calculated as

$$\delta \left(\frac{dP}{dx} \right) = \left[\left(\delta P_1 \left(\partial \left(\frac{dP}{dx} \right) / \partial P_1 \right) \right)^2 + \left(\delta P_2 \left(\partial \left(\frac{dP}{dx} \right) / \partial P_2 \right) \right)^2 + \left(\delta L_2 \left(\partial \left(\frac{dP}{dx} \right) / \partial L_2 \right) \right)^2 \right]^{1/2} \quad (\text{B.26})$$

with the partial derivatives in the above equation are given by

$$\left(\partial \left(\frac{dP}{dx} \right) / \partial P_1 \right) = \left(\frac{1}{L_2} \right) \quad (\text{B.27})$$

$$\left(\partial \left(\frac{dP}{dx} \right) / \partial P_2 \right) = \left(\frac{-1}{L_2} \right) \quad (\text{B.28})$$

$$\left(\partial \left(\frac{dP}{dx} \right) / \partial L_2 \right) = \left(-\frac{P_1 - P_2}{L_2^2} \right) \quad (\text{B.29})$$

The calculated uncertainties $\delta \left(\frac{dP}{dx} \right)$ are given in Table B.5.

Table B.3: Calculated uncertainties in the superficial velocity of the water.

Run	J_L	Q	95% C.L. Uncertainty	
#	m/s	m ³ /s	m/s	%
Annular flow test runs				
53	0.016	8.16E-06	0.0158	98.76%
54	0.023	1.15E-05	0.0158	68.75%
55	0.202	1.02E-04	0.0177	8.75%
56	0.278	1.40E-04	0.0192	6.90%
57	0.431	2.19E-04	0.0232	5.38%
58	0.599	3.04E-04	0.0284	4.74%
59	0.822	4.16E-04	0.0360	4.38%
126	1.092	5.53E-04	0.0458	4.19%
127	1.061	5.38E-04	0.0447	4.21%
128	1.070	5.42E-04	0.0450	4.20%
129	0.731	3.71E-04	0.0329	4.49%
130	0.293	1.48E-04	0.0195	6.67%
131	0.002	8.02E-07	0.0158	789.41%
132	0.173	8.78E-05	0.0172	9.94%
133	0.357	1.80E-04	0.0211	5.91%
134	0.481	2.44E-04	0.0246	5.12%
135	0.553	2.81E-04	0.0269	4.87%
136	0.674	3.42E-04	0.0309	4.58%
Plug flow test runs				
63	0.918	4.65E-04	0.0394	4.30%
64	0.931	4.72E-04	0.0399	4.29%
67	0.831	4.21E-04	0.0363	4.37%
68	0.829	4.20E-04	0.0362	4.37%
69	0.839	4.25E-04	0.0366	4.36%
70	0.901	4.57E-04	0.0388	4.31%
72	0.259	1.31E-04	0.0188	7.26%
73	0.145	7.35E-05	0.0168	11.58%
74	0.279	1.41E-04	0.0192	6.89%
75	0.526	2.67E-04	0.0260	4.95%
76	0.915	4.64E-04	0.0393	4.30%
81	0.201	1.02E-04	0.0177	8.79%
82	0.477	2.41E-04	0.0245	5.14%
83	0.837	4.24E-04	0.0365	4.36%
84	0.915	4.64E-04	0.0393	4.30%
85	0.266	1.35E-04	0.0190	7.13%
86	0.657	3.33E-04	0.0303	4.61%
87	0.896	4.54E-04	0.0386	4.31%
88	1.061	5.38E-04	0.0447	4.21%

Run	J_L	Q	95% C.L. Uncertainty	
#	m/s	m ³ /s	m/s	%
89	1.076	5.45E-04	0.0452	4.20%
90	0.810	4.11E-04	0.0356	4.40%
91	0.809	4.10E-04	0.0355	4.39%
92	1.022	5.18E-04	0.0432	4.23%
93	0.893	4.53E-04	0.0386	4.32%
94	0.149	7.54E-05	0.0168	11.30%
95	0.221	1.12E-04	0.0180	8.16%
96	0.748	3.79E-04	0.0334	4.46%
102	0.227	1.15E-04	0.0182	8.00%
107	0.275	1.39E-04	0.0191	6.96%
108	0.275	1.39E-04	0.0191	6.96%
109	0.273	1.38E-04	0.0191	7.00%
110	0.499	2.53E-04	0.0252	5.05%
111	0.796	4.03E-04	0.0351	4.40%
Slug flow test runs				
60	0.890	4.51E-04	0.0384	4.32%
61	0.901	4.56E-04	0.0388	4.31%
62	0.911	4.62E-04	0.0392	4.31%
71	0.894	4.53E-04	0.0386	4.31%
77	0.917	4.65E-04	0.0394	4.30%
78	0.911	4.62E-04	0.0392	4.30%
79	0.901	4.56E-04	0.0388	4.30%
80	0.854	4.33E-04	0.0371	4.35%
97	0.741	3.76E-04	0.0332	4.48%
98	0.743	3.77E-04	0.0333	4.48%
99	0.733	3.72E-04	0.0329	4.49%
100	0.797	4.04E-04	0.0351	4.41%
101	0.811	4.11E-04	0.0356	4.39%
103	0.242	1.22E-04	0.0184	7.61%
104	0.236	1.19E-04	0.0183	7.76%
105	0.235	1.19E-04	0.0183	7.79%
106	0.437	2.21E-04	0.0233	5.34%
112	0.951	4.82E-04	0.0406	4.27%
113	1.026	5.20E-04	0.0434	4.23%
114	1.122	5.69E-04	0.0470	4.19%
115	1.173	5.94E-04	0.0488	4.16%
116	1.173	5.94E-04	0.0488	4.16%
117	0.142	7.19E-05	0.0167	11.79%
118	0.207	1.05E-04	0.0178	8.59%
119	0.311	1.57E-04	0.0200	6.42%
120	0.502	2.55E-04	0.0253	5.04%
121	0.710	3.60E-04	0.0321	4.52%

Run	J_L	Q	95% C.L. Uncertainty	
#	m/s	m ³ /s	m/s	%
122	0.824	4.17E-04	0.0360	4.37%
123	1.068	5.41E-04	0.0449	4.21%
124	1.173	5.94E-04	0.0488	4.16%
125	1.168	5.91E-04	0.0486	4.16%
Stratified-wavy flow test runs				
45	0.002	6.12E-09	0.0158	1052.54%
46	0.023	1.18E-05	0.0158	68.76%
47	0.069	3.47E-05	0.0160	23.21%
48	0.051	2.59E-05	0.0159	31.21%
49	0.057	2.90E-05	0.0159	27.98%
50	0.057	2.90E-05	0.0159	27.98%
51	0.002	6.12E-09	0.0158	1052.54%
52	0.017	8.82E-06	0.0158	92.96%

Table B.4: Calculated uncertainties in the superficial velocity of the air.

Run	J_a	G	95% C.L. Uncertainty	
#	m/s	kg/m ² .s	m/s	%
Annular flow test runs				
53	23.346	3.02E+01	0.9603	4.1%
54	22.984	2.98E+01	0.9620	4.2%
55	21.258	2.97E+01	0.8954	4.2%
56	20.939	3.00E+01	0.8730	4.2%
57	19.981	3.01E+01	0.8300	4.2%
58	18.613	2.99E+01	0.7811	4.2%
59	16.811	2.90E+01	0.7287	4.3%
126	17.637	3.22E+01	0.6856	3.9%
127	22.43	4.09E+01	0.6777	3.0%
128	21.349	3.89E+01	0.6755	3.2%
129	22.447	3.96E+01	0.7011	3.1%
130	26.81	4.00E+01	0.8215	3.1%
131	30.062	3.95E+01	0.9355	3.1%
132	27.645	3.96E+01	0.8567	3.1%
133	25.525	3.89E+01	0.8042	3.2%
134	24.593	3.91E+01	0.7713	3.1%
135	23.821	3.89E+01	0.7502	3.1%
136	22.316	3.83E+01	0.7178	3.2%
Plug flow test runs				
63	0.665	8.55E-01	0.9602	144.4%
64	0.095	1.20E-01	0.9764	1027.8%
67	0.807	1.03E+00	0.9744	120.7%
68	1.629	2.11E+00	0.9572	58.8%
69	0.457	5.78E-01	0.9800	214.4%
70	0.454	5.76E-01	0.9772	215.2%
72	0.793	9.72E-01	1.0091	127.3%
73	0.792	9.64E-01	1.0142	128.1%
74	0.787	9.65E-01	1.0085	128.1%
75	0.701	8.72E-01	0.9950	141.9%
76	0.751	9.59E-01	0.9698	129.1%
81	0.221	2.69E-01	1.0112	457.6%
82	0.218	2.69E-01	1.0047	460.9%
83	0.214	2.70E-01	0.9823	459.0%
84	0.215	2.74E-01	0.9746	453.3%
85	0.191	2.34E-01	1.0096	528.6%
86	0.189	2.35E-01	0.9927	525.3%
87	0.19	2.40E-01	0.9790	515.2%
88	0.188	2.40E-01	0.9688	515.3%

Run	J_a	G	95% C.L. Uncertainty	
#	m/s	kg/m ² .s	m/s	%
89	0.181	2.31E-01	0.9699	535.8%
90	1.048	1.34E+00	0.9658	92.2%
91	1.566	2.03E+00	0.9550	61.0%
92	1.448	1.91E+00	0.9372	64.7%
93	0.695	8.92E-01	0.9651	138.9%
94	0.998	1.22E+00	1.0082	101.0%
95	0.995	1.22E+00	1.0043	100.9%
96	0.987	1.26E+00	0.9726	98.5%
102	1.454	1.79E+00	1.0052	69.1%
107	0.224	2.73E-01	1.0135	452.4%
108	1.365	1.67E+00	1.0096	74.0%
109	1.402	1.72E+00	1.0092	72.0%
110	1.313	1.63E+00	0.9949	75.8%
111	1.304	1.66E+00	0.9734	74.6%
Slug flow test runs				
60	4.2	6.16E+00	0.8545	20.3%
61	2.016	2.72E+00	0.9225	45.8%
62	1.672	2.23E+00	0.9324	55.8%
71	1.707	2.23E+00	0.9504	55.7%
77	1.607	2.10E+00	0.9511	59.2%
78	2.22	2.94E+00	0.9369	42.2%
79	4.346	6.03E+00	0.9019	20.8%
80	8.781	1.29E+01	0.8523	9.7%
97	2.417	3.16E+00	0.9473	39.2%
98	4.033	5.42E+00	0.9264	23.0%
99	6.611	9.17E+00	0.8958	13.6%
100	10.362	1.56E+01	0.8366	8.1%
101	13.464	2.12E+01	0.8000	5.9%
103	3.74	4.63E+00	1.0001	26.7%
104	5.95	7.45E+00	0.9836	16.5%
105	13.139	1.70E+01	0.9608	7.3%
106	12.296	1.69E+01	0.9103	7.4%
112	1.292	1.67E+00	0.9610	74.4%
113	1.278	1.66E+00	0.9536	74.6%
114	1.248	1.64E+00	0.9460	75.8%
115	1.266	1.67E+00	0.9378	74.1%
116	1.268	1.68E+00	0.9348	73.7%
117	5.201	6.38E+00	1.0103	19.4%
118	5.062	6.26E+00	1.0020	19.8%
119	5.054	6.35E+00	0.9845	19.5%
120	4.986	6.47E+00	0.9532	19.1%
121	4.762	6.35E+00	0.9283	19.5%

Run	J_a	G	95% C.L. Uncertainty	
#	m/s	kg/m ² .s	m/s	%
122	4.646	6.30E+00	0.9147	19.7%
123	4.434	6.20E+00	0.8855	20.0%
124	4.368	6.25E+00	0.8654	19.8%
125	8.786	1.38E+01	0.7935	9.0%
Stratified-wavy flow test runs				
45	0.373	4.50E-01	1.0225	274.1%
46	0.916	1.11E+00	1.0219	111.6%
47	2.034	2.46E+00	1.0193	50.1%
48	3.036	3.69E+00	1.0142	33.4%
49	7.612	9.31E+00	1.0109	13.3%
50	12.865	1.59E+01	1.0009	7.8%
51	15.344	1.91E+01	0.9931	6.5%
52	19.326	2.45E+01	0.9791	5.1%

Table B.5: Uncertainties in the calculated frictional pressure gradients.

	(dP/dx)	95% C.L. Uncertainty	
Run	Average		
#	kPa/m	Pa/m	%
Annular flow test runs			
53	0.966	13.21	1.37%
54	0.941	13.21	1.40%
55	2.133	13.21	0.62%
56	2.581	13.21	0.51%
57	3.484	13.22	0.38%
58	4.636	13.22	0.29%
59	6.029	13.24	0.22%
126	7.219	13.24	0.18%
127	7.494	13.26	0.18%
128	7.556	13.26	0.18%
129	6.743	13.25	0.20%
130	3.582	13.22	0.37%
131	1.339	13.21	0.99%
132	2.827	13.21	0.47%
133	3.986	13.22	0.33%
134	4.786	13.23	0.28%
135	5.335	13.24	0.25%
136	6.243	13.24	0.21%
Plug flow test runs			
63	0.977	13.21	1.35%
64	0.715	13.21	1.85%
67	0.602	13.20	2.19%
68	0.867	13.21	1.52%
69	0.499	13.20	2.65%
70	0.545	13.20	2.42%
72	0.102	13.20	12.95%
73	0.039	20.74*	52.56%
74	0.107	13.20	12.37%
75	0.292	13.20	4.52%
76	0.675	13.20	1.96%
81	0.081	20.74*	25.65%
82	0.150	13.20	8.78%
83	0.446	13.20	2.96%
84	0.538	13.20	2.46%
85	0.058	13.20	22.75%
86	0.269	13.20	4.90%
87	0.470	13.20	2.81%
88	0.628	13.20	2.10%

	(dP/dx)	95% C.L. Uncertainty	
Run	Average		
#	kPa/m	Pa/m	%
89	0.608	13.20	2.17%
90	0.670	13.20	1.97%
91	0.844	13.21	1.56%
92	1.145	13.21	1.15%
93	0.685	13.20	1.93%
94	0.059	20.74*	35.22%
95	0.118	13.20	11.18%
96	0.587	13.20	2.25%
102	0.153	13.20	8.64%
107	0.076	20.74*	27.25%
108	0.132	13.20	9.98%
109	0.137	13.20	9.62%
110	0.351	13.20	3.76%
111	0.682	13.20	1.94%
Slug flow test runs			
60	2.936	13.21	0.45%
61	1.625	13.21	0.81%
62	1.447	13.21	0.91%
71	0.988	13.21	1.34%
77	0.971	13.21	1.36%
78	1.198	13.21	1.10%
79	1.802	13.21	0.73%
80	2.753	13.21	0.48%
97	0.990	13.21	1.33%
98	1.344	13.21	0.98%
99	1.892	13.21	0.70%
100	3.075	13.21	0.43%
101	3.900	13.22	0.34%
103	0.205	13.20	6.45%
104	0.443	13.20	2.98%
105	0.804	13.20	1.64%
106	1.645	13.21	0.80%
112	0.878	13.21	1.50%
113	1.001	13.21	1.32%
114	1.128	13.21	1.17%
115	1.265	13.21	1.04%
116	1.313	13.21	1.01%
117	0.140	13.20	9.42%
118	0.251	13.20	5.26%
119	0.515	13.20	2.56%
120	1.016	13.21	1.30%

	(dP/dx)	95% C.L. Uncertainty	
Run	Average		
#	kPa/m	Pa/m	%
121	1.438	13.21	0.92%
122	1.678	13.21	0.79%
123	2.221	13.21	0.59%
124	2.616	13.21	0.51%
125	4.197	13.22	0.31%
Stratified-wavy flow test runs			
45	0.012	20.74*	173.26%
46	0.020	20.74*	103.14%
47	0.054	20.74*	38.47%
48	0.129	20.74*	16.08%
49	0.171	13.20	7.71%
50	0.323	13.20	4.09%
51	0.442	13.20	2.99%
52	0.663	13.20	1.99%

Note: (*) As discussed in Section 6.3.1, readings of pressure transducer at the exit of the test section were discarded for these test runs. As a result, the average pressures were determined from the measured static pressures at Port 1 and Port 2, i.e., P_1 and P_2 .

Appendix C**Listing of Computer Codes**

Listing of Computer Codes

C.1 Data Acquisition Program

- (a) **TPDACQS.BAS**: A Quick-Basic computer program used to read the output voltage of the instruments, and to traverse the optical probes along the diameter of the test tube.

C.2 Data Reduction Programs

- (a) **TPDATRED.FOR**: A FORTRAN code developed for the analysis of the experimental data. The program provides the option of processing one experimental test run or all the test runs at the same time.
- (b) **SIGNALS.M**: A MATLAB routine used to identify and loads the untreated signals of the optical probes.
- (c) **HISTOGRAM.M**: A MATLAB routine used to calculate the Probability Density Function (PDF) and draw the histograms of the time traces of the optical probes.
- (d) **VOID.M**: A MATLAB routine used to calculate the local and area-weighted average void fractions.

Note: A number of Excel spreadsheets have been developed to calculate the predictions of some of the models considered in this study, and to plot the figures presented in this document. The main spreadsheets are:

- (1) **TP-REGIME.XLS**: Flow pattern transitions and flow regime maps.
- (2) **TP-PRESSURE.XLS**: Measured and predicted pressure gradients.
- (3) **TP-VOID.XLS**: local and area weighted void fractions.
- (4) **TP-TF_MODELS.XLS**: Two-fluid models and interfacial friction factors.
- (5) **TP-CALIBRATION.XLS**: Calibration of instruments and estimates of uncertainties.

Data Acquisition Program:

TPDACQS.BAS: A Quick-Basic computer program used to read the output voltage of the instruments, and to traverse the optical probes along the diameter of the test tube.


```

COMMON SHARED ASUPV, WSUPV, AVTPR, AINPR, WTFIT
COMMON SHARED GAIR(), MAIR(), GWAT(), MWAT()
COMMON SHARED GAIRAV, GAIRD, GAIRPD, MAIRAV, MAIRD, MAIRPD
COMMON SHARED GWATAV, GWATD, GWATPD, MWATAV, MWATD, MWATPD
COMMON SHARED ROAIRIN, ROAIRMD, ROAIREX, ROAIRAV, ROWATER
COMMON SHARED WINPR, MIXINPR, MIXMDPR, MIXEXPR
COMMON SHARED AIRTEM, WATTEM, MIXTEM
COMMON SHARED PRSATMSV(), PRSATMS(), PATM
COMMON SHARED PRSEXTV(), PRSEXT(), PEXIT
COMMON SHARED PGAGEV(), PGAGE()
COMMON SHARED RMSV(), MDIVV(), DIFF(), PDIFF()
COMMON SHARED SWEEP1, SMPLNG, DELT, SFREQ

' Declare subroutines.

DECLARE SUB INITIALIZE ()
DECLARE SUB LOCKCHN (CH%)
DECLARE SUB MEASURE ()
DECLARE SUB SAMPDAT (CH%, NC, VALUE)
DECLARE SUB ADCNV (CH%, NC, VALUE)
DECLARE SUB TEMPERATURE (CJC, ATEMPR, WTEMPR, MXTEMPR)
DECLARE SUB ERRCODES (FLAG%, SUBR%)
DECLARE SUB SCRNOOUT ()
DECLARE SUB CJCGET (NC, CJC)
DECLARE SUB TEMPINTRP (VOLT, CJC, TC, TF)
DECLARE SUB TEMPLOOKUP ()
DECLARE SUB AVERAGEAD (II, READNGS(), AVERAGEI)
DECLARE SUB WATFLOWRATE ()
DECLARE SUB AIRFLOWRATE ()
DECLARE SUB BORDERLINES ()
DECLARE SUB TESTSETUP ()
DECLARE SUB INTRODUCTION ()
DECLARE SUB CELESCO (AIRVPRSV, AVTPRS)
DECLARE SUB OMEGA15 (OMEGAVI, PRSKPAI)
DECLARE SUB OMEGA05 (OMEGAVI, PRSKPAI)
DECLARE SUB OMEGA01 (OMEGAVI, PRSKPAI)
DECLARE SUB PRBHOME ()
DECLARE SUB PROBE (PPOS%)
DECLARE SUB OUTPUTFILE ()
DECLARE SUB VOIDMEASURE ()

' Declare the DAS8 Driver, this is essential to the execution
' of the DAS8 call. Declaring the following subroutine makes
' the driver known to this module.

DECLARE SUB DAS8 (MODE%, BYVAL dummy%, FLAG%)

' Clear screen, initialize DAS8-PGA and load the thermocouple
' lookup table.
CLS
CALL INTRODUCTION
CALL INITIALIZE

```

```

CALL TEMPLOOKUP
' Call routine to enter experiment initial setup, ie. run number and zero readings.
999 CALL TESTSETUP

' Measure air, water and exit mixture temperature; at beginning of experiment.
LOCATE 22, 3: PRINT " Air, Water and Mixture temperature. "

CALL TEMPERATURE(CJC, ATEMPR, WTEMPR, MXTEMPR)
AIRTEMP(1) = ATEMPR
WATTEMP(1) = WTEMPR
MIXTEMP(1) = MXTEMPR
LOCATE 22, 4: PRINT "

' Drive fibre probe to home position.

'!! CALL PRBHOME

' Perform measurements; at beginning of experiment.
RP% = 1
CALL MEASURE
BEEP: BEEP
' Local void fraction measurements.

'!! CALL VOIDMEASURE

'!! LOCATE 20, 3
'!! PRINT " Void fraction measurements completed proceed? Y or N"
'!! LOCATE 22, 3: INPUT ANSW$: IF ANSW$ = "N" OR ANSW$ = "n" THEN END

' Measure air, water and exit mixture temperature; at end of experiment.
' Calling sub-routine TEMPERATURE send one pulse to the stepper.

LOCATE 20, 3: PRINT "
LOCATE 22, 3: PRINT " Air, Water and Mixture temperature. "

CALL TEMPERATURE(CJC, ATEMPR, WTEMPR, MXTEMPR)
AIRTEMP(2) = ATEMPR
WATTEMP(2) = WTEMPR
MIXTEMP(2) = MXTEMPR

AIRTEM = (AIRTEMP(1) + AIRTEMP(2)) / 2
WATTEM = (WATTEMP(1) + WATTEMP(2)) / 2
MIXTEM = (MIXTEMP(1) + MIXTEMP(2)) / 2

BEEP: BEEP
' Perform measurements; at end of experiment.

RP% = 2
CALL MEASURE
' Air, water flow rates and pressure conversions.

```

```

FOR I = 1 TO 13
  PDIFF(J) = 0
  NEXT J
  Calculate std dev.
  IF ASUPV <> 0! THEN PDIFF(1) = ((DIFF(1) / ASUPV) * 100)
  IF WSUPV <> 0! THEN PDIFF(2) = ((DIFF(2) / WSUPV) * 100)
  IF AVTPR <> 0! THEN PDIFF(3) = ((DIFF(3) / AVTPR) * 100)
  IF MIXINPR <> 0! THEN PDIFF(4) = ((DIFF(4) / MIXINPR) * 100)
  IF MIXMDPR <> 0! THEN PDIFF(5) = ((DIFF(5) / MIXMDPR) * 100)
  IF MIXEXPR <> 0! THEN PDIFF(6) = ((DIFF(6) / MIXEXPR) * 100)
  IF AIRTEM <> 0! THEN PDIFF(7) = ((DIFF(7) / AIRTEM) * 100)
  IF WATTEM <> 0! THEN PDIFF(8) = ((DIFF(8) / WATTEM) * 100)
  IF MIXTEM <> 0! THEN PDIFF(9) = ((DIFF(9) / MIXTEM) * 100)
  Display measurements on the screen.
  CALL SCRNOOUT
  LOCATE 3, 53: PRINT "Record this data? Y or N"
  LOCATE 4, 53: INPUT ANSW$: IF ANSW$ = "N" OR ANSW$ = "n" THEN END
  LOCATE 3, 53
  Record experimental data on a file.
  CALL OUTPUTFILE
  PRINT "Perform another RUN? Y or N"
  LOCATE 4, 53: INPUT ANSW$: IF ANSW$ = "Y" OR ANSW$ = "y" THEN GOTO 999
  CLS
  End of data acquisition program.
  END
  *****
  SUB ADCONV (CH%, NC, VALUE) STATIC
  SUBR% = 1
  This subroutine is used to perform a total number of NC single
  A/D conversions and convert the binary numbers read from
  channel CH% to voltage.
  The passed variables:
  CH% : Number of channel to be sampled.
  NC : Number of A/D conversion for time averaging.
  The returned variables:
  VALUE : Average voltage read from channel # CH%, (Volts)
  DATVAL() : Array to store measured values. (Binary form)
  Size must be greater or equal to NC.
  DIM DATVAL(5000)
  Perform NC A/D conversions on channel CH% using mode 4.
  FOR I = 1 TO NC
    MODE% = 4
    Single 12-bit A/D conversion.

```

```

  FOR I = 1 TO 2
    RP% = I
    AIRPRS = AIRVPRSV(RP%)
    CALL CELESCO(AIRPRS, AVTPRS(RP%))
    CALL OMEGA15(MIXPRS1(RP%), MIXINPRS(RP%))
    CALL OMEGA05(MIXPRS2(RP%), MIXMDPRS(RP%))
    CALL OMEGA01(MIXPRS3(RP%), MIXEXPRS(RP%))
    CALL AIRFLOWRATE
    CALL WATFLOWRATE
    NEXT I
    Calculate average of experimental readings; beginning and end of
    experimental test.
    ASUPV = (ASUPVEL(1) + ASUPVEL(2)) / 2
    WSUPV = (WSUPVEL(1) + WSUPVEL(2)) / 2
    AVTPR = (AVTPRS(1) + AVTPRS(2)) / 2
    MIXINPR = (MIXINPRS(1) + MIXINPRS(2)) / 2
    MIXMDPR = (MIXMDPRS(1) + MIXMDPRS(2)) / 2
    MIXEXPR = (MIXEXPRS(1) + MIXEXPRS(2)) / 2
    Calculate average mass vel. and mass flow rates for air and water.
    GAIRAV = (GAIR(1) + GAIR(2)) / 2
    GWATAV = (GWAT(1) + GWAT(2)) / 2
    MAIRAV = (MAIR(1) + MAIR(2)) / 2
    MWATAV = (MWAT(1) + MWAT(2)) / 2
    Calculate diff. between the two data sets.
    GAIRD = ABS(GAIR(1) - GAIR(2))
    GWATD = ABS(GWAT(1) - GWAT(2))
    MAIRD = ABS(MAIR(1) - MAIR(2))
    MWATD = ABS(MWAT(1) - MWAT(2))
    Calculate standard deviation for air and water act. vel. and
    flow rates.
    AACTVPD = 0
    WACTVPD = 0
    AFLOWPD = 0
    WFLOWPD = 0
    IF GAIRAV <> 0! THEN GAIRPD = ((GAIRD / GAIRAV) * 100)
    IF GWATAV <> 0! THEN GWATPD = ((GWATD / GWATAV) * 100)
    IF MAIRAV <> 0! THEN MAIRPD = ((MAIRD / MAIRAV) * 100)
    IF MWATAV <> 0! THEN MWATPD = ((MWATD / MWATAV) * 100)
    Calculate diff. between data set 1 and 2.
    DIFF(1) = ABS(ASUPVEL(1) - ASUPVEL(2))
    DIFF(2) = ABS(WSUPVEL(1) - WSUPVEL(2))
    DIFF(3) = ABS(AVTPRS(1) - AVTPRS(2))
    DIFF(4) = ABS(MIXINPRS(1) - MIXINPRS(2))
    DIFF(5) = ABS(MIXMDPRS(1) - MIXMDPRS(2))
    DIFF(6) = ABS(MIXEXPRS(1) - MIXEXPRS(2))
    DIFF(7) = ABS(AIRTEMP(1) - AIRTEMP(2))
    DIFF(8) = ABS(WATTEMP(1) - WATTEMP(2))
    DIFF(9) = ABS(MIXTEMP(1) - MIXTEMP(2))

```

```

CALL DAS8(MODE%, VARPTR(D%(CH%)), FLAG%)
IF FLAG% <> 0 THEN CALL ERRCODES(FLAG%, SUBR%)
DATVAL(I) = D%(CH%)
NEXT I
  * Average the NC values read from channel CH%.
  CALL AVERAGE(READNC, DATVAL(I), VALUE)
  *
  * Convert the binary data to volts.
  * Analog Input V = Binary Output X Span resolution.
  * Here User should note the Data Acquisition Board is configured
  * for a UNIPOLAR input range of 0 - 10 Volts, reference [1].
  VALUE = VALUE * 10 / 4095
  *
  RMS = RMS * 10 / 4095
  MDIV = MDIV * 10 / 4095
  * Return to calling module.
  END SUB
  *
  * =====
  SUB AIRFLOWRATE STATIC
  SUBR% = 2
  * This subroutine calculates the air actual velocity, flow-rate
  * and superficial velocity.
  * The air velocity transducer has a linear output of 0 - 5 volts
  * for a standard air velocity of 0 - 12000 sfpm. This gives calibration
  * curve a slope of 2400 sfpm / volts.
  * The air velocity transducer has been re-calibrated, June '94,
  * The transducer new slope is equal to 10.05 smps / volt.
  *
  * The passed variables:
  * ASVELV : Air vel. transducer output voltage (Volts).
  * AVTPRS : Air pressure at vel. transducer (Pa).
  * AIRTEMP : Air inlet temperature (C).
  *
  * The returned variables:
  * AIRFLOW : Air actual flow rate. (m^3/s).
  * ASUPVEL : Air superficial velocity. (m/s).
  *
  * The variables used in this subroutine:
  * ASVEL : Air standard velocity. (sm/s).
  * STDPRES : Standard Air pressure. (Pa).
  * AVTPRSA : Absolute Air pressure at vel. transducer (Pa).
  * STDTEMP : Standard Air temperature (C).
  * AACTVEL : Air actual velocity at position where it's
    measured, inlet pipe 1.5 in I.D., (m / sec).
  * AIRFLOW: Air actual flow rate, in inlet pipe (m^3/s).
  * ADIAM : Diameter of the air inlet pipe (m).
  * AINAREA : Area of the air inlet pipe (m^2).
  * DIAM : Test section diameter (m).
  * AREA : Test section cross-sectional area (m^2).
  * RE : Reynolds number based on pipe diameter and
    the air actual velocity at the center of pipe

```

```

  * (where it is measured).
  * smps/Volt : Standard meter per second per volt of transducer's output.
  *
  * The Air-velocity transducer has a linear output.
  * 10.05 (smps)/volt.
  * Read air flow condition: pressure and temperature.
  * The pressure and temperature standard conditions under which
  * the velocity transducer is calibrated.
  *
  PI = 3.141593
  STDPRES = 101.33 ' kPa
  STDTEMP = 20! ' deg. C
  AIRCONST = 287! ' Air Gas Constant J/kg.K
  *
  * Cross-sectional area of the test section . (m**2)
  DIAMT = TUBEN% * .0254 ' 1" ID Test section
  AREAT = (PI * DIAMT ^ 2) / 4
  *
  AVTPRSA = AVTPRS(RP%) + 101.33 ' kPa absolute.
  AVTDELV = AIRSVELV(RP%) - AIRZR
  ASVEL = AVTDELV * 10.05
  *
  ROAIRIN = (MIXINPRS(RP%) + 101.33) * 1000 / (AIRCONST * (AIRTEMP(RP%) + 273.15))
  ROAIRMD = (MIXMDPRS(RP%) + 101.33) * 1000 / (AIRCONST * (AIRTEMP(RP%) + 273.15))
  ROAIREX = (MIXEXPRS(RP%) + 101.33) * 1000 / (AIRCONST * (AIRTEMP(RP%) + 273.15))
  ROAIRAV = (ROAIRIN + ROAIREX) / 2
  *
  * Air mass velocity and mass flow rate.
  IF AVTDELV < .5 THEN GAIR(RP%) = AVTDELV * 13.613
  IF AVTDELV > .5 THEN GAIR(RP%) = AVTDELV * 23.449
  MAIR(RP%) = GAIR(RP%) * AREAT
  *
  * Air superficial velocity . (m/sec)
  ASUPVEL(RP%) = GAIR(RP%) / ROAIRAV
  *
  * AACTVEL(RP%) = ASVEL * (STDPRES / AVTPRSA) * (AIRTEMP(RP%) / STDTEMP)
  * Find Reynolds number based on actual velocity (where it is
  * measured) and pipe diameter. Air kinematic viscosity is
  * assumed constant and at 20 deg. C,  $\nu = 1.51 \times 10^{-5}$  (m^2/m/sec),
  * from fluid mechanics textbook by Robinson.
  *
  * Area of the air inlet pipe. 1.5" ID.
  ADIAM = 1.5 * .0254
  AINAREA = PI * ADIAM ^ 2 / 4
  *
  * RE = AACTVEL(RP%) * ADIAM / (.0000151#)
  * IF RE < 2400 THEN AACTVEL(RP%) = AACTVEL(RP%) / 2 ' Laminar flow
  * IF RE > 2400 THEN AACTVEL(RP%) = AACTVEL(RP%) * .82 ' Turbulent flow
  * Air flow-rate. (m**3/sec)
  * AIRFLOW(RP%) = AINAREA * AACTVEL(RP%)
  * Return to calling program.
  END SUB

```

```

*=====
SUB AVERAGE(I, READINGS(I), AVERAGE) STATIC
SUBR% = 3
* This subroutine is to average values passed in an array form.
*
* The passed variables:
* I : Number of values to be averaged.
* READINGS : Array of values to be averaged.
* The returned variables:
* AVERAGE : Average value of Readings.
* SUM : Sum of the array values.
SUM = 0
FOR J = 1 TO I
    SUM = SUM + READINGS(J)
NEXT J
AVERAGE = SUM / I
* Calculate RMS values and maximum deviation from average value.
*
SUM = 0
MDIV = 0
FOR J = 1 TO I
    SUM = SUM + (READINGS(J) - AVERAGE) ^ 2
NEXT J
DIV = ABS(READINGS(J) - AVERAGE)
IF DIV > MDIV THEN MDIV = DIV
NEXT J
RMS = SQR(SUM / I)
END SUB
*=====
SUB BORDERLINES STATIC
SUBR% = 4
* This subroutine is used to draw a table that will displayed
* to the user when entering experimental test initial setup.
* This subroutine is called by TESTSETUP subroutine.
* NOTE: The printed characters in this routine may differ from this
* printer's output.
CLS
* Draw horizontal lines.
FOR I = 1 TO 50: LOCATE 1, I: PRINT "A"; NEXT I
FOR I = 1 TO 50: LOCATE 3, I: PRINT "A"; NEXT I
FOR I = 1 TO 50: LOCATE 5, I: PRINT "A"; NEXT I
FOR I = 1 TO 50: LOCATE 7, I: PRINT "A"; NEXT I
FOR I = 1 TO 50: LOCATE 9, I: PRINT "A"; NEXT I
FOR I = 1 TO 50: LOCATE 11, I: PRINT "A"; NEXT I
FOR I = 1 TO 50: LOCATE 13, I: PRINT "A"; NEXT I
FOR I = 1 TO 50: LOCATE 15, I: PRINT "A"; NEXT I
FOR I = 1 TO 50: LOCATE 17, I: PRINT "A"; NEXT I
* Draw vertical lines.
FOR I = 2 TO 16: LOCATE 1, I: PRINT "a"; NEXT I
FOR I = 2 TO 16: LOCATE 1, 50: PRINT "a"; NEXT I
FOR I = 2 TO 12: LOCATE 1, 38: PRINT "a"; NEXT I
LOCATE 14, 25: PRINT "a"
* Draw corners.
LOCATE 1, 1: PRINT "U"
LOCATE 1, 50: PRINT "U"
LOCATE 17, 1: PRINT "A"
LOCATE 17, 50: PRINT "U"
* Draw lines intersection.
LOCATE 3, 1: PRINT "A"
LOCATE 5, 1: PRINT "A"
LOCATE 7, 1: PRINT "A"
LOCATE 9, 1: PRINT "A"
LOCATE 11, 1: PRINT "A"
LOCATE 13, 1: PRINT "A"
LOCATE 15, 1: PRINT "A"
LOCATE 3, 38: PRINT "A"
LOCATE 5, 38: PRINT "A"
LOCATE 7, 38: PRINT "A"
LOCATE 9, 38: PRINT "A"
LOCATE 11, 38: PRINT "A"
LOCATE 13, 38: PRINT "A"
LOCATE 1, 38: PRINT "A"
LOCATE 13, 25: PRINT "A"
LOCATE 15, 25: PRINT "A"
LOCATE 3, 50: PRINT ""
LOCATE 5, 50: PRINT ""
LOCATE 7, 50: PRINT ""
LOCATE 9, 50: PRINT ""
LOCATE 11, 50: PRINT ""
LOCATE 13, 50: PRINT ""
LOCATE 15, 50: PRINT ""
END SUB
*=====
SUB CELESCO (CELESCOV, PRSKPA) STATIC
SUBR% = 13
* From calibration certificate, the transducer's voltage is 0.2975 mV/V.psi.
* With 15 VDC power supply, the transducer's slope becomes equal to
* 4.476 mV/psi .
* The pressure transducer recalibrated, the new slope is
* 16.718 psi/volts, pp 56 Ph.D. 2.
*
PRSPSI = ((CELESCOV - CELESCZR) * 16.718) ' Psi
PRSKPA = PRSPSI * 6.8948 ' kPa
END SUB
*=====
SUB CJCGET (NC, CJC)
SUBR% = 5
* CJCGET is designed to read cold junction compensation temperature
* provided by a sensor built in the EXP-16 board.

```

```

* Output of CJC channel is scaled at 24.4mV/deg.C. This corresponds to
* 0.1 deg.C/bit( 10/4095=2.4 mV/Bit). Dividing output in bits by 10 yields degrees C.
* Note that the CJC channel is connected to DAS-8 channel # 7,
* this could be changed by changing the EXP-16 jumper.

```

```

* The passed variables:
* NC : Number of readings
* The returned variables:
* CJC: Cold junction compensation temperature. ( C)

```

```

CH% = 7
* Next get CJC data from this channel using Mode 4

```

```

CALL SAMPDAT(CH%, NC, VALUE)

```

```

* Change output from volts back to bits
CJ% = VALUE * 4095 / 10
* Change output in bits to real temperature
CJC = CJ% / 10
* Return to calling module.
END SUB

```

```

*****

```

```

SUB ERRCODES (FLAG%, SUBR%) STATIC

```

```

SUBR% = 6

```

```

* This subroutine gives a summary of the error codes:
* Depending on the problem a message will be displayed to the
* user and the program execution ends here. This subroutine
* will be called only if the passed variable FLAG% is different
* from zero.
* For more information consult the appropriate section in
* the DAS-8PGA User's Manual.

```

```

PRINT " The returned FLAG% , SUBR. = "; FLAG%, SUBR%
SELECT CASE FLAG%

```

```

CASE IS > 70
  PRINT " FLAG% > 70 - Error type not specified..!": END
CASE 1
  PRINT "1: Base address unknown.(I/O location not initialized)": END
CASE 2
  PRINT "2: Mode number <0 or > 22.": END
CASE 3
  PRINT "3: Invalid base address, < 256 or > 1016.": END
CASE 4
  PRINT "4: Mux scan limits out of range or reversed order,<0 or >7.": END
CASE 5
  PRINT "5: Channel number out of range, <0 or >7.": END
CASE 6
  PRINT "6: A/D timeout. Incorrect busy signal from A/D.": END
CASE 7
  PRINT "7: Interrupt level out of range, <2 or >7.": END
CASE 8

```

```

PRINT "8: Mode 8 data transfer parameters out of range.": END
CASE 9
  PRINT "9: Mode 9 data transfer parameters out of range.": END
CASE 10
  PRINT "10: Counter number out of range, <0 or >2.": END
CASE 11
  PRINT "11: Counter configuration number out of range, <0 or >5.": END
CASE 12
  PRINT "12: Digital output data out of range, <0 or >15.": END
CASE 13
  PRINT "13: Mode 18 trigger channel number out of range <0 or >7.": END
CASE 14
  PRINT "14: Mode 18 trigger level out of range, <2048 or >2047.": END
CASE 15
  PRINT "15: Mode 18 trigger slope not 0 or 1.": END
CASE 16
  PRINT "16: PGA gain codes illegal, <0 or >3.": END
CASE 17
  PRINT "17: PGA function not available (no DAS-8PGA).": END
CASE 18
  PRINT "18: Mode number unavailable for this language.": END
CASE 19
  PRINT "19: Mode 21 not initialized.": END
CASE 51
  PRINT "51: Fluid switch wafer cannot be set to home position.": END
CASE 52
  PRINT "52: Fluid switch wafer not set on the desired port.": END
CASE 60
  PRINT "60: New probe position out of range, check file - PROBE.POS.": END
END SELECT
END SUB

```

```

* *****
SUB INITIALIZE STATIC
SUBR% = 7

```

```

* This is to Initialize DAS-8PGA. Address is
* stored in DAS.ADR file.

```

```

OPEN "C:DAS8.ADR" FOR INPUT AS #1
INPUT #1, BASADR%
CLOSE #1

```

```

FLAG% = 0
MODE% = 0
* Initialize DAS8-PGA.
CALL DAS8(MODE%, VARPTR(BASADR%), FLAG%)
IF FLAG% <> 0 THEN CALL ERRCODES(FLAG%, SUBR%)

```

```

END SUB

```

```

* *****
SUB INTRODUCTION

```

```

SUBR% = 8
Introduction routine.
FOR I = 15 TO 65: LOCATE 5, I: PRINT "[I]"; NEXT I
FOR I = 15 TO 65: LOCATE 20, I: PRINT "[I]"; NEXT I
FOR I = 6 TO 19: LOCATE 1, 15: PRINT "om"; NEXT I
FOR I = 6 TO 19: LOCATE 1, 65: PRINT "om"; NEXT I
Draw corners.
LOCATE 5, 15: PRINT "É"
LOCATE 5, 65: PRINT "ÿ"
LOCATE 20, 15: PRINT "È"
LOCATE 20, 65: PRINT "¼"

LOCATE 7, 32: PRINT " REGTRANS.BAS "
LOCATE 9, 27: PRINT " Data Acquisition Program"
LOCATE 12, 22: PRINT " FLOW REGIME TRANSITION IN TWO-PHASE"
LOCATE 13, 26: PRINT " HORIZONTAL GAS LIQUID FLOW"
LOCATE 16, 33: PRINT " M.L. LAMARI"
LOCATE 18, 30: PRINT " Carleton University"
LOCATE 19, 33: PRINT " August 1994"

LOCATE 25, 55: PRINT "Please WAIT !!!!!!! "; LOCATE 25, 73
SLEEP 2
END SUB

*****
SUB LOCKCHN (CH%) STATIC
SUBR% = 9
This subroutine when called, it locks DAS8-PGA
to CH% and make sure that the multiplexer is actually
at that channel.

CH% : is the channel number.
Lock channel using MODE 1.
LT%(0) = CH%
LT%(1) = CH%

MODE% = 1
CALL DAS8(MODE%, VARPTR(LT%(0)), FLAG%)
IF FLAG% <> 0 THEN CALL ERRCODES(FLAG%, SUBR%)

Read the multiplexer channel using mode # 3
MODE% = 3
read mux channel
D%(0) = 1
mux addrflag% = 0
CALL DAS8(MODE%, VARPTR(D%(0)), FLAG%)
IF FLAG% <> 0 THEN CALL ERRCODES(FLAG%, SUBR%)
Check if A/D multiplexer is on desired channel.
IF D%(0) <> CH% THEN PRINT " Mux chn is different from CH%"; END
END SUB

*****
SUB MEASURE STATIC
SUBR% = 10
NC is number of readings, if changed here, same change should be done

```

```

for value of NC in OUTPUTFILE routine.

NC = 5000
LOCATE 20, 4: PRINT "Program is running, please WAIT. I "
LOCATE 21, 4: PRINT " Now measuring....."
LOCATE 22, 4: PRINT " Air superficial velocity . "

Measure the air velocity.
CH% = 0
CALL SAMPDAT(CH%, NC, VALUE)
AIRSVLV(RP%) = VALUE
RMSV(RP%, 1) = RMS
MDIVV(RP%, 1) = MDIV

Measure static pressure at air velocity transducer.
This pressure is measured using the Celesco Transducer.

LOCATE 22, 4: PRINT " Air press. at AVT using Celesco Trans."
CH% = 2
CALL SAMPDAT(CH%, NC, VALUE)
AIRVPRSV(RP%) = VALUE
RMSV(RP%, 3) = RMS
MDIVV(RP%, 3) = MDIV

Measure water flow-rate.
LOCATE 22, 4: PRINT " Water superficial velocity . "
CH% = 1
CALL SAMPDAT(CH%, NC, VALUE)
WATFLOWV(RP%) = VALUE
RMSV(RP%, 2) = RMS
MDIVV(RP%, 2) = MDIV

EXP-16 is connected to channel #6, see jumper on EXP-16 board.

CH% = 6
CALL LOCKCHN(CH%)

LOCATE 22, 4: PRINT " Test tube static pres: OMEGA 1,2 and 3"

FOR SBCH% = 3 TO 5
MODE% = 14
CALL DAS8(MODE%, VARPTR(SBCH%), FLAG%)
IF FLAG% <> 0 THEN CALL ERRCODES(FLAG%, SUBR%)

CALL ADCONV(SBCH%, NC, VALUE)
OMEGAV = VALUE

SELECT CASE SBCH%
CASE 3
MIXPRSV1(RP%) = OMEGAV
RMSV(RP%, 4) = RMS
MDIVV(RP%, 4) = MDIV

```

```

CASE 4
  MIXPRSV2(RP%) = OMEGAV
  RMSV(RP%, 5) = RMS
  MDIVV(RP%, 5) = MDIV
CASE 5
  MIXPRSV3(RP%) = OMEGAV
  RMSV(RP%, 6) = RMS
  MDIVV(RP%, 6) = MDIV
END SELECT
NEXT SBCH%
  Return to calling module.
END SUB

SUB OMEGA01 (OMEGAV, PRSKPA) STATIC
  SUBR% = 13
  This subroutine is used to calculate the pressure when OMEGA1
  transducer is used. OMEGA01 model PX136; (0 to 1 Psig Range)
  From calibration, see Ph.D. 4 p. 31, the transducer's slope
  is 1.449 kPa/Volts. (of amplified output voltage).
  The OMEGA01 pressure transducer was calibrated Dec. '95
  PRSKPA = (OMEGAV - OMEG01ZR) * 1.449 ' kPa (d)
  PRSPSI = PRSKPA / 6.8949 ' Psid
  Return to calling module.
END SUB

SUB OMEGA05 (OMEGAV, PRSKPA) STATIC
  SUBR% = 13
  This subroutine is used to calculate the pressure when OMEGA1
  transducer is used. OMEGA1 model PX26; (0 to 5 Psid Range)
  From calibration, see Ph.D. 4 p. 31, the transducer's slope
  is 7.014 kPa/Volts. (of amplified output voltage).
  The OMEGA05 pressure transducer was calibrated Dec. '95
  PRSKPA = (OMEGAV - OMEG05ZR) * 7.014 ' kPa (d)
  PRSPSI = PRSKPA / 6.8949 ' Psid
  Return to calling module.
END SUB

SUB OMEGA15 (OMEGAV, PRSKPA) STATIC
  SUBR% = 13
  This subroutine is used to calculate the pressure when OMEGA15
  transducer is used. OMEGA15 model PX26; (0 to 15 Psid Range)
  From calibration, see Ph.D. 4 p. 31, the transducer's slope
  is 20.796 kPa/Volts. (of amplified output voltage).
  The OMEGA15 pressure transducer was calibrated Dec. '95
  PRSKPA = (OMEGAV - OMEG15ZR) * 20.796 ' kPa (d)
  PRSPSI = PRSKPA / 6.8949 ' Psid

```

```

' Return to calling module.
END SUB

' =====
SUB OUTPUTFILE STATIC
  SUBR% = 11
  This sub. is conceived to produced an output file for each
  of test run. The file is called "RUN+RUNUMB%+.DAT"

  OPEN "A:FILEN.NAM" FOR OUTPUT AS #1
  PRINT #1, RUNUMB%
  CLOSE #1
  OPEN "A:FILEN.NAM" FOR INPUT AS #1
  INPUT #1, FLN2$
  CLOSE #1
  FLN1$ = "B:RUN"
  FLN3$ = ".DAT"
  FLN4$ = ".VOL"
  FLNAME$ = FLN1$ + FLN2$ + FLN3$
  FLNAME2$ = FLN1$ + FLN2$ + FLN4$ ' For voltages output

  OPEN FLNAME$ FOR OUTPUT AS #1

  Experimental data output file part #1; i.e. first page.
  PRINT #1, ""
  PRINT #1, ""
  FORMAT$ = "      Test tube I.D.: #.#### m      Experiment no.: #####"
  PRINT #1, USING FORMAT$; DIAMT; RUNUMB%
  PRINT #1, SPC(10); "      Exp. data part: 1"
  PRINT #1, "      #1, SPC(10); "
  PRINT #1, ""
  PRINT #1, "      Data stored on disk no : #### Paddle wheel zero read: #.####"
  PRINT #1, USING "      Video tape counter : ##### Celesco zero reading : #.#### (V)";
  VCOUNT%; CELESCZR
  PRINT #1, USING "      Lab. Temperature(deg.C.): ##.## OMEGA15 zero reading : #.####"
  (V); TAMB; OMEG15ZR
  PRINT #1, USING "      Water surf. tension mN/m: ##.## OMEGA05 zero reading : #.####"
  (V); WSTEN; OMEG05ZR
  PRINT #1, USING "      AVT zero reading (V) : #.#### OMEGA01 zero reading : #.####"
  (V); AIRZR; OMEG01ZR
  PRINT #1, SPC(10); "      "

  PRINT #1, ""
  PRINT #1, SPC(10); " Mixture flow pattern : "; PATTERNS
  PRINT #1, SPC(10); "
  PRINT #1, "
  PRINT #1, SPC(10); " Air / Water phases      Set #1 Set #2 Diff. % Var. Average"
  PRINT #1, SPC(10); "
  FORMAT$ = "      Air sup. velocity (m/s) ##.#### ##.#### ##.#### ##.####"
  PRINT #1, USING FORMAT$; ASUPVEL(1); ASUPVEL(2); DIFF(1); PDIFF(1); ASUPV

```



```
PRINT #1, SPC(14); J; SPC(1); " ..."; SPC(4); FLVOID$(I); SPC(5); FLVOID$(I + 1); SPC(5)(J)
FLSGLN$(U)
J = J + 1
NEXT I
PRINT #1, ""
PRINT
=====
#1,                                     SPC(10);
=====
PRINT #1, SPC(10); " Part 2 of file: "; RIGHT$(FLNAME$, 12); SPC(5); "Date: "; DATE$;
SPC(3); "Time: ", TIME$
,
CLOSE #1
,
OPEN "A:REGTRANS.DAT" FOR APPEND AS #2
    PRINT #2, ""
    FORMAT$ = ##### ##.###### ##.###### ##.###### ##.###### ##.######
    PRINT #2, USING FORMAT$: RUNUMB%; ASUPV; GAIRAV; MAIRAV; ROAIRAV; WSUPV;
GWATAV; MWATAV; ROWATER
    PRINT #2, USING FORMAT$: RUNUMB%; AVTPR; MIXINPR; MIXMDPR; MIXEXPR;
AIRTEM; WATTEM; MIXTEM; WSTEN
    CLOSE #2
,
OPEN "A:REGTRANS.VOL" FOR APPEND AS #3
    PRINT #3, ""
    FORMAT$ = ##### ##.#### ##.#### ##.#### ##.#### ##.#### ##.### ##.##
    PRINT #3, USING FORMAT$: RUNUMB%; AIRZR; WATRZ; CELESCZR; OMEG15ZR;
OMEG05ZR; OMEG01ZR
    FORMAT$ = ##### ##.#### ##.#### ##.#### ##.#### ##.#### ##.### ##.##
    PRINT #3, USING FORMAT$: RUNUMB%; AIRSVELV(1); WATERLOWV(1); AIRVPRSV(1);
MIXPRSV1(1); MIXPRSV2(1); MIXPRSV3(1); AIRTEMP(1); WATERMP(1); MIXTEMP(1)
    PRINT #3, USING FORMAT$: RUNUMB%; AIRSVELV(2); WATERLOWV(2); AIRVPRSV(2);
MIXPRSV2(2); MIXPRSV3(2); AIRTEMP(2); WATERMP(2); MIXTEMP(2)
    CLOSE #3
,
OPEN FLNAME$2$ FOR OUTPUT AS #4
    FORMAT$ = #####
    PRINT #4, USING FORMAT$: RUNUMB%
    FORMAT$ = ###.#### ##.#### ##.#### ##.#### ##.#### ##.####
    PRINT #4, USING FORMAT$: AIRZR; WATRZ; CELESCZR; OMEG15ZR; OMEG05ZR;
OMEG01ZR
    FORMAT$ = ###.#### ##.#### ##.#### ##.#### ##.#### ##.#### ##.### ##.###
    PRINT #4, USING FORMAT$: AIRSVELV(1); WATERLOWV(1); AIRVPRSV(1); MIXPRSV1(1);
MIXPRSV2(1); MIXPRSV3(1); AIRTEMP(1); WATERMP(1)
    PRINT #4, USING FORMAT$: AIRSVELV(2); WATERLOWV(2); AIRVPRSV(2); MIXPRSV2(2);
MIXPRSV3(2); AIRTEMP(2); WATERMP(2); MIXTEMP(2)
    CLOSE #4
Return to calling module.
END SUB
*****
SUB PRBHOME STATIC
SUBR% = 12
```

- ```

' This subroutine PRHOME is conceived to step the fiber=optic
' probe to the home position,i.e. lower position of the test tube.
' The routine reads the current position of the probe from
' data file called "PROBE.POS", changes direction of the stepper
' and then set the probe to the home position. Note that the
' current position of the probe is updated everytime the probe
' is traverssed.
' The routine, also displays the remaining number of steps for the
' probe to reach the final destination.
LOCATE 20, 4
PRINT "Moving Probe to home position; Please WAIT..."
LOCATE 21, 10: PRINT "Remaining number of steps: "
' Read probe current position.
'
OPEN "A:PROBE.POS" FOR INPUT AS #1
INPUT #1, TNSTEPS%
CLOSE #1
'
' Limit probe traverssing distance to a maximum of 1 inch;
' i.e. the test section tube I.D. 1000 steps limit.
'
IF TNSTEPS% > 1701 THEN FLAG% = 60
IF FLAG% <> 0 THEN CALL ERRCODES(FLAG%, SUBR%)
'
' Mode 14 is used to write to digital output ports. In order
' to step the motor a positive edge pulse is required thru
' port OP=1. This is accomplished by setting OUTP% to 14 then 15.
' Note that OP=2 is kept high in order for the probe to step
' downward, i.e. Motor turning ccw.
'
' !!!
PPP1% = &H37A
PPP2% = &H378
'
OUT PPP2%, 0
'
FOR I = 1 TO TNSTEPS%
LOCATE 21, 50: PRINT TNSTEPS% - I
'
' The following loop is used to allow some time before sending
' the next stepping pulse, from initial testing, 120 seems reasonable.
'
OUT PPP1%, 1
FOR W = 1 TO 120: WAITING% = WAITING% + PULSE: NEXT W
OUT PPP1%, 0
FOR W = 1 TO 120: WAITING% = WAITING% + PULSE: NEXT W
'
NEXT I
'
' At this point the probe is at home position, reset TNSTEPS%

```

```

* variable and update PROBE.POS file.
*
* OPEN "A:PROBE.POS" FOR OUTPUT AS #1
* TNSTEPS% = 0
* PRINT #1, TNSTEPS%
* CLOSE #1
*
* LOCATE 20, 4
* PRINT " "
* LOCATE 21, 9
* PRINT " "
*
* END SUB
*=====
* SUB PROBE (PPOS%)
* SUBR% = 14
*
* This subroutine, PRHOME, is conceived to step the fiber-optic
* probe to the next local void fraction measurement position.
* Stepping the probe proceeds from lower to higher position
* within the test section tube. Probe moving upward, i.e. motor
* running cw.
* The actual position of the probe is updated everytime the probe
* is traversed.
*
* The stepping Controller MODE is grounded, for HALF mode operation.
* To operate the stepper at FULL step, for more accuracy, the controller
* MODE should be set at high.
* In HALF mode 400 steps result in one revolution. Lead screw is double
* threaded at 10 threads per inch total (trav. 0.2 in per rev. or
* 400 seteps).
* PSTEP% = 200 i.e. half a turn == 0.1 inch displacement.
*
* PPP1% = &H37A
* PPP2% = &H378
* PSTEP% = 200
*
* OUT PPP2%, 1
*
* FOR I = 1 TO PSTEP%
* LOCATE 23, 50: PRINT PSTEP% - I
*
* The following loop is used to allow some time before sending
* the next stepping pulse, 120 seems reasonable.
*
* OUT PPP1%, 1
* FOR W = 1 TO 120: WAITING% = WAITING + PULSE: NEXT W
* OUT PPP1%, 0
* FOR W = 1 TO 120: WAITING% = WAITING + PULSE: NEXT W
*
* NEXT I
*
* TNSTEPS% = TNSTEPS% + PSTEP%
*
* Update PROBE.POS file.
* OPEN "A:PROBE.POS" FOR OUTPUT AS #1
* PRINT #1, TNSTEPS%
* CLOSE #1
*
* END SUB
*=====
* SUB SAMPDAT (CH%, NC, VALUE) STATIC
* SUBR% = 15
*
* This subroutine is designed to take a sample measurement
* of the specified channel.
* CALL LOCKCHN(CH%)
* CALL ADCONV(CH%, NC, VALUE)
* END SUB
*=====
* SUB SCRNOOUT STATIC
* SUBR% = 16
*
* This is used to display the experimental data on the computer screen.
*
* CLS
* LOCATE 1, 1
* PRINT "-----"
* PRINT USING "Run Number... : #### ; Video counter No.: ####": RUNUMB%, VCOUNT%
* PRINT "-----"
* PRINT "Flow type.....", PATTERNS$
* PRINT "-----"
* PRINT " % Dev. Avg. Value "
* PRINT "-----"
* PRINT USING "Air sup. vel..... ##.#, ##.### m/sec.": PDIFF(1); ASUPV
* PRINT USING "Water sup. vel..... ##.#, ##.### m/sec.": PDIFF(2); WSUPV
* PRINT "-----"
* PRINT USING "Air mass vel..... ##.#, ##.### kg/sm2.": GAIRPD; GAIRAV
* PRINT USING "Water mass vel..... ##.#, ##.### kg/sm2.": GWATPD; GWATAV
* PRINT "-----"
* PRINT USING "Air mass flow rate... ##.#, ##.### kg/s.": MAIRPD; MAIRAV
* PRINT USING "Water mass flow rate.. ##.#, ##.### kg/s.": MWATPD; MWATAV
* PRINT "-----"
* PRINT USING "Air press. @ trans.... ##.#, ##### KPa.": PDIFF(3); AVTPR
* PRINT USING "Tube inlet press..... ##.#, ##### KPa.": PDIFF(4); MIXINPR
* PRINT USING "Tube press. @ probe... ##.#, ##### KPa.": PDIFF(5); MIXMDPR
* PRINT USING "Tube exit press..... ##.#, ##### KPa.": PDIFF(6); MIXEXPR
* PRINT "-----"
* PRINT USING "Air temperature..... ##.#, ##### Deg.C.": PDIFF(7); AIRTEM
* PRINT USING "Water temperature.... ##.#, ##### Deg.C.": PDIFF(8); WATTEM
* PRINT USING "Mixture exit temp.... ##.#, ##### Deg.C.": PDIFF(9); MIXTEM
* PRINT "-----"
* END SUB
*=====
* SUB TEMPERATURE (CJC, ATEMPR, WTEMPR, MTEMPR) STATIC

```

```

SUBR% = 19
* This subroutine measures air, water and two-phase mixture
* temperatures. This is performed as follow:
* 1 - Lock DAS8-PGA to EXP-16 channel. (# 6 in this case)
* 2 - Measure output voltage of thermocouple on EXP-16
* sub-multiplexer.
* 3 - Measure temperature of connector block from CJC channel.
* 4 - Convert, correct and linearize thermocouple output
* to degrees
*
* The returned variables:
* ATEMP : Air inlet temperature, thermocouple 1. (deg. C).
* WTEMP : Water inlet temperature, thermocouple 2. (deg. C).
* MXTEMP : Mixture exit temperature, thermocouple 3. (deg. C).
* The local variables:
* AV : EXP-16 gain setting. (see dip switch on EXP-16 board)
* NC : Number of A/D conversions, whenever sub. ADCONV is called.
* ATEMPR : Air thermocouple voltage output. (Volts)
* WTEMPR : Water thermocouple voltage output. (Volts)
* MXTEMPR : Mixture thermocouple voltage output. (Volts)
*
* TC : Interpolated temperature in degrees Centigrade (deg. C).
* TF : Interpolated temperature in degrees Fahrenheit (deg. F).
* AV = 200
* NC = 1000
* Lock DAS8-PGA to channel that EXP-16 is connected to, here
* it channel number 6. Change to suit.
* CH% = 6
* CALL LOCKCHN(CH%)
* FOR SBCH% = 0 TO 2
* Use mode 14 to set the EXP-16 sub-multiplexer channel.
* MODE% = 14
* CALL DAS8(MODE%, VARPTR(SBCH%), FLAG%)
* IF FLAG% <> 0 THEN CALL ERRCODES(FLAG%, SUBR%)
* Perform A/D conversion of sub-multiplexer channel.
* CALL ADCONV(SBCH%, NC, VALUE)
* SELECT CASE SBCH%
* CASE 0
* ATEMPR = VALUE / AV
* CASE 1
* WTEMPR = VALUE / AV
* CASE 2
* MXTEMPR = VALUE / AV
* END SELECT
* NEXT SBCH%
*
* Get cold compensation temperature. CJC (deg. C)
* CALL CJCGET(NC, CJC)
*
* Call the interpolation subroutine to find T-type thermocouple
* temperature for air, water and the phase mixture.
* CALL TEMPINTRP(ATEMPR, CJC, TC, TF)
* ATEMPR = TC

```

---

```

CALL TEMPINTRP(WTEMPR, CJC, TC, TF)
WTEMPR = TC
CALL TEMPINTRP(MXTEMPR, CJC, TC, TF)
MXTEMPR = TC
END SUB
*=====
SUB TEMPINTRP (VOLT, CJC, TC, TF) STATIC
SUBR% = 20
* Interpolation routine to find "T" thermocouple temperature
* Entry variables:
* CJC = cold junction compensator temperature in deg. C.
* VOLT= thermocouple voltage in volts
* Exit variables:
* TC = temperature in (deg. C).
* TF = temperature in (deg. F).
* Perform CJC compensation for "T" type, see reference [1].
* VT = 1000 * VOLT + .992 + (CJC - 25) * .040667*VT in mV
*
* Find look up element
* ET = INT((VT - SVT) / SIT)
* IF ET < 0 THEN TC = TT(0): GOTO BOUND'Out of bounds, round to lower limit
* IF ET > NT - 2 THEN TC = TT(NT - 1): GOTO BOUND'Out of bounds,round to upper limit
* Do Interpolation
* TC = TT(ET) + (TT(ET + 1) - TT(ET)) * (VT - ET * SIT - SVT) / SIT'Centigrade
* BOUND: TF = TC * 9 / 5 + 32
* 'Fahrenheit
* END SUB
*=====
SUB TEMPLOOKUP STATIC
SUBR% = 21
* This subroutine loads the "T" Type thermocouple lookup
* tables. This data is stored in A:T_TEMP1.DAT. The
* information read from this file are in common shared variables.
*
* NT: Number of data points.
* SIT: Voltage step interval (mV).
* SVT: Starting voltage (mV).
* TT() : T type thermocouple temperature.
* Open the file where the "T" type thermocouple data is stored.
*
* OPEN "A:T_TEMP1.DAT" FOR INPUT AS #1
* Read values in that file.
*
* INPUT #1, NT, SIT, SVT
* FOR L = 0 TO NT - 1
* INPUT #1, TT(L)
* NEXT L
* Close data file and return to calling routine.
* CLOSE #1
* END SUB
*=====
SUB TESTSETUP STATIC

```

```

SUBR% = 22
* This subroutine allows the user to enter the experimental initial
* setup parameters. Enter values only if different from those used
* in previous experiment. Hit return to accept the displayed values.
CLS
* Read initial setup variables, values from previous experiment.
OPEN "A:DACS.INI" FOR INPUT AS #1
INPUT #1, TUBEN%
INPUT #1, RUNUMB%
INPUT #1, DATDISK%
INPUT #1, VCOUNT%
INPUT #1, SWEEPL
INPUT #1, SMPLNG
INPUT #1, TAMB
INPUT #1, CELESCZR
INPUT #1, OMEG15ZR
INPUT #1, OMEG05ZR
INPUT #1, OMEG01ZR
CLOSE #1
*
CALL BORDERLINES
*
LOCATE 2, 3
PRINT "Test tube number "
LOCATE 4, 3
PRINT "Experiment number :"
LOCATE 6, 3
PRINT "Exp. data stored on disk no.:"
LOCATE 8, 3
PRINT "Video tape counter number.:"
LOCATE 10, 3
PRINT "Scope setup.:"
LOCATE 12, 3
PRINT "Lab. temperature deg. C.:"
LOCATE 14, 3
PRINT "Mixture flow pattern.:"
LOCATE 16, 3
PRINT "Zero readings (V):"
*
LOCATE 2, 41, 1, 0, 7
VACCEP% = TUBEN%; PRINT TUBEN%; LOCATE 2, 40: INPUT TUBEN%
IF TUBEN% = 0 THEN TUBEN% = VACCEP
LOCATE 2, 40: PRINT " "; LOCATE 2, 42: PRINT TUBEN%
*
LOCATE 4, 41
VACCEP% = RUNUMB%; PRINT RUNUMB%; LOCATE 4, 40: INPUT RUNUMB%
IF RUNUMB% = 0 THEN RUNUMB% = VACCEP
LOCATE 4, 40: PRINT " "; LOCATE 4, 42: PRINT RUNUMB%
*
VACCEP% = DATDISK%; LOCATE 6, 41: PRINT DATDISK%; LOCATE 6, 40: INPUT
DATDISK%
IF DATDISK% = 0 THEN DATDISK% = VACCEP
LOCATE 6, 40: PRINT " "; LOCATE 6, 42: PRINT DATDISK%
*
VACCEP% = VCOUNT%; LOCATE 8, 41: PRINT VCOUNT%; LOCATE 8, 40: INPUT
VCOUNT%
IF VCOUNT% = 0 THEN VCOUNT% = VACCEP
LOCATE 8, 40: PRINT " "; LOCATE 8, 42: PRINT VCOUNT%
*
LOCATE 10, 16
PRINT "No. of samples"
*
VACCEP% = SWEEPL: LOCATE 10, 40: PRINT SWEEPL: LOCATE 10, 39: INPUT SWEEPL
IF SWEEPL = 0 THEN SWEEPL = VACCEP
LOCATE 10, 39: PRINT " "; LOCATE 10, 39: PRINT SWEEPL
*
LOCATE 10, 16
PRINT "Time Scale:Div.(sec)"
*
VACCEP% = SMPLNG: LOCATE 10, 40: PRINT SMPLNG: LOCATE 10, 39: INPUT SMPLNG
IF SMPLNG = 0 THEN SMPLNG = VACCEP
LOCATE 10, 39: PRINT " "; LOCATE 10, 39: PRINT SMPLNG
*
* Calculate sampling frequency (SFREQ) and sampling time (DELT).
SFREQ = SWEEPL / (SMPLNG * 10I) ' SMPLNG is in seconds
DELT = (SMPLNG * 10I) ' DELT is in seconds
*
VACCEP% = TAMB: LOCATE 12, 41: PRINT TAMB: LOCATE 12, 40: INPUT TAMB
IF TAMB = 0 THEN TAMB = VACCEP
LOCATE 12, 40: PRINT " "; LOCATE 12, 41: PRINT TAMB
*
LOCATE 8, 53: PRINT "B -- Dispersed Bubble Flow."
LOCATE 9, 53: PRINT "P -- Plug Flow."
LOCATE 10, 53: PRINT "ST -- Stratified Flow."
LOCATE 11, 53: PRINT "W -- Wavy Flow."
LOCATE 12, 53: PRINT "S -- Slug Flow."
LOCATE 13, 53: PRINT "A -- Annular Flow."
*
LOCATE 15, 53: PRINT "AIR -- Single Phase AIR."
LOCATE 16, 53: PRINT "WAT -- Single Phase WATER."
LOCATE 17, 53: PRINT "ZR -- Test; Zero Readings."
LOCATE 14, 40
*
VACCEP$ = PATTERN$: LOCATE 14, 28: PRINT PATTERN$: LOCATE 14, 26: INPUT
PATTERN$
*
SELECT CASE PATTERN$
CASE IS = ""
PATTERN$ = VACCEP$
CASE IS = "B", IS = "b"

```

```

SUBR% = 22
* This subroutine allows the user to enter the experimental initial
* setup parameters. Enter values only if different from those used
* in previous experiment. Hit return to accept the displayed values.
CLS
* Read initial setup variables, values from previous experiment.
OPEN "A:DACS.INI" FOR INPUT AS #1
INPUT #1, TUBEN%
INPUT #1, RUNUMB%
INPUT #1, DATDISK%
INPUT #1, VCOUNT%
INPUT #1, SWEEPL
INPUT #1, SMPLNG
INPUT #1, TAMB
INPUT #1, CELESCZR
INPUT #1, OMEG15ZR
INPUT #1, OMEG05ZR
INPUT #1, OMEG01ZR
CLOSE #1
*
CALL BORDERLINES
*
LOCATE 2, 3
PRINT "Test tube number "
LOCATE 4, 3
PRINT "Experiment number :"
LOCATE 6, 3
PRINT "Exp. data stored on disk no.:"
LOCATE 8, 3
PRINT "Video tape counter number.:"
LOCATE 10, 3
PRINT "Scope setup.:"
LOCATE 12, 3
PRINT "Lab. temperature deg. C.:"
LOCATE 14, 3
PRINT "Mixture flow pattern.:"
LOCATE 16, 3
PRINT "Zero readings (V):"
*
LOCATE 2, 41, 1, 0, 7
VACCEP% = TUBEN%; PRINT TUBEN%; LOCATE 2, 40: INPUT TUBEN%
IF TUBEN% = 0 THEN TUBEN% = VACCEP
LOCATE 2, 40: PRINT " "; LOCATE 2, 42: PRINT TUBEN%
*
LOCATE 4, 41
VACCEP% = RUNUMB%; PRINT RUNUMB%; LOCATE 4, 40: INPUT RUNUMB%
IF RUNUMB% = 0 THEN RUNUMB% = VACCEP
LOCATE 4, 40: PRINT " "; LOCATE 4, 42: PRINT RUNUMB%
*
VACCEP% = DATDISK%; LOCATE 6, 41: PRINT DATDISK%; LOCATE 6, 40: INPUT
DATDISK%
IF DATDISK% = 0 THEN DATDISK% = VACCEP
LOCATE 6, 40: PRINT " "; LOCATE 6, 42: PRINT DATDISK%
*
VACCEP% = VCOUNT%; LOCATE 8, 41: PRINT VCOUNT%; LOCATE 8, 40: INPUT
VCOUNT%
IF VCOUNT% = 0 THEN VCOUNT% = VACCEP
LOCATE 8, 40: PRINT " "; LOCATE 8, 42: PRINT VCOUNT%
*
LOCATE 10, 16
PRINT "No. of samples"
*
VACCEP% = SWEEPL: LOCATE 10, 40: PRINT SWEEPL: LOCATE 10, 39: INPUT SWEEPL
IF SWEEPL = 0 THEN SWEEPL = VACCEP
LOCATE 10, 39: PRINT " "; LOCATE 10, 39: PRINT SWEEPL
*
LOCATE 10, 16
PRINT "Time Scale:Div.(sec)"
*
VACCEP% = SMPLNG: LOCATE 10, 40: PRINT SMPLNG: LOCATE 10, 39: INPUT SMPLNG
IF SMPLNG = 0 THEN SMPLNG = VACCEP
LOCATE 10, 39: PRINT " "; LOCATE 10, 39: PRINT SMPLNG
*
* Calculate sampling frequency (SFREQ) and sampling time (DELT).
SFREQ = SWEEPL / (SMPLNG * 10I) ' SMPLNG is in seconds
DELT = (SMPLNG * 10I) ' DELT is in seconds
*
VACCEP% = TAMB: LOCATE 12, 41: PRINT TAMB: LOCATE 12, 40: INPUT TAMB
IF TAMB = 0 THEN TAMB = VACCEP
LOCATE 12, 40: PRINT " "; LOCATE 12, 41: PRINT TAMB
*
LOCATE 8, 53: PRINT "B -- Dispersed Bubble Flow."
LOCATE 9, 53: PRINT "P -- Plug Flow."
LOCATE 10, 53: PRINT "ST -- Stratified Flow."
LOCATE 11, 53: PRINT "W -- Wavy Flow."
LOCATE 12, 53: PRINT "S -- Slug Flow."
LOCATE 13, 53: PRINT "A -- Annular Flow."
*
LOCATE 15, 53: PRINT "AIR -- Single Phase AIR."
LOCATE 16, 53: PRINT "WAT -- Single Phase WATER."
LOCATE 17, 53: PRINT "ZR -- Test; Zero Readings."
LOCATE 14, 40
*
VACCEP$ = PATTERN$: LOCATE 14, 28: PRINT PATTERN$: LOCATE 14, 26: INPUT
PATTERN$
*
SELECT CASE PATTERN$
CASE IS = ""
PATTERN$ = VACCEP$
CASE IS = "B", IS = "b"

```

```

PATTERN$ = " Dispersed Bubble Flow."
CASE IS = "p", IS = "p"
PATTERN$ = " Plug Flow. "
CASE IS = "st", IS = "st", IS = "st", IS = "st"
PATTERN$ = " Stratified Flow."
CASE IS = "w", IS = "w"
PATTERN$ = " Wavy Flow. "
CASE IS = "s", IS = "s"
PATTERN$ = " Slug Flow. "
CASE IS = "a", IS = "a"
PATTERN$ = " Annular Flow. "

CASE IS = "AIR", IS = "air"
PATTERN$ = " Single Phase AIR. "
CASE IS = "WAT", IS = "wat"
PATTERN$ = " Single Phase WATER. "
CASE IS = "ZR", IS = "zr"
PATTERN$ = " Test; Zero Readings. "

END SELECT
LOCATE 14, 26: PRINT "
LOCATE 14, 27: PRINT PATTERN$

FOR I = 8 TO 17
LOCATE I, 53: PRINT "
NEXT I

LOCATE 16, 22
PRINT "Air vel. trans. "
VACCEP = AIRZR: LOCATE 16, 41: PRINT AIRZR: LOCATE 16, 40: INPUT AIRZR
IF AIRZR = 0 THEN AIRZR = VACCEP
LOCATE 16, 40: PRINT " "; LOCATE 16, 41: PRINT AIRZR

LOCATE 16, 22
PRINT "Water flow meter"
VACCEP = WATZR: LOCATE 16, 41: PRINT WATZR: LOCATE 16, 40: INPUT WATZR
IF WATZR = 0 THEN WATZR = VACCEP
LOCATE 16, 40: PRINT " "; LOCATE 16, 41: PRINT WATZR

LOCATE 16, 22
PRINT "Sensor fitting "
VACCEP = WTFIT: LOCATE 16, 43: PRINT WTFIT: LOCATE 16, 40: INPUT WTFIT
IF WTFIT = 0 THEN WTFIT = VACCEP
LOCATE 16, 40: PRINT " "; LOCATE 16, 41: PRINT WTFIT

LOCATE 16, 22
PRINT "Celeco trans. "
VACCEP = CELESCZR: LOCATE 16, 41: PRINT CELESCZR: LOCATE 16, 40: INPUT
CELESCZR
IF CELESCZR = 0 THEN CELESCZR = VACCEP
LOCATE 16, 40: PRINT " "; LOCATE 16, 41: PRINT CELESCZR

LOCATE 16, 22
PRINT "OMEGA15 trans. "
VACCEP = OMEG15ZR: LOCATE 16, 41: PRINT OMEG15ZR: LOCATE 16, 40: INPUT
OMEG15ZR
IF OMEG15ZR = 0 THEN OMEG15ZR = VACCEP
LOCATE 16, 40: PRINT " "; LOCATE 16, 41: PRINT OMEG15ZR

LOCATE 16, 22
PRINT "OMEGA05 trans. "
VACCEP = OMEG05ZR: LOCATE 16, 41: PRINT OMEG05ZR: LOCATE 16, 40: INPUT
OMEG05ZR
IF OMEG05ZR = 0 THEN OMEG05ZR = VACCEP
LOCATE 16, 40: PRINT " "; LOCATE 16, 41: PRINT OMEG05ZR

LOCATE 16, 22
PRINT "OMEGA01 trans. "
VACCEP = OMEG01ZR: LOCATE 16, 41: PRINT OMEG01ZR: LOCATE 16, 40: INPUT
OMEG01ZR
IF OMEG01ZR = 0 THEN OMEG01ZR = VACCEP
LOCATE 16, 40: PRINT " "; LOCATE 16, 41: PRINT OMEG01ZR

LOCATE 16, 3: PRINT "Do you wish to continue
INPUT ANSW$: IF ANSW$ = "N" OR ANSW$ = "n" THEN END
LOCATE 16, 3: PRINT "
; Write initial setup variables, used in this experiment.

OPEN "A:\DACQS.INI" FOR OUTPUT AS #1
PRINT #1, TUBEN%
PRINT #1, RUNUMB%
PRINT #1, DATDISK%
PRINT #1, VCOUNT%
PRINT #1, SWEEP1
PRINT #1, SEMPLG
PRINT #1, TAMB
PRINT #1, PATTERN$
PRINT #1, WTFIT
PRINT #1, AIRZR
PRINT #1, WATZR
PRINT #1, CELESCZR
PRINT #1, OMEG15ZR
PRINT #1, OMEG05ZR
PRINT #1, OMEG01ZR
CLOSE #1
END SUB

;*****
SUB VOIDMEASURE STATIC
SUBR% = 23
; This routine steps the user thru the procedure of the local void
; fraction measurement. Moves the probe to the next position keeps
; record of file names stored on digital oscilloscope.
; For the 1 inch ID test section the local void fraction is measured

```

```

* at 9 positions, this gives a spacing of 0.1 inch.
* To cover all 9 positions requires stepping the probe 8 times.
*
J = 1
FOR I = 1 TO 9
 PPOS% = I
 *
 * Measure void fraction.
 LOCATE 20, 4: PRINT " Void fraction measurements. "
 LOCATE 21, 3: PRINT "
 LOCATE 22, 3: PRINT "
 *
 LOCATE 16, 3
 PRINT USING "Void fraction at probe position No. ##"; PPOS%
 LOCATE 23, 4
 IF I = 1 THEN
 PRINT " Store data on the scope and enter FILE No. "
 LOCATE 23, 50: INPUT FILEN%
 ELSE
 FILEN% = FILEN% + 2
 END IF
 *
 FLVOID$(J) = "WAVE" + STR$(FILEN%) + ".WFT"
 FLVOID$(J + 1) = "WAVE" + STR$(FILEN% + 1) + ".WFT"
 FLGNL$(I) = "SGN" + STR$(RUNUMB%) + " " + STR$(I) + ".WFT"
 LOCATE 23, 4
 PRINT " Store data on the scope and press any key to continue."
 LOCATE 23, 4: SLEEP
 LOCATE 23, 4
 PRINT "
 *
 LOCATE 5, 53: PRINT "Probe pos. File name:"
 LOCATE 6 + I, 56
 PRINT ; PPOS%; SPC(7); FLVOID$(J)
 *
 * Move the probe to the new position then press anykey.
 LOCATE 22, 4
 PRINT " Moving the probe to the next position, WAIT.."
 LOCATE 23, 10: PRINT " Remaining number of steps: "
 IF PPOS% > 8 THEN GOTO 888
 *
 CALL PROBE(PPOS%)
 *
 888 LOCATE 21, 3: PRINT "
 LOCATE 22, 3: PRINT "
 LOCATE 23, 3: PRINT "
 *
 J = J + 2
 NEXT I
*

END SUB
SUB WATFLOWRATE STATIC
 SUBR% = 24
 *
 * This subroutine calculates the water flow rate and superficial
 * velocity. The factory calibrated Paddlewheel sensor has a
 * output voltage linear to the water flow rate.
 * The slope of this linear relationship is: 1.0094 * 10^-3 (m**3/s)/(volt)
 * Entry variable:
 * WATFLOWV: Paddlewheel sensor voltage output. (Volts)
 * Returned variables:
 * WATFLOW: Water flow rate. (m**3/s)
 * WSUPVEL: Water superficial velocity. (m/s)
 * DIAMT: Test section inside diameter. (m)
 * AREAT: Test section tube cross-sectional area. (m**2)
 * DIAMP: Water inlet pipe, inside diameter. (m)
 * AREAP: Water inlet pipe, cross-sectional area. (m**2)
 *
 PI = 3.141593
 DIAMT = TUBEN% * .0254
 AREAT = PI * DIAMT ^ 2 / 4
 TEMP = ((WATTEM + MIXTEM) / 2) + 273.15
 ROWATER = 553.1 + 3.706 * TEMP - .009089 * TEMP ^ 2 + 5.52E-06 * TEMP ^ 3
 *
 * Calculate surface tension of the water at the measured temperature.
 * Equation represents a curve fit of data from Findlay's 9th ed.
 *
 WSTEN = 75.77 - .1551 * WATTEM
 *
 IF WTFIT = 1.5 THEN
 WATFLOW(RP%) = 1.0094 * 10 ^ -3 * (WATFLOWV(RP%) - WATZR)
 MWAT(RP%) = WATFLOW(RP%) * ROWATER
 GWAT(RP%) = MWAT(RP%) / AREAT
 WSUPVEL(RP%) = GWAT(RP%) / ROWATER
 * Water actual velocity, in 1.5 inch pipe.
 DIAMP = 1.5 * .0254
 AREAP = PI * DIAMP ^ 2 / 4
 END IF
 *
 IF WTFIT = .5 THEN
 WATFLOW(RP%) = .12253 * 10 ^ -3 * (WATFLOWV(RP%) - WATZR)
 MWAT(RP%) = WATFLOW(RP%) * ROWATER
 GWAT(RP%) = MWAT(RP%) / AREAT
 WSUPVEL(RP%) = GWAT(RP%) / ROWATER
 END IF
 *
END SUB

```

## **Data Reduction Programs**

**TPDATRED.FOR:** A Fortran code developed for the analysis of the experimental data. The program provides the option of processing one experimental test run or all the test runs at the same time.

```

PROGRAM PRESSURE
C *=====
C *
C * PRESSURE.FOR
C *
C * FLOW REGIME TRANSITION IN TWO-PHASE HORIZONTAL
C * GAS-LIQUID FLOW
C *
C * Written by: M.L. LAMARI
C * date: May 1996
C * Updated : Sept. 98
C *=====
C * This program is developed for the analysis of the
C * experimental data.
C *=====
C *
C * List of variables:
C Subroutines.
C
C CELESCO : Celeco transducer output to pressure (Psi and kPa).
C OMEGA15 : OMEGA15 transducer output to pressure (Psi and kPa).
C OMEGA05 : OMEGA05 transducer output to pressure (Psi and kPa).
C OMEGA01 : OMEGA01 transducer output to pressure (Psi and kPa).
C WATFLOWRATE: Convert voltage output to water flow rate.
C AIRFLOWRATE: Convert voltage output to air flow rate.
C OUTPUTFILE : Create an output file for the experimental data.
C PDROPICAL : Calculate the measured frictional pressure drop.
C
C List of variables:
C
C Note that average reading refers to the average of two measurements,
C taken at the beginning and the end of the each experimental test.
C
C AREAT : Test section cross-sectional area (m^2)
C AINPR : Air static pressure, at inlet pipe, average reading (kPa).
C AIRTEMP : Air inlet temperature (deg. C).
C AIRFLOW : Air flow rate, actual (m^3/s).
C ASUPVEL : Air superficial velocity, in test section, (m/s).
C AIRZR : Measured air velocity transducer zero reading (Volts).
C ASUPV: Air superficial velocity, average reading (m/s).
C AVTPR : Air static pressure, at vel. transducer, average reading (kPa).
C AIRSELY : Measured air superficial velocity (Volts).
C AIRTEM : Air inlet temperature, average of two readings (deg. C).
C AIRVPRSV : Air static pressure, at vel. transducer (Volts).
C AIRINPRV : Measured air static pressure, at inlet pipe (Volts).
C AVTPRS : Air static pressure, at vel. transducer (kPa).
C AINPRS : Air static pressure, at inlet pipe (kPa).
C ACTVEL : Air actual velocity, at inlet pipe (m/s).
C ANSW : Variable used thru the program, "Y" continue and "N" end execution.
C DIAMT : Test section diameter (m)
C DIFF(1..9) : Difference between Data set no. 1 and no. 2. Order Air,

```

```

C Water, pressure ports 1 to 4, then temp. (air, water and mixture)
C FLNAME : File name to store the experimental data.
C FLYOID : File name to store local void measurements.
C MIXINPR : Test section static pressure, at inlet, average reading (kPa).
C MIXINPRS : Test section static pressure, at inlet (kPa).
C MIXMDPR : Test section static pressure, at probe, average reading (kPa).
C MIXMDPRS : Test section static pressure, at probe (kPa).
C MIXEXPR : Test section static pressure, at exit, average reading (kPa).
C MIXEXPRS : Test section static pressure, at exit (kPa).
C MIXPRSV1 : Measured test section static pressure, at inlet (Volts).
C MIXPRSV2 : Measured test section static pressure, at probe (Volts).
C MIXPRSV3 : Measured test section static pressure, at exit (Volts).
C MIXTEMP : Mixture, air-water, exit temperature (deg. C).
C MIXTEM : Mixture exit temperature, average of two readings (deg. C).
C CELESCZR : Measured Celeco pressure transducer zero reading (Volts).
C OMEG15ZR : Measured OMEGA15 pressure transducer zero reading (Volts).
C OMEG05ZR : Measured OMEGA05 pressure transducer zero reading (Volts).
C OMEG01ZR : Measured OMEGA01 pressure transducer zero reading (Volts).
C
C PATTERN$: Structure of the air-water mixture, based on visual observations.
C PRSZR : Measured Selesco pressure transducer zero reading (Volts).
C PDIFF(1..9) : Array, percentage deviation assigned as in DIFF().
C RMSV(RP.1..6) : Array, rms values of each data set, assigned as in
C DIFF(), not for temperatures. (Volts)
C RP : Variable to indicate measurements taken at beginning (1), or end of experiment (2).
C TUBEN : Test tube number; 1 (1 in ID), 2 (2 in ID) and 3 (3 in ID).
C WATFLOW : Water flow rate, actual (m^3/s).
C WATFLOWV : Measured water flow rate, actual (Volts).
C WATINPRV : Measured water inlet static pressure (Volts).
C WSUPVEL : Water superficial velocity, in test section, (m/s).
C WACTVEL : Water velocity at inlet pipe (m/s).
C WATTEMP : Water inlet temperature (deg. C).
C WSTEN : Water surface tension, measured (mN/m).
C WINPR : Water inlet static pressure, average reading (kPa).
C WINPRS : Water inlet static pressure (kPa).
C WSUPV : Water superficial velocity, average reading. (m/s)
C WATZR : Measured paddle wheel sensor zero reading (Volts).
C WATTEM : Water inlet temperature, average of two readings (deg. C).
C WTFIT : Variable to identify fitting used with paddle wheel sensor.
C
C AIRCONST : Air gas constant (287 J/kg.k)
C GAIR : Air mass velocity (kg/m2.s)
C MAIR : Air mass flowrate (kg/s)
C GWAT : Water mass velocity (kg/m2.s)
C MWAT : Water mass flowrate (kg/s)
C ROAIRIN : Air density at test tube inlet (kg/m3)
C ROAIRMD : Air density at probe position (kg/m3)
C ROAIREX : Air density at test tube exit (kg/m3)
C ROAIRAV : Average air density (IN+EX)/2 (kg/m3)
C ROWATER : Water density (kg/m3)
C
C MG = mdot-gas (kg/s)

```



```

C MF = m-dot-liquid (kg/s)
C MT = m-dot-total (kg/s)
C GG = air mass flux (kg/m.m.s)
C GF = water mass flux (kg/m.m.s)
C GT = total mass flux (kg/m.m.s)
C JF = Water Superficial Velocity (m/s)
C JG = Air superficial Velocity (m/s)
C
C Dimension the arrays.
C
C DIMENSION PRESSAV(2),AIRSVELV(2),WATFLOWV(2),AIRVPRSV(2)
C DIMENSION MIXPRSV1(2),MIXPRSV2(2),MIXPRSV3(2)
C DIMENSION AIRTEMP(2),WATTEMP(2),MIXTEMP(2)
C DIMENSION TEMPC(2),TEMPK(2)
C DIMENSION AVTPRS(2),MIXINPRS(2),MIXMDPRS(2),MIXEXPRS(2)
C
C DIMENSION RHOA(2),MUAIR(2),RHOW(2),MUWAT(2),WSTEN(2)
C DIMENSION AIRG(2),AIRM(2),ASUPVEL(2)
C DIMENSION WATG(2),WATM(2),WSUPVEL(2),WATQ(2)
C DIMENSION DIFF(15),PDIFF(15)
C
C DIMENSION SETN(200),RUNT(200),WSAMPLE(200)
C DIMENSION WATFIT(200)
C DIMENSION VOIDMAV(200),VOIDM1(200),VOIDM2(200),VOIDDIF(200)
C
C INTEGER PRUN,RUNUMB,TUBEN,TRUNS,RUN,K
C INTEGER SETN(200),RUNT(200),WSAMPLE(200)
C REAL PI,G,L1,L2,LT
C REAL WFIT,DIAMT,AREAT
C REAL AIRZR,WATZR,CELESCZR,OMEG15ZR,OMEG05ZR,OMEG01ZR
C REAL WATTEM,AIRTEM,MIXTEM,AIRTDIF
C REAL AVTPR,MIXINPR,MIXMDPR,MIXEXPR
C REAL DELP1,DELP2,DELP3,DPGM1,DPGM2,DPGM3,DPGM
C REAL JG,JF,RHOG,RHOF,GG,GF,GT,MG,MF,MT,MUG,MUF,WST,TC,TK,QX
C REAL GAIRD,GWATD,MAIRD,MWATD,GAIRPD,GWATPD,MAIRPD,MWATPD
C REAL AIRSVELV,WATFLOWV,AIRVPRSV,MIXPRSV1
C REAL MIXPRSV2,MIXPRSV3,AIRTEMP,WATTEMP,MIXTEMP
C REAL PRESSAV,MIXINPRS,MIXMDPRS,MIXEXPRS
C REAL MUAIR,RHOW,MUWAT,WSTEN
C REAL VM1,VM2,VMAV
C
C REAL QM2FO,QM2GO
C
C REAL DPHMC,DPHMB,DPHMF,QHMC,F,QHMBF,QHMMF,QHMDG,QHMDG,QHMDG,QHMDG
C REAL MUHB,MUHC,MUHD,MUHF,MUHM
C
C CHARACTER*6 FILE1/'Prop-'/,FILE2/'Press-'/,FILE3/'Phi_f-'/
C CHARACTER*6 FILE4/'Phi_g-'/,FILE5/'Void-'/,FILE6/'Sup_v-'/
C CHARACTER*6 FILE7/'Pdiff-'/
C CHARACTER*2 FILEC
C CHARACTER*4 FTYP/'OUT/'
C CHARACTER*13 FOUT1,FOUT2,FOUT3,FOUT4,FOUT5,FOUT6,FOUT7

```

```

C CHARACTER*3 FLN1/'RUN/'
C CHARACTER*4 FLN4/'VOL/'
C CHARACTER*2 FLN2
C CHARACTER*3 FLN3
C CHARACTER*12 FLNAME
C CHARACTER*4 REGIME(100)
C
C Declare the common shared variables.
C
C COMMON /MAIN1/ TUBEN,DIAMT,AREAT,G,PI,L1,L2,LT,RUNUMB
C COMMON /SPEC1/ SETN,RUNT,REGIME,WSAMPLE
C COMMON /VOID1/ ALFH,ALFLM,ALFC
C COMMON /PRESS1/ DPHMD,DPHMM,DPLM,DPFRD
C COMMON /PRESS2/ DPGM1,DPGM2,DPGM3,DPGM
C COMMON /PRESS3/ DPHMC,DPHMB,DPHMF
C COMMON /PRESS4/ QM2FO,QM2GO
C
C COMMON /PHI_G/ X,XX,QM1G,QM2G,QM3G,QMAVG,QHMMG,QHMDG,QLMG,QFRDG
C COMMON /PHI_F/ QM1F,QM2F,QM3F,QMAVF,QHMMF,QHMDF,QLMF,QFRDF
C COMMON /PHI_G1/ QHMC,F,QHMBF,QHMF
C COMMON /PHI_F1/ QHMC,G,QHMBG,QHMF
C COMMON /MOMG_MU/ MUHB,MUHC,MUHD,MUHF,MUHM
C
C WRITE(*,*) 'Enter 1 to process a single RUN, 2 for all exp. runs:'
C READ(*,*) PRUN
C
C OPEN (UNIT=1, FILE='EXP_WAVY.DAT', STATUS='OLD')
C K=0
C LOOP
C K=K+1
C READ (1,*) SETN(K),RUNT(K),REGIME(K),WATFIT(K),WSAMPLE(K)
C WRITE (*,*) SETN(K),RUNT(K),REGIME(K),WATFIT(K),WSAMPLE(K)
C IF (SETN(K).EQ.999) THEN
C EXIT
C END IF
C END LOOP
C CLOSE (1)
C TRUNS=K-1
C
C IF (PRUN.EQ.1) THEN
C TRUNS = 1
C WRITE(*,*) 'Enter run number to be processed:'
C READ (*,*) RUNUMB
C WRITE(*,*) ''
C WRITE(*,*) 'Experimental Run Number to be Processed: ',RUNUMB
C
C OPEN (UNIT=2, FILE='FILEN.NAM')
C WRITE(2,100) RUNUMB
C CLOSE (2)

```

```

C
IF (RUNUMBLE.99) THEN
 OPEN (UNIT=1, FILE='FILE.NAM', STATUS='OLD')
 READ (1,101) FLN2
 CLOSE (1)
 FOUT1=FILE1//FLN2//FTYP
 FOUT2=FILE2//FLN2//FTYP
 FOUT3=FILE3//FLN2//FTYP
 FOUT4=FILE4//FLN2//FTYP
 FOUT5=FILE5//FLN2//FTYP
 FOUT6=FILE6//FLN2//FTYP
 FOUT7=FILE7//FLN2//FTYP
ELSE
 OPEN (UNIT=1, FILE='FILE.NAM', STATUS='OLD')
 READ (1,101) FLN3
 CLOSE (1)
 FOUT1=FILE1//FLN3//FTYP
 FOUT2=FILE2//FLN3//FTYP
 FOUT3=FILE3//FLN3//FTYP
 FOUT4=FILE4//FLN3//FTYP
 FOUT5=FILE5//FLN3//FTYP
 FOUT6=FILE6//FLN3//FTYP
 FOUT7=FILE7//FLN3//FTYP
END IF
C
ELSE
C
 WRITE(*,*)''
 WRITE(*,*) TRUNS,' Experimental Runs will be Processed: '
C
 WRITE(*,*) 'Please enter OUTPUT files Identifier !'
 WRITE(*,*) 'Enter TWO (2) characters only,- eg. "09" OR "EX" - '
 READ (5,2) FILEC
 WRITE(*,*)''
C
 FOUT1=FILE1//FILEC//FTYP
 FOUT2=FILE2//FILEC//FTYP
 FOUT3=FILE3//FILEC//FTYP
 FOUT4=FILE4//FILEC//FTYP
 FOUT5=FILE5//FILEC//FTYP
 FOUT6=FILE6//FILEC//FTYP
 FOUT7=FILE7//FILEC//FTYP
C
 WRITE(*,*) 'Please WAIT, PROGRAM Running !'
C
END IF
C
OPEN (3,FILE=FOUT1)
WRITE (3,*) 'Two-Phase Mixture Properties:'
WRITE (3,*) 'Set Run Reg ---TC--- ---TK--- ---RHOG---' ;
+ ' ---RHOF--- ---MUG--- ---MUF--- ---WST---'
WRITE (3,*) ' # ID, Deg.C Deg.K kg/m3 ' ;
C
+ ' kg/m3 Pa.sec Pa.sec N/m '
C
OPEN (4,FILE=FOUT2)
WRITE (4,*) 'Two-Phase Flow Pressure Gradients:'
WRITE (4,*) 'Set Run Reg ---DPGM1--- --DPGM2--- --DPGM3---' ;
+ ' ---DPGM--- --DPHMD--- --DPHMM--- --DPLM--- --DPFRD---'
WRITE (4,*) ' # ID, Pa/m Pa/m Pa/m ' ;
+ ' Pa/m Pa/m Pa/m Pa/m '
C
OPEN (6,FILE=FOUT3)
WRITE (6,*) 'Two-Phase Flow Multipliers'
WRITE (6,*) 'Set Run Reg L-M-param L-M-param T-PH-Mult ' ;
+ ' T-PH-Mult T-PH-Mult T-PH-Mult T-PH-Mult T-PH-Mult ' ;
+ ' T-PH-Mult T-PH-Mult'
WRITE (6,*) ' # ID, ---X--- --XX--- --QM1F---' ;
+ ' ---QM2F--- --QM3F--- --QMAVF--- --QHMMF--- --QHMDF--- --QLMF' ;
+ ' --- --QFRDF--'
C
OPEN (7,FILE=FOUT4)
WRITE (7,*) 'Two-Phase Flow Multipliers'
WRITE (7,*) 'Set Run Reg L-M-param L-M-param T-PH-Mult ' ;
+ ' T-PH-Mult T-PH-Mult T-PH-Mult T-PH-Mult T-PH-Mult ' ;
+ ' T-PH-Mult T-PH-Mult'
WRITE (7,*) ' # ID, ---X--- --XX--- --QM1G---' ;
+ ' ---QM2G--- --QM3G--- --QMAVG--- --QHMMG--- --QHMDG--- --QLMG' ;
+ ' --- --QFRDG--'
C
OPEN (8,FILE=FOUT5)
WRITE (8,*) 'Two-Phase Flow Void Fractions:'
WRITE (8,*) 'Set Run Reg VoidFrac VoidFrac VoidFrac ' ;
+ ' VoidFrac -Quality- VoidFrac VoidFrac VoidFrac'
WRITE (8,*) ' # ID, Meas-Up. Meas-Dn. Meas-avg ' ;
+ ' Rel-Diff -QX-frac- -Homog.- LM-model CISE-cor'
C
OPEN (9,FILE=FOUT6)
WRITE (9,*) 'Phases Superficial and Mass Velocities:'
WRITE (9,*) 'Set Run Reg W-Sup_Vel G-Sup_Vel W-MassVel ' ;
+ ' G-MassVel Wat-M_Dot Gas-M_Dot'
WRITE (9,*) ' # ID, m/sec m/sec kg/s.m2 ' ;
+ ' kg/s.m2 kg/sec kg/sec '
C
OPEN (10,FILE=FOUT7)
WRITE (10,*) 'Exp. Readings, Relative Differences:'
WRITE (10,*) 'Set Run Reg G-Sup_Vel W-Sup_Vel AVT-Press ' ;
+ ' MixIN_PRS MixMD_PRS MixEX_PRS Air-Temp, Wat-Temp. Mix-Temp.'
WRITE (10,*) ' # ID, % % % % % ' ;
+ ' % % % % % % % % % % % '
C
OPEN (UNIT=11, FILE='HOMOG_DP.OUT')
WRITE (11,*) 'Homogeneous Model - Pressure Gradient'
WRITE (11,*) 'Run Beattie Cicchitti Dukler Owens ' ;
+ ' McAdams'

```

```

C WRITE(11,*)'# DPHMB DPHMC DPHMD DPHMF';
 +DPHMM';
C OPEN (UNIT=12, FILE='HOMOG_QF.OUT')
 WRITE(12,*)'Homogeneous Model - 2 Phase Multiplier - Phi_F'
 WRITE(12,*)'Run Beattie Cicchitti Dukler Owens';
 +McAdams';
 WRITE(12,*)'# QHMBF QHMBF QHMBF QHMBF';
 +QHMMF';
C OPEN (UNIT=13, FILE='HOMOG_QG.OUT')
 WRITE(13,*)'Homogeneous Model - 2 Phase Multiplier - Phi_G'
 WRITE(13,*)'Run Beattie Cicchitti Dukler Owens';
 +McAdams';
 WRITE(13,*)'# QHMBG QHMBG QHMBG QHMBG';
 +QHMMG';
C OPEN (UNIT=14, FILE='HOMOG_MU.OUT')
 WRITE(14,*)'Homogeneous Model - Mixture Viscosity.'
 WRITE(14,*)'Run Quality Beattie Cicchitti Dukler Owens';
 +McAdams';
 WRITE(14,*)'# QX MUHB MUHC MUHD MUHF';
 +MUHM';
C OPEN (UNIT=15, FILE='PHIM2_O.OUT')
 WRITE(15,*)'Two_phase multiplier wrt Total mass of mixture.'
 WRITE(15,*)'Run L-M_Param Measured_2 Measured_2'
 WRITE(15,*)'# XX PH12_M2_to PH12_M2_Go';
C C Define some of the parameters.
C
C TUBEN = 1 ! Test Tube Number
 PI = 3.141593
 G = 9.81 ! grav. accel. (m**m/s)
 L1 = 10.53 ! tube length-1 (m)
 L2 = 3.37 ! tube length-2 (m)
 LT = 13.9 ! tube total leng. (m)
C
C Cross-sectional area of the test section. (m**2)
C
C DIAMT = TUBEN*0.0254 ! 1" ID Test section
 AREAT = (PI*DIAMT**2)/4 ! Area of Test Section
C
C Read DACQS Output Voltage File: A:RUN+#.VOL
C Repeat for all test runs: Number of runs TRUNS.
C
C WRITE(*,*)'----- Output Files -----'
C
C #####
C #####
C DO 30 N = 1, TRUNS

```

```

C IF (PRUN.EQ.1) THEN
 DO 31 J=1, K
 IF (RUNT(J).EQ.RUNUMB) THEN
 REGIME(N)=REGIME(J)
 SETN(N)=SETN(J)
 WATFIT(N)=WATFIT(J)
 WSAMPLE(N)=WSAMPLE(J)
 GOTO 32
 END IF
 31 CONTINUE
 END IF
C RUNUMB = RUNT(N)
 32 WFIT = WATFIT(N)
C
 OPEN (UNIT=2, FILE='FILE.NAM')
 WRITE(2,100) RUNUMB
 CLOSE (2)
C
 IF (RUNUMB.LE.99) THEN
 OPEN (UNIT=1, FILE='FILE.NAM', STATUS='OLD')
 READ (1,101) FLN2
 CLOSE (1)
 FLNAME=FLN1//FLN2//FLN4
 ELSE
 OPEN (UNIT=1, FILE='FILE.NAM', STATUS='OLD')
 READ (1,101) FLN3
 CLOSE (1)
 FLNAME=FLN1//FLN3//FLN4
 END IF
C
 OPEN (UNIT=1, FILE=FLNAME, STATUS='OLD')
 WRITE(*,*)'Reading Input from: ', FLNAME
C
 READ (1,*) RUNUMBW
 READ (1,*) AIRZR,WATZR,CELESCZR,OMEG15ZR,OMEG05ZR,OMEG01ZR
 READ (1,*) AIRSVLV(1),WATFLOWV(1),AIRVPRSV(1),MIXPRSV1(1),
 + MIXPRSV2(1),MIXPRSV3(1),AIRTEMP(1),WATTEMP(1),MIXTEMP(1)
 READ (1,*) AIRSVLV(2),WATFLOWV(2),AIRVPRSV(2),MIXPRSV1(2),
 + MIXPRSV2(2),MIXPRSV3(2),AIRTEMP(2),WATTEMP(2),MIXTEMP(2)
C
 IF (RUNUMBW.NE.RUNUMB) THEN
 WRITE (*,*) 'Run number and file identifier are NOT the same'
 END IF
 CLOSE (1)
C
C Calculate average values of the temperatures.
C
 AIRTEM = (AIRTEMP(1) + AIRTEMP(2)) / 2
 WATTEM = (WATTEMP(1) + WATTEMP(2)) / 2
 MIXTEM = (MIXTEMP(1) + MIXTEMP(2)) / 2

```

```

C
C Calculate the average temperature: ???
C The average temperature is calculated as the arithmetic average
C of temp. of water at inlet and temp. of the two-phase
C mixture at the outlet of the test section.
C
C AIRTDIF = AIRTEM - MIXTEM
IF (AIRTDIF.GT.10.0) WRITE (*,*) 'Check measured temperatures!!'
C
C
C DO 35 I = 1, 2
C
C TEMPC (I) = (WATTEMP(I) + MIXTEMP(I)) / 2
C TEMPK (I) = TEMPC (I) + 273.15
C
C CALL CELESCO(CELESCZR,AIRVPRSV(I),AVTPRS(I))
C CALL OMEGA15(OMEGA15ZR,MIXPRSV1(I),MIXINPRS(I),RUNUMB)
C CALL OMEGA05(OMEGA05ZR,MIXPRSV2(I),MIXMDPRS(I),RUNUMB)
C CALL OMEGA01(OMEGA01ZR,MIXPRSV3(I),MIXEXPRS(I),RUNUMB)
C
C Calculte the average pressure of the test section (inlet - outlet).
C
C PRESSAV(I) = (MIXINPRS(I) + MIXEXPRS(I)) / 2
C
C Calculate some of the phases proporties.
C
C CALL PROPERTIES (TEMPK(I),PRESSAV(I),RHOA(I),MUAIR(I),RHOW(I),
C + MUWAT(I),WSTEN(I))
C
C CALL AIRFLOWRATE (AIRZR,AIRSVLV(I),RHOA(I),AIRM(I),
C + ASUPVEL(I))
C CALL WATFLOWRATE (WTFIT,RHOW(I),WATFLOWV(I),WATZR,WATQ(I),
C + WATM(I),WSUPVEL(I))
C
C AIRG(I) = AIRM(I) / AREAT
C WATG(I) = WATM(I) / AREAT
C
C 35 CONTINUE
C
C Calculate average of experimental readings; begining and end of
C experimental test.
C
C JG = (ASUPVEL(1) + ASUPVEL(2)) / 2
C JF = (WSUPVEL(1) + WSUPVEL(2)) / 2
C RHOG = (RHOA(1) + RHOA(2)) / 2
C RHOF = (RHOW(1) + RHOW(2)) / 2
C
C AVTPR = (AVTPRS(1) + AVTPRS(2))/2
C MIXINPR = (MIXINPRS(1) + MIXINPRS(2)) / 2
C MIXMDPR = (MIXMDPRS(1) + MIXMDPRS(2)) / 2
C MIXEXPR = (MIXEXPRS(1) + MIXEXPRS(2)) / 2
C
C Measured Pressure drop.
C
C DELP1 = (MIXINPR - MIXEXPR) * 1000 ! Pres. drop L1
C DELP2 = (MIXINPR - MIXMDPR) * 1000 ! Pres. drop L1
C DELP3 = (MIXMDPR - MIXEXPR) * 1000 ! Pres. drop L2
C
C Measured pressure gradient.
C
C DPGM1 = DELP1 / LT
C DPGM2 = DELP2 / L1
C DPGM3 = DELP3 / L2
C DPGM = (DPGM1+DPGM2+DPGM3) / 3
C
C Calculate average mass vel. and mass flow rates for air and water.
C
C GG = (AIRG(1) + AIRG(2)) / 2
C GF = (WATG(1) + WATG(2)) / 2
C MG = (AIRM(1) + AIRM(2)) / 2
C MF = (WATM(1) + WATM(2)) / 2
C
C MUG = (MUAIR(1) + MUAIR(2)) / 2
C MUF = (MUWAT(1) + MUWAT(2)) / 2
C WST = (WSTEN(1) + WSTEN(2)) / 2
C
C TC = (TEMPC(1) + TEMPC(2)) / 2
C TK = TC + 273.15
C
C Calculate diff. between the two data sets.
C
C GAIRD = ABS(AIRG(1) - AIRG(2))
C GWATD = ABS(WATG(1) - WATG(2))
C MAIRD = ABS(AIRM(1) - AIRM(2))
C MWATD = ABS(WATM(1) - WATM(2))
C
C Calculate standard deviation for air and water act. vel. and
C flow rates.
C
C IF (GG.NE.0.) GAIRPD=((GAIRD/GG)*100)
C IF (GF.NE.0.) GWATPD=((GWATD/GF)*100)
C IF (MG.NE.0.) MAIRPD=((MAIRD/MG)*100)
C IF (MF.NE.0.) MWATPD=((MWATD/MF)*100)
C
C Calculate diff. between data set 1 and 2.
C
C DIFF(1) = ABS(ASUPVEL(1) - ASUPVEL(2))
C DIFF(2) = ABS(WSUPVEL(1) - WSUPVEL(2))
C DIFF(3) = ABS(AVTPRS(1) - AVTPRS(2))
C DIFF(4) = ABS(MIXINPRS(1) - MIXINPRS(2))
C DIFF(5) = ABS(MIXMDPRS(1) - MIXMDPRS(2))
C DIFF(6) = ABS(MIXEXPRS(1) - MIXEXPRS(2))
C DIFF(7) = ABS(AIRTEMP(1) - AIRTEMP(2))

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DIFF(8) = ABS(WATTEMP(1) - WATTEMP(2))
DIFF(9) = ABS(MIXTEMP(1) - MIXTEMP(2))

C
DO 40 J = 1, 10
 PDIFF(J) = 0.
40 CONTINUE
C
C Calculate std dev.
C
IF (JG.NE.0.) PDIFF(1) = ((DIFF(1)/JG)**100)
IF (JF.NE.0.) PDIFF(2) = ((DIFF(2)/JF)**100)
IF (AVTPR.NE.0.) PDIFF(3) = ((DIFF(3)/AVTPR)**100)
IF (MIXINPR.NE.0.) PDIFF(4) = ((DIFF(4)/MIXINPR)**100)
IF (MIXMDPR.NE.0.) PDIFF(5) = ((DIFF(5)/MIXMDPR)**100)
IF (MIXEXPR.NE.0.) PDIFF(6) = ((DIFF(6)/MIXEXPR)**100)
IF (AIRTEM.NE.0.) PDIFF(7) = ((DIFF(7)/AIRTEM)**100)
IF (WATTEM.NE.0.) PDIFF(8) = ((DIFF(8)/WATTEM)**100)
IF (MIXTEM.NE.0.) PDIFF(9) = ((DIFF(9)/MIXTEM)**100)

C
C =====
C Calculate the Mixture quality.
C
MT = MG + MF
GT = GG + GF
QX = MG / (MG + MF)
IF (QX.LT.0.) THEN QX = 0.0000001

C
C =====
C Read the measured average void fraction from "VOID_EXP.dat".
C
OPEN (UNIT=1, FILE=VOID_WVY.DAT, STATUS='OLD')

C
LOOP
READ (1,*) RUN,VM1,VM2,VMAV
IF (RUN.EQ.RUNUMB) THEN
 VOIDM1(N) = VM1
 VOIDM2(N) = VM2
 VOIDMAV(N) = VMAV
 IF (VMAV.NE.0.0) THEN
 VOIDDIF(N) = (VM1-VM2)/VMAV
 ELSE
 VOIDDIF(N) = 0.0
 END IF
 EXIT
END IF
END LOOP
CLOSE (1)

C
CALL EMPIRICAL(RHOG,RHOF,QX,GG,GF,GT,MUG,MUF,WST)

C
C =====
C

```

```

C Two-Phase Mixture Properties: "Prop-xx.out".
C
WRITE(3,111) SETN(N),RUNUMB,REGIME(N),TC,TK,RHOG,RHOF,MUG,MUF,WST
111 FORMAT(I3,2X,I3,2X,A3,4(2X,F8.3),2(2X,E9.3),2X,F8.4)
C
C Two-Phase Flow Pressure Gradients: "Press-xx.out".
C
WRITE(4,112) SETN(N),RUNUMB,REGIME(N),DPGM1,DPGM2,DPGM3,DPGM,
+ DPHMD,DPHMM,DPLM,DPFRD
112 FORMAT(I3,2X,I3,2X,A3,8(2X,F9.3))
C
C Two-Phase Flow Multipliers: "Phi_f-xx.out".
C
WRITE(6,113) SETN(N),RUNUMB,REGIME(N),X,XX,QM1F,QM2F,QM3F,QMAVF,
+ QHMMF,QHMDF,QLMF,QFRDF
113 FORMAT(I3,2X,I3,2X,A3,10(2X,E9.3))
C
C Two-Phase Flow Multipliers: "Phi_g-xx.out".
C
WRITE(7,114) SETN(N),RUNUMB,REGIME(N),X,XX,QM1G,QM2G,QM3G,QMAVG,
+ QHMMG,QHMDG,QLMG,QFRDG
114 FORMAT(I3,2X,I3,2X,A3,10(2X,E9.3))
C
C Two-Phase Flow Void Fractions: "Void-xx.out".
C
WRITE(8,115) SETN(N),RUNT(N),REGIME(N),VOIDM1(N),VOIDM2(N),
+ VOIDMAV(N),VOIDDIF(N),QX,ALFH,ALFLM,ALFC
115 FORMAT(I3,2X,I3,2X,A3,4(2X,F8.4),2X,E9.3,3(2X,F8.4))
C
C Phases Superficial and Mass Velocities: "Sup_v-xx.out".
C
WRITE(9,116) SETN(N),RUNT(N),REGIME(N),JF,JG,GF,GG,MF,MG
116 FORMAT(I3,2X,I3,2X,A3,2(2X,F9.4),4(2X,E9.3))
C
C Phases Superficial and Mass Velocities: "Pdiff-xx.out".
C
WRITE(10,117) SETN(N),RUNT(N),REGIME(N),PDIFF(1),PDIFF(2),
+ PDIFF(3),PDIFF(4),PDIFF(5),PDIFF(6),PDIFF(7),PDIFF(8),PDIFF(9)
117 FORMAT(I3,2X,I3,2X,A3,9(2X,F9.3))
C
C
WRITE(11,221) RUNUMB,DPHMB,DPHMC,DPHMD,DPHMF,DPHMM
WRITE(12,222) RUNUMB,QHMBF,QHMCF,QHMDF,QHMMF,QHMMF
WRITE(13,222) RUNUMB,QHMBG,QHMCB,QHMDG,QHMMG,QHMMG
WRITE(14,223) RUNUMB,QX,MUHB,MUHC,MUHD,MUHF,MUHM
WRITE(15,224) RUNUMB,XX,QM2FO,QM2GO
C
221 FORMAT(I3,5(2X,F9.3))
222 FORMAT(I3,5(2X,E9.3))
223 FORMAT(I3,2X,E9.3,5(2X,E9.3))
224 FORMAT(I3,3(2X,E9.3))
C

```

```

CALL STRATTAU(RHOG,RHOF,QX,GG,GF,MUG,MUF,JG,JF,VMAV)
C
C Repeat for the test run (30 CONTINUE)
C
30 CONTINUE
C
C #####
C
CLOSE(3)
CLOSE(4)
CLOSE(6)
CLOSE(7)
CLOSE(8)
CLOSE(9)
CLOSE(10)
CLOSE(11)
CLOSE(12)
CLOSE(13)
CLOSE(14)
CLOSE(15)
CLOSE(18)
CLOSE(19)
C
1 FORMAT (A12)
2 FORMAT (A2)
100 FORMAT (I3)
101 FORMAT (A3)
C
C End of Post-processor program.
C
WRITE(*,*) '----- Run Complete !! -----'
READ(*,*) PASS
C
PAUSE
STOP
END
C
*-----
SUBROUTINE EMPIRICAL(RHOG,RHOF,QX,GG,GF,GT,MUG,MUF,WST)
*-----
C
C Pressure Drop calculation routine
C
REAL RHOG,RHOF,QX,GG,GF,GT,MUG,MUF,WST
REAL MUHM,MUHD,RHOH,ALFH,MUHB,L1,L2,LT
REAL MUHC,MUHF
REAL QLMF,QLMG,QHMMF,QHMMG,QHMDF,QHMDG,QFRDF,QFRDG
REAL QM1F,QM1G,QM2F,QM2G,QM3F,QM3G,QMAVF,QMAVG
REAL DPGO,DPFO,QM2FO,QM2GO
INTEGER RUNUMB
C
COMMON /MAIN1/ TUBEN,DIAMT,AREAT,G,PI,L1,L2,LT,RUNUMB
COMMON /VOID1/ ALFH,ALFLM,ALFC
COMMON /PRESS1/ DPHMD,DPHMM,DPLM,DPFRD
COMMON /PRESS2/ DPGM1,DPGM2,DPGM3,DPGM
COMMON /PRESS3/ DPHMC,DPHMB,DPHMF
COMMON /PRESS4/ QM2FO,QM2GO
C
COMMON /PHI_G/ X,XX,QM1G,QM2G,QM3G,QMAVG,QHMMG,QHMDG,QLMG,QFRDG
COMMON /PHI_F/ QM1F,QM2F,QM3F,QMAVF,QHMMF,QHMDF,QLMF,QFRDF
C
COMMON /PHI_G1/ QHMCF,QHMBF,QHMF
COMMON /PHI_F1/ QHMCB,QHMBG,QHMBF
COMMON /MOMG_MU/ MUHB,MUHC,MUHD,MUHF,MUHM
C
ALFH=1/((1-QX)/QX*(RHOG/RHOF)) ! Collier 3Ed. eq. (2.26)
RHOH=1/((QX/RHOG)*((1-QX)/RHOF)) ! Collier 3Ed. eq. (2.23)
C
Homogeneous density, as a function of ALFH, same as above.
C
RHOH=(1-ALFH)*RHOH+ALFH*RHOG
C
McAdams et al. (1942)
C
MUHM=1/(((QX/MUG)+(1-QX)/MUF)) ! Collier 3Ed. eq. (2.38)
C
Dukler et al. (1964), from Whalley, 1987
C
MUHD=(MUG*QX*RHOH/RHOG)+(MUF*(1-QX)*RHOH/RHOF) ! Collier 3Ed. eq. (2.40)
C
Substitute Eq. (2.26) gives the following.
C
MUHD=(1-ALFH)*MUF+ALFH*MUG
C
Beattie and Whalley (1981), =
C
MUHB=MUG*ALFH+MUF*(1-ALFH)*(1+2.5*ALFH) !Walley eq. (5.15)
C
Cicchitti et al. (1960) from Collier Eq. (2.39)
C
MUHC=QX*MUG+(1-QX)*MUF
C
Owens (1962) from Hewitt and Taylor (1981)
C
MUHF=MUF
C
OPEN (UNIT=1, FILE='MU_BDMCF.DAT', STATUS='OLD')
WRITE(1,222) RUNUMB,MUHB,MUHD,MUHM,MUHC,MUHF
CLOSE (1)
C
Using Dukler eq. for Homogeneous viscosity.
C
RETPD = GT * DIAMT / MUHD
C
IF (RETPD.LT.2400) THEN

```

```

 FTPD = 64 / RETPD
 ELSE
 FTPD = 4*(0.046*RETPD**(-0.2))
 END IF
 C
 C DPHMD = FTPD*GT**2/(2*RHOH*DIAMT)
 C
 C Using McAdams eq. for Homogeneous viscosity.
 C
 RETPM = GT * DIAMT / MUHM
 C
 IF (RETPM.LT.2400) THEN
 FTPM = 64 / RETPM
 ELSE
 FTPM = 4*(0.046*RETPM**(-0.2))
 END IF
 C
 DPHMM = FTPM*GT**2/(2*RHOH*DIAMT)
 C
 C Using Cicchitti eq. for Homogeneous viscosity.
 C
 RETPC = GT * DIAMT / MUHC
 C
 IF (RETPC.LT.2400) THEN
 FTPC = 64 / RETPC
 ELSE
 FTPC = 4*(0.046*RETPC**(-0.2))
 END IF
 C
 DPHMC = FTPC*GT**2/(2*RHOH*DIAMT)
 C
 C Using Beattie & Whalley eq. for Homogeneous viscosity.
 C
 RETPB = GT * DIAMT / MUHB
 C
 IF (RETPB.LT.2400) THEN
 FTPB = 64 / RETPB
 ELSE
 FTPB = 4*(0.046*RETPB**(-0.2))
 END IF
 C
 DPHMB = FTPB*GT**2/(2*RHOH*DIAMT)
 C
 C Using Owens eq. for Homogeneous viscosity.
 C
 RETPF = GT * DIAMT / MUHF
 C
 IF (RETPF.LT.2400) THEN
 FTPF = 64 / RETPF
 ELSE
 FTPF = 4*(0.046*RETPF**(-0.2))
 END IF
 C
 C DPHMF = FTPF*GT**2/(2*RHOH*DIAMT)
 C
 C ---
 C
 C Lokhart-Martenelli Model
 C
 C Isothermal, no vaporization, no condensation, thus momentum
 C
 C pressure drop is zero.
 C
 REF = GF * DIAMT / MUJF
 IF (REF.LT.2400) THEN
 FF = 64 / REF
 C1 = 5
 C2 = 12
 ELSE
 FF = 4 * (.046 * REF**(-.2))
 C1 = 10
 C2 = 20
 END IF
 C
 DPF = FF * GF**2 / (2 * RHOF * DIAMT)
 C
 REG = GG * DIAMT / MUG
 IF (REG.LT.2400) THEN
 FG = 64 / REG
 C = C1
 ELSE
 FG = 4 * (.046 * REG**(-.2))
 C = C2
 END IF
 C
 DPG = FG * GG**2 / (2 * RHOG * DIAMT)
 C
 XX = DPF / DPG
 IF (XX.LT.0.) THEN XX = 0.0000001
 X = SQRT(XX)
 C
 PHIF = 1 + (C / X) + (1 / XX)
 PHIG = 1 + C * X + XX
 C
 ALFLM = (SQRT(PHIF) - 1) / SQRT(PHIG)
 C
 ALFLM = 1 / (1 + X**.8)**.378
 C
 DPLM = PHIF * DPF
 DPLM2 = PHIG * DPG
 C
 C Friedel correlation for frictional two-phase pressure gradient.
 C
 C Whalley 1987, PP 274.
 C
 REFO = GT * DIAMT / MUJF
 IF (REFO.LT.2400) THEN

```

```

 FFO = 64 / REFO
 ELSE
 FFO = 4 * (.046 * REFO**(-.2))
 END IF
 DPFO=FFO*GT**2/(2*RHO*DIAMT)
 C
 REGO = GT * DIAMT / MUG
 IF (REGO.LT.2400) THEN
 FGO = 64 / REGO
 ELSE
 FGO = 4*(0.046*REGO**(-0.2))
 END IF
 DPGO =FGO*GT**2/(2*RHO*DIAMT)
 C
 WST = .071
 C
 WE = (GT**2*DIAMT)/(WST*RHOH)
 FR = GT**2/(G*DIAMT*RHOH**2)
 H = (RHOH/RHO)*.91*(MUG/MUF)**.19*(1-MUG/MUF)**.7
 F = QX**0.78*(1-QX)**0.224
 E=(1-QX)**2*QX**2*(RHO*FGO)/(RHO*FFO)
 PHIFR=E+(3.24*F*H)/(FR**0.045*WE**0.035)
 C
 DPF RD = DPFO * PHIFR
 C
 Pressure Drop Multipliers:
 C
 QLMF = DPLM / DPF
 QLMG = DPLM / DPG
 QHMMF = DPHMM / DPF
 QHMMG = DPHMM / DPG
 QHMDf = DPHMD / DPF
 QHMDG = DPHMD / DPG
 QFRDF = DPF RD / DPF
 QFRDG = DPF RD / DPG
 QHMCf = DPHMC / DPF
 QHMCg = DPHMC / DPG
 QHMBf = DPHMB / DPF
 QHMBg = DPHMB / DPG
 QHMFg = DPHMF / DPG
 QHMFf = DPHMF / DPF
 C
 QM1F = DPGM1 / DPF
 QM1G = DPGM1 / DPG
 QM2F = DPGM2 / DPF
 QM2G = DPGM2 / DPG
 QM3F = DPGM3 / DPF
 QM3G = DPGM3 / DPG
 QMAVf = DPGM / DPF
 QMAVG = DPGM / DPG
 C
 QM2FO = DPGM2 / DPFO
 QM2GO = DPGM2 / DPGO
 C
 OPEN (UNIT=1, FILE='HOMOG_DP.OUT', STATUS='OLD')
 WRITE(1,222) RUNUMB,DPHMM,DPHMC,DPHMB,DPHMF
 CLOSE (1)
 C
 OPEN (UNIT=1, FILE='HOMOG_QF.OUT', STATUS='OLD')
 WRITE(1,222) RUNUMB,QHMMF,QHMDf,QHMCf,QHMBf,QHMFf
 CLOSE (1)
 C
 OPEN (UNIT=1, FILE='HOMOG_QG.OUT', STATUS='OLD')
 WRITE(1,222) RUNUMB,QHMMG,QHMDG,QHMCg,QHMBG,QHMFg
 CLOSE (1)
 C
 OPEN "SET7C-QG.OUT" FOR APPEND AS #1
 PRINT #1,RUNUMB,X,XX,QM1G,QM2G,QMAVG,QHMMG,QHMDG,QLMG,QFRDG
 CLOSE (1)
 C
 CISE Correlation for void fraction; Premoli et al. (1970)
 Whalley (1987), PP 273
 C
 WE1 =GT*GT*DIAMT/(WST*RHO*F)
 REFO = GT * DIAMT/MUF
 E2 =0.0273*WE1*REFO**(-.51)*(RHOH/RHO)*(-.08)
 E1 =1.578*REFO**(-.19)*(RHOH/RHO)*(-.22)
 BT =(RHO*QX)/(RHO*QX+(RHO*(1-QX)))
 Y =BT/(1-BT)
 S =1+E1*(Y/(1+Y*E2)-Y*E2)**(.5)
 ALFC =(1+(S*(1-QX)*RHO/(QX*RHO)))**(-1)
 C
 C 222 FORMAT(I3,5(2X,F9.3))
 C
 RETURN
 C
 END
 C
 *=====
 SUBROUTINE PROPERTIES(TEMPK,PRESSAV,ROAIRAV,MUG,
 + ROWATER,MUF,WSTEN)
 *=====
 C
 Calculate surface tension of the water at the measured temperature.
 Equation represents a curve fit of data from Findlay's 9th ed.
 C
 REAL TEMPK,PRESSAV,ROAIRAV,MUG,ROWATER,MUF,WSTEN
 REAL STDPRES,STDTEMP,AIRCONST,TREF,MUREF,S
 C
 STDPRES = 101.33 ! kPa
 STDTEMP = 20. ! deg. C
 AIRCONST = 287. ! Air Gas Constant J/kg.K
 C
 Gas Phase !
 C
 Calculate the density of the gas phase at the average measured

```



```

C pressure and temperature.
C
C ROAIRAV=(PRESSAV+101.33)*1000/(AIRCONST*TEMPK)
C ===
C MUG : air dyn. viscosity (Pa.s)
C
C MUG Calculated using the Sutherland (1893) approximation;
C From F.M. White, Viscous Fluid Flow, 2nd, pp. 28.
C
C TREF = 273.
C MUREF = 1.716E-05
C S = 111.
C ! For Air.
C
C MUG=MUREF*(TEMPK/TREF)**(1.5)*(TREF+S)/(TEMPK+S)
C ===
C Liquid Phase !
C
C ROWATER=553.1+3.706*TEMPK-0.009089*TEMPK**2+5.52E-06*TEMPK**3
C
C WSTEN = 75.77-.1551*(TEMPK-273.15)
C WSTEN = WSTEN*.001
C ! mN/m
C ! N/m
C
C MUF: water dyn. viscosity (Pa.s)
C
C MUF=.04405-.0003315*TEMPK+8.366E-07*TEMPK**2-7.036E-10*TEMPK**3
C
C RETURN
C END
C
C *=====
C SUBROUTINE AIRFLOWRATE(AIRZR,AIRSVELV,ROAIRAV,AIRM,
C + ASUPVEL)
C *=====
C
C REAL AIRZR,AIRSVELV,ROAIRAV,AIRG,AIRM,ASUPVEL
C REAL AIRGN
C
C COMMON /MAIN1/ TUBEN,DIAMT,AREAT,G,PI,L1,L2,LT,RUNUMB
C INTEGER RUNUMB
C
C AVTPRSA = AVTPRS(RP) + 101.33
C ! kPa absolute.
C AVTDELV = AIRSVELV(RP) - AIRZR
C ASVEL = AVTDELV * 10.05
C
C AVTDELV = AIRSVELV - AIRZR
C
C Air mass velocity and mass flow rate.
C
C IF (AVTDELV.LT.0.5) THEN
C AIRG=AVTDELV**13.613
C ELSE
C AIRG=AVTDELV**23.449
C END IF
C
C New calibration, AVT linearizer defective, see Ph.D. 5-p.77.
C
C IF(RUNUMB.GT.64) THEN
C AIRGN = 1.3104 * AVTDELV**2 + 0.3045
C AIRG = AIRGN**(1./0.75)
C END IF
C
C IF (AVTDELV.LT.0.5) THEN
C AIRG=AIRG*13.613/23.449
C END IF
C
C END IF
C
C AIRM = AIRG * AREAT
C
C Air superficial velocity (m/sec)
C
C ASUPVEL = AIRM / (ROAIRAV*AREAT)
C
C Return to calling program.
C
C RETURN
C END
C
C *=====
C SUBROUTINE WATFLOWRATE (WTFIT,ROWATER,WATVOLT,WATZR,WATQ,
C + WATM,WATSUPV)
C *=====
C
C This subroutine calculates the water flow rate and superficial
C velocity. The factory calibrated Paddlewheel sensor has a
C output voltage linear to the water flow rate.
C The slope of this linear relationship is:1.0094*10^-3(m**3/s)/(V)
C
C REAL WTFIT,ROWATER,WATVOLT,WATZR,WATQ,WATM,WATSUPV
C REAL AREAT
C
C COMMON /MAIN1/ TUBEN,DIAMT,AREAT,G,PI,L1,L2,LT,RUNUMB
C INTEGER RUNUMB
C
C IF (WTFIT.EQ.1.5) THEN
C WATQ=1.0094E-03*(WATVOLT-WATZR)
C WATM=WATQ*ROWATER
C WATSUPV=WATM/(ROWATER*AREAT)
C END IF
C
C IF (WTFIT.EQ.0.5) THEN
C WATQ=0.12253E-03*(WATVOLT-WATZR)
C IF(RUNUMB.GT.64) THEN
C WATQ=0.11875E-03*(WATVOLT-WATZR)

```

```

END IF
WATM=WATQ*ROWATER
WATSUPV=WATM/(ROWATER*AREAT)
ELSE
 WRITE(*,*) 'Check Value of WFIT: Paddlewheel Sensor fitting-'
 WRITE(*,*) 'Should be equal to 0.5 or 1.5'
 WRITE(*,*) 'Value used is: ', WFIT
END IF
C

RETURN
END
C *=====
SUBROUTINE CELESCO (CELESCZR,CELESCOV,PRSKPA)
C *=====
C
C From calibration certificate, the transducer's voltage is
C 0.2975 mV/V.psi.
C With 15 VDC power supply, the transducer's slope becomes equal to
C 4.476 mV/psi .
C The pressure transducer recalibrated, the new slope is
C 16.718 psi/Volts, pp 56 Ph.D. 2.
C
REAL PRSKPA,CELESCOV,CELESCZR,PRSPSI
C
PRSPSI=((CELESCOV-CELESCZR)*16.718) / Psi
PRSKPA=PRSPSI*6.8948
C
RETURN
END
C *=====
SUBROUTINE OMEGA01 (OMEG01ZR,OMEGAV,PRSKPA,RUNUMB)
C *=====
C
C This subroutine is used to calculate the pressure when OMEGA1
C transducer is used. OMEGA01 model PX136; (0 to 1 Psig Range)
C From calibration, see Ph.D. 4 p. 31, the transducer's slope
C is 1.449 kPa/Volts. (of amplified output voltage).
C
C The OMEGA01 pressure transducer was calibrated Dec. '95
C
REAL PRSKPA,OMEGAV,OMEG01ZR,PRSPSI
INTEGER RUNUMB
C
PRSKPA=(OMEGAV-OMEG01ZR)*1.449 / kPa (d)
IF(RUNUMB.GT.64) THEN
 PRSKPA=(OMEGAV-OMEG01ZR)*1.465
END IF
PRSPSI=PRSKPA/6.8949 / Psid
C
C Return to calling module.
C
RETURN
END
C *=====
SUBROUTINE OMEGA05 (OMEG05ZR,OMEGAV,PRSKPA,RUNUMB)
C *=====
C
C This subroutine is used to calculate the pressure when OMEGA1
C transducer is used. OMEGA1 model PX26; (0 to 5 Psid Range)
C From calibration, see Ph.D. 4 p. 31, the transducer's slope
C is 7.014 kPa/Volts. (of amplified output voltage).
C
C The OMEGA05 pressure transducer was calibrated Dec. '95
C
REAL PRSKPA,OMEGAV,OMEG05ZR,PRSPSI
INTEGER RUNUMB
C
PRSKPA=(OMEGAV-OMEG05ZR)*7.014 / kPa (d)
IF(RUNUMB.GT.64) THEN
 PRSKPA=(OMEGAV-OMEG05ZR)*7.011
END IF
PRSPSI=PRSKPA/6.8949 / Psid
C
C Return to calling module.
C
RETURN
END
C *=====
SUBROUTINE OMEGA15 (OMEG15ZR,OMEGAV,PRSKPA,RUNUMB)
C *=====
C
C This subroutine is used to calculate the pressure when OMEGA15
C transducer is used. OMEGA15 model PX26; (0 to 15 Psid Range)
C From calibration, see Ph.D. 4 p. 31, the transducer's slope
C is 20.796 kPa/Volts. (of amplified output voltage).
C
C The OMEGA15 pressure transducer was calibrated Dec. '95
C
REAL PRSKPA,OMEGAV,OMEG15ZR,PRSPSI
INTEGER RUNUMB
C
PRSKPA=(OMEGAV-OMEG15ZR)*20.796 / kPa (d)
IF(RUNUMB.GT.64) THEN
 PRSKPA=(OMEGAV-OMEG15ZR)*20.565
END IF
PRSPSI=PRSKPA/6.8949 / Psid
C
C Return to calling module.
C
RETURN
END
C *=====

```

## Data Reduction Programs

**Signals.m:** A MATLAB routine used to identify and loads the untreated signals of the optical probes.

---

```
% File name Signals.m
%
% Matlab program used to plot fibre-optic
% probe output signal, all positions.
% Input Run No., Program opens file
% "runwave.dat" to find files for specified
% run.
%%
disp(' ')
disp('Run_sgnl_1.m is used to read and plot the probe output signals:')
disp(' ')
disp(' ')
disp('PLEASE WAIT !!!!! ')
%
fid = fopen('runwave.dat','rt');
[runinfo] = fscanf(fid,'%hd %hd %s %g',[4,inf]);
status = fclose(fid);
runinfo = runinfo';
[regime] = setstr(runinfo(:,[3]));
%%
d = size(runinfo);
runs = d(1,1);
runumb = input('Enter Run no.: ');
%
for i=1 :runs;
if runumb == runinfo(i,1)
 n11=runinfo(i,2);
 dtdiv=runinfo(i,4);
 if regime(i) == 'F'; pattern = 'Stratified Flow';end
 if regime(i) == 'S'; pattern = 'Slug Flow';end
 if regime(i) == 'D'; pattern = 'Pseudo-Slug Flow';end
 if regime(i) == 'A'; pattern = 'Annular Flow';end
 if regime(i) == 'B'; pattern = 'Bubbly Flow';end
 if regime(i) == 'P'; pattern = 'Plug Flow';end
 if regime(i) == 'W'; pattern = 'Wavy Flow';end
 if regime(i) == 'U'; pattern = 'Undefined Regime';end
end
end
%
if n11 == 0
 disp('Output Signals were not recorded for run'), runumb
 break
end
%
if runumb <= 64
 disp('Nicolet Scope was used to store the probe output signals for Run: '), runumb
 disp('Insert CD # 1 & press Return to proceed')
 %
 n12=n11+1;n21=n11+2;n22=n11+3;n31=n11+4;n32=n11+5;
 n41=n11+6;n42=n11+7;n51=n11+8;n52=n11+9;n61=n11+10;
 n62=n11+11;n71=n11+12;n72=n11+13;n81=n11+14;
 n82=n11+15;n91=n11+16;n92=n11+17;
 %
 src='d:\Nicolet\w';
 src='c:\Exp-Data\w';
 %
 n11=int2str(n11);nm11=[src,n11,'.dat'];n12=int2str(n12);nm12=[src,n12,'.dat'];
 n21=int2str(n21);nm21=[src,n21,'.dat'];n22=int2str(n22);nm22=[src,n22,'.dat'];
 n31=int2str(n31);nm31=[src,n31,'.dat'];n32=int2str(n32);nm32=[src,n32,'.dat'];
 n41=int2str(n41);nm41=[src,n41,'.dat'];n42=int2str(n42);nm42=[src,n42,'.dat'];
 n51=int2str(n51);nm51=[src,n51,'.dat'];n52=int2str(n52);nm52=[src,n52,'.dat'];
```

```

n61=int2str(n61);nm61=[src,n61,'.dat'];n62=int2str(n62);nm62=[src,n62,'.dat'];
n71=int2str(n71);nm71=[src,n71,'.dat'];n72=int2str(n72);nm72=[src,n72,'.dat'];
n81=int2str(n81);nm81=[src,n81,'.dat'];n82=int2str(n82);nm82=[src,n82,'.dat'];
n91=int2str(n91);nm91=[src,n91,'.dat'];n92=int2str(n92);nm92=[src,n92,'.dat'];
%
end
%
if runumb > 64
 disp('LeCroy Scope was used to store the probe output signals for Run: '), runumb
 disp('Insert CD # 2/3 & press Return to proceed')
%
tz=0;
if dtdiv == 0.1; npt = 50000;end
if dtdiv == 0.2; npt = 40000;end
if dtdiv == 0.5; npt = 50000;end
if dtdiv == 1.0; npt = 50000;end
if dtdiv == 2.0; npt = 40000;end
tend = dtdiv*10; dt = tend / npt;
%
run=int2str(runumb);
nm11=['f:\Run-',run,'w',run,'01.dat'];nm12=['f:\Run-',run,'w',run,'02.dat'];
nm21=['f:\Run-',run,'w',run,'03.dat'];nm22=['f:\Run-',run,'w',run,'04.dat'];
nm31=['f:\Run-',run,'w',run,'05.dat'];nm32=['f:\Run-',run,'w',run,'06.dat'];
nm41=['f:\Run-',run,'w',run,'07.dat'];nm42=['f:\Run-',run,'w',run,'08.dat'];
nm51=['f:\Run-',run,'w',run,'09.dat'];nm52=['f:\Run-',run,'w',run,'10.dat'];
nm61=['f:\Run-',run,'w',run,'11.dat'];nm62=['f:\Run-',run,'w',run,'12.dat'];
nm71=['f:\Run-',run,'w',run,'13.dat'];nm72=['f:\Run-',run,'w',run,'14.dat'];
nm81=['f:\Run-',run,'w',run,'15.dat'];nm82=['f:\Run-',run,'w',run,'16.dat'];
nm91=['f:\Run-',run,'w',run,'17.dat'];nm92=['f:\Run-',run,'w',run,'18.dat'];
% Note: For run number 72, last file was not named properly !
if runumb == 72;nm92='f:\Run-72w18.dat';end;
if runumb == 73;
nm71=['f:\Run-73_Amendedw7313.dat'];nm72=['f:\Run-73_Amendedw7314.dat'];
nm81=['f:\Run-73_Amendedw7315.dat'];nm82=['f:\Run-73_Amendedw7316.dat'];
nm91=['f:\Run-73_Amendedw7317.dat'];nm92=['f:\Run-73_Amendedw7318.dat'];
end;
end;
%
disp('Reading files for probe position # 1')
fid = fopen(nm11,'rt');[w11,count]=fscanf(fid,'%g',inf);status = fclose(fid);
fid = fopen(nm12,'rt');[w12,count]=fscanf(fid,'%g',inf);status = fclose(fid);
disp('Reading files for probe position # 2')
fid = fopen(nm21,'rt');[w21,count]=fscanf(fid,'%g',inf);status = fclose(fid);
fid = fopen(nm22,'rt');[w22,count]=fscanf(fid,'%g',inf);status = fclose(fid);
disp('Reading files for probe position # 3')
fid = fopen(nm31,'rt');[w31,count]=fscanf(fid,'%g',inf);status = fclose(fid);
fid = fopen(nm32,'rt');[w32,count]=fscanf(fid,'%g',inf);status = fclose(fid);
disp('Reading files for probe position # 4')
fid = fopen(nm41,'rt');[w41,count]=fscanf(fid,'%g',inf);status = fclose(fid);
fid = fopen(nm42,'rt');[w42,count]=fscanf(fid,'%g',inf);status = fclose(fid);
disp('Reading files for probe position # 5')
fid = fopen(nm51,'rt');[w51,count]=fscanf(fid,'%g',inf);status = fclose(fid);
fid = fopen(nm52,'rt');[w52,count]=fscanf(fid,'%g',inf);status = fclose(fid);
disp('Reading files for probe position # 6')
fid = fopen(nm61,'rt');[w61,count]=fscanf(fid,'%g',inf);status = fclose(fid);
fid = fopen(nm62,'rt');[w62,count]=fscanf(fid,'%g',inf);status = fclose(fid);
disp('Reading files for probe position # 7')
fid = fopen(nm71,'rt');[w71,count]=fscanf(fid,'%g',inf);status = fclose(fid);
fid = fopen(nm72,'rt');[w72,count]=fscanf(fid,'%g',inf);status = fclose(fid);
disp('Reading files for probe position # 8')
fid = fopen(nm81,'rt');[w81,count]=fscanf(fid,'%g',inf);status = fclose(fid);
fid = fopen(nm82,'rt');[w82,count]=fscanf(fid,'%g',inf);status = fclose(fid);
disp('Reading files for probe position # 9')
fid = fopen(nm91,'rt');[w91,count]=fscanf(fid,'%g',inf);status = fclose(fid);
fid = fopen(nm92,'rt');[w92,count]=fscanf(fid,'%g',inf);status = fclose(fid);
%
if runumb <= 64
%
dt1=w11(1);tz1=w11(2);npt1=w11(3);tend1=tz1+dt1*(npt1-1);t1=tz1:dt1:tend1;t1=t1';

```

```

dt2=w21(1);tz2=w21(2);npt2=w21(3);tend2=tz2+dt2*(npt2-1);t2=tz2:dt2:tend2;t2=t2';
dt3=w31(1);tz3=w31(2);npt3=w31(3);tend3=tz3+dt3*(npt3-1);t3=tz3:dt3:tend3;t3=t3';
dt4=w41(1);tz4=w41(2);npt4=w41(3);tend4=tz4+dt4*(npt4-1);t4=tz4:dt4:tend4;t4=t4';
dt5=w51(1);tz5=w51(2);npt5=w51(3);tend5=tz5+dt5*(npt5-1);t5=tz5:dt5:tend5;t5=t5';
dt6=w61(1);tz6=w61(2);npt6=w61(3);tend6=tz6+dt6*(npt6-1);t6=tz6:dt6:tend6;t6=t6';
dt7=w71(1);tz7=w71(2);npt7=w71(3);tend7=tz7+dt7*(npt7-1);t7=tz7:dt7:tend7;t7=t7';
dt8=w81(1);tz8=w81(2);npt8=w81(3);tend8=tz8+dt8*(npt8-1);t8=tz8:dt8:tend8;t8=t8';
dt9=w91(1);tz9=w91(2);npt9=w91(3);tend9=tz9+dt9*(npt9-1);t9=tz9:dt9:tend9;t9=t9';
%
v1f=w11(4:npt1+3);v1b=w12(4:npt1+3);clear w11;clear w12;
v2f=w21(4:npt2+3);v2b=w22(4:npt2+3);clear w21;clear w22;
v3f=w31(4:npt3+3);v3b=w32(4:npt3+3);clear w31;clear w32;
v4f=w41(4:npt4+3);v4b=w42(4:npt4+3);clear w41;clear w42;
v5f=w51(4:npt5+3);v5b=w52(4:npt5+3);clear w51;clear w52;
v6f=w61(4:npt6+3);v6b=w62(4:npt6+3);clear w61;clear w62;
v7f=w71(4:npt7+3);v7b=w72(4:npt7+3);clear w71;clear w72;
v8f=w81(4:npt8+3);v8b=w82(4:npt8+3);clear w81;clear w82;
v9f=w91(4:npt9+3);v9b=w92(4:npt9+3);clear w91;clear w92;
%
end
%
if runumb > 64
%
tz1=tz;tz2=tz;tz3=tz;tz4=tz;tz5=tz;tz6=tz;tz7=tz;tz8=tz;tz9=tz;
dt1=dt;dt2=dt;dt3=dt;dt4=dt;dt5=dt;dt6=dt;dt7=dt;dt8=dt;dt9=dt;
t1=tz:dt1:tend-dt;t1=t1';t2=t1;t3=t1;t4=t1;t5=t1;t6=t1;t7=t1;t8=t1;t9=t1;
tend1=tend;tend2=tend;tend3=tend;tend4=tend;tend5=tend;tend6=tend;tend7=tend;
tend8=tend;tend9=tend;
%
v1f=w11(2:npt+1);v1b=w12(2:npt+1);clear w11;clear w12;
v2f=w21(2:npt+1);v2b=w22(2:npt+1);clear w21;clear w22;
v3f=w31(2:npt+1);v3b=w32(2:npt+1);clear w31;clear w32;
v4f=w41(2:npt+1);v4b=w42(2:npt+1);clear w41;clear w42;
v5f=w51(2:npt+1);v5b=w52(2:npt+1);clear w51;clear w52;
v6f=w61(2:npt+1);v6b=w62(2:npt+1);clear w61;clear w62;
v7f=w71(2:npt+1);v7b=w72(2:npt+1);clear w71;clear w72;
v8f=w81(2:npt+1);v8b=w82(2:npt+1);clear w81;clear w82;
v9f=w91(2:npt+1);v9b=w92(2:npt+1);clear w91;clear w92;
%
end
%
if fig == 1
disp(' ')
disp('Preparing Figure - Please wait')
%
subplot(9,1,9);plot(t1,v1f,'-',t1,v1b,':');axis ([tz1,tend1+dt1,0,1]);
xlabel('Time (sec)','fontname','helvetica','fontsize',10);
set(gca,'xtick',[tz1:(tend1-tz1)/10:tend1],'fontname','helvetica','fontsize',6);
set(gca,'ytick',[0:0.5:1],'fontname','helvetica','fontsize',6);
clear w11;clear w12;
box off
%
subplot(9,1,8);plot(t2,v2f,'-',t2,v2b,':');axis ([tz2,tend2+dt2,0,1]);
set(gca,'xtick',[tz2:tend2:tend2],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:0.5:1],'fontname','helvetica','fontsize',6);
box off
%
subplot(9,1,7);plot(t3,v3f,'-',t3,v3b,':');axis ([tz3,tend3+dt3,0,1]);
set(gca,'xtick',[tz3:tend3:tend3],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:0.5:1],'fontname','helvetica','fontsize',6);
box off
%
subplot(9,1,6);plot(t4,v4f,'-',t4,v4b,':');axis ([tz4,tend4+dt4,0,1]);
set(gca,'xtick',[tz4:tend4:tend4],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:0.5:1],'fontname','helvetica','fontsize',6);
box off

```

```

%
subplot(9,1,5);plot(t5,v5f,'-',t5,v5b,':');axis([tz5,tend5+dt5,0,1]);
ylabel('Output Voltage (Volts)','fontname','helvetica','fontsize',10);
set(gca,'xtick',[tz5:tend5:tend5],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:0.5:1],'fontname','helvetica','fontsize',6);
box off
%
subplot(9,1,4);plot(t6,v6f,'-',t6,v6b,':');axis([tz6,tend6+dt6,0,1]);
set(gca,'xtick',[tz6:tend6:tend6],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:0.5:1],'fontname','helvetica','fontsize',6);
box off
%
subplot(9,1,3);plot(t7,v7f,'-',t7,v7b,':');axis([tz7,tend7+dt7,0,1]);
set(gca,'xtick',[tz7:tend7:tend7],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:0.5:1],'fontname','helvetica','fontsize',6);
box off
%
subplot(9,1,2);plot(t8,v8f,'-',t8,v8b,':');axis([tz8,tend8+dt8,0,1]);
set(gca,'xtick',[tz8:tend8:tend8],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:0.5:1],'fontname','helvetica','fontsize',6);
box off
%
subplot(9,1,1);plot(t9,v9f,'-',t9,v9b,':');axis([tz9,tend9+dt9,0,1]);
set(gca,'xtick',[tz9:tend9:tend9],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:0.5:1],'fontname','helvetica','fontsize',6);
box off
%
%title(['Twin-needle Fibre-optic Probe, Output Signals: Run # ',int2str(runumb), ' Regime: '
pattern'],'fontname','helvetica','fontsize',10);
orient tall;
end;
set(gcf,'Renderer','painters');
% clear Matlab workspace of some variable which are no longer needed.
clear n11 nm* t* dt dt* dtdiv count run runs runinfo ans d fid i regime status
%
```

## Data Reduction Programs

**Histogram.m:** A MATLAB routine used to calculate the Probability Density Function (PDF) and draw the histograms of the time traces of the optical probes.

---

```
% Histograms.m
% calculates and draws PDF of the time traces of the optical probes.
figure (2)
a1=1.0;
a2=2.0;
b1=0.2;
b2=0.4;
c2=10000;
c1=40000;
d2=5000;
d1=20000;
if runumb > 64
 a=a1;b=b1;c=c1;d=d1;xh=0:0.025:1;
else
 a=a2;b=b2;c=c2;d=d2;xh=0:0.05:2;
end
% For sample test run 68.
%a=0.6;b=0.1;c=30000;d=10000;
%
subplot (9,2,17);hist(v1f,xh);axis([0,a,0,c]);
set(gca,'xtick',[0:b:a],'fontname','helvetica','fontsize',6);
set(gca,'ytick',[0:d:c],'fontname','helvetica','fontsize',6);
xlabel('Output Voltage (V)','fontname','helvetica','fontsize',10);
box off
%
subplot (9,2,18);hist(v1b,xh);axis([0,a,0,c]);
set(gca,'xtick',[0:b:a],'fontname','helvetica','fontsize',7);
set(gca,'ytick',[0:d:c],'fontname','helvetica','fontsize',7);
xlabel('Output Voltage (V)','fontname','helvetica','fontsize',10);
box off
%
subplot (9,2,15);hist(v2f,xh);axis([0,a,0,c]);
set(gca,'xtick',[0:a:a],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:d:c],'fontname','helvetica','fontsize',6);
box off
%
subplot (9,2,16);hist(v2b,xh);axis([0,a,0,c]);
set(gca,'xtick',[0:a:a],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:d:c],'fontname','helvetica','fontsize',6);
box off
%
subplot (9,2,13);hist(v3f,xh);axis([0,a,0,c]);
set(gca,'xtick',[0:a:a],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:d:c],'fontname','helvetica','fontsize',6);
box off
%
subplot (9,2,14);hist(v3b,xh);axis([0,a,0,c]);
set(gca,'xtick',[0:a:a],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:d:c],'fontname','helvetica','fontsize',6);
box off
%
subplot (9,2,11);hist(v4f,xh);axis([0,a,0,c]);
set(gca,'xtick',[0:a:a],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:d:c],'fontname','helvetica','fontsize',6);
box off
%
subplot (9,2,12);hist(v4b,xh);axis([0,a,0,c]);
```

```

set(gca,'xtick',[0:a:a],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:d:c],'fontname','helvetica','fontsize',6);
box off
%
subplot (9,2,9);hist(v5f,xh);axis([0,a,0,c]);
set(gca,'xtick',[0:a:a],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:d:c],'fontname','helvetica','fontsize',6);
box off
%
ylabel('PDF (Counts)','fontname','helvetica','fontsize',10);
%
subplot (9,2,10);hist(v5b,xh);axis([0,a,0,c]);
set(gca,'xtick',[0:a:a],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:d:c],'fontname','helvetica','fontsize',6);
box off
%
subplot (9,2,7);hist(v6f,xh);axis([0,a,0,c]);
set(gca,'xtick',[0:a:a],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:d:c],'fontname','helvetica','fontsize',6);
box off
%
subplot (9,2,8);hist(v6b,xh);axis([0,a,0,c]);
set(gca,'xtick',[0:a:a],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:d:c],'fontname','helvetica','fontsize',6);
box off
%
subplot (9,2,5);hist(v7f,xh);axis([0,a,0,c]);
set(gca,'xtick',[0:a:a],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:d:c],'fontname','helvetica','fontsize',6);
box off
%
subplot (9,2,6);hist(v7b,xh);axis([0,a,0,c]);
set(gca,'xtick',[0:a:a],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:d:c],'fontname','helvetica','fontsize',6);
box off
%
subplot (9,2,3);hist(v8f,xh);axis([0,a,0,c]);
set(gca,'xtick',[0:a:a],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:d:c],'fontname','helvetica','fontsize',6);
box off
%
subplot (9,2,4);hist(v8b,xh);axis([0,a,0,c]);
set(gca,'xtick',[0:a:a],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:d:c],'fontname','helvetica','fontsize',6);
box off
%
subplot (9,2,1);hist(v9f,xh);axis([0,a,0,c]);
set(gca,'xtick',[0:a:a],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:d:c],'fontname','helvetica','fontsize',6);
box off
%
subplot (9,2,2);hist(v9b,xh);axis([0,a,0,c]);
set(gca,'xtick',[0:a:a],'fontname','helvetica','fontsize',6);
set(gca,'XTickLabel','')
set(gca,'ytick',[0:d:c],'fontname','helvetica','fontsize',6);
box off
%title(['Twin-needle Fibre-optic Probe, Histogram: Run # ',int2str(runumb),' Regime: '
pattern],'fontname','helvetica','fontsize',10);
orient tall;
set(gcf,'Renderer','painters');

```


Data Reduction Programs

Void.m: A MATLAB routine used to calculate the local and area-weighted average void fractions.

```
% void.m
% Run this after histograms.m;
%
h=figure (3);
%
s=0:0.8:1.6;
%[n,x]=hist(v1f,s);v(1,1)=(n(2)+n(3))/(n(1)+n(2)+n(3));[n,x]=hist (v1b,s);v(1,2)=(n(2)+n(3))/(n(1)+n(2)+n(3));
%[n,x]=hist(v2f,s);v(2,1)=(n(2)+n(3))/(n(1)+n(2)+n(3));[n,x]=hist (v2b,s);v(2,2)=(n(2)+n(3))/(n(1)+n(2)+n(3));
%[n,x]=hist(v3f,s);v(3,1)=(n(2)+n(3))/(n(1)+n(2)+n(3));[n,x]=hist (v3b,s);v(3,2)=(n(2)+n(3))/(n(1)+n(2)+n(3));
%[n,x]=hist(v4f,s);v(4,1)=(n(2)+n(3))/(n(1)+n(2)+n(3));[n,x]=hist (v4b,s);v(4,2)=(n(2)+n(3))/(n(1)+n(2)+n(3));
%[n,x]=hist(v5f,s);v(5,1)=(n(2)+n(3))/(n(1)+n(2)+n(3));[n,x]=hist (v5b,s);v(5,2)=(n(2)+n(3))/(n(1)+n(2)+n(3));
%[n,x]=hist(v6f,s);v(6,1)=(n(2)+n(3))/(n(1)+n(2)+n(3));[n,x]=hist (v6b,s);v(6,2)=(n(2)+n(3))/(n(1)+n(2)+n(3));
%[n,x]=hist(v7f,s);v(7,1)=(n(2)+n(3))/(n(1)+n(2)+n(3));[n,x]=hist (v7b,s);v(7,2)=(n(2)+n(3))/(n(1)+n(2)+n(3));
%[n,x]=hist(v8f,s);v(8,1)=(n(2)+n(3))/(n(1)+n(2)+n(3));[n,x]=hist (v8b,s);v(8,2)=(n(2)+n(3))/(n(1)+n(2)+n(3));
%[n,x]=hist(v9f,s);v(9,1)=(n(2)+n(3))/(n(1)+n(2)+n(3));[n,x]=hist (v9b,s);v(9,2)=(n(2)+n(3))/(n(1)+n(2)+n(3));
%
vs=input('Enter Threshold Voltage.: ');
v1fc=0;v1bc=0;v2fc=0;v2bc=0;v3fc=0;v3bc=0;v4fc=0;v4bc=0;v5fc=0;v5bc=0;
v6fc=0;v6bc=0;v7fc=0;v7bc=0;v8fc=0;v8bc=0;v9fc=0;v9bc=0;
%
for i = 1: npt1;if v1f(i) > vs; v1fc = v1fc + 1; end; end; v(1,1) = v1fc / npt1;
for i = 1: npt1;if v1b(i) > vs; v1bc = v1bc + 1; end; end; v(1,2) = v1bc / npt1;
for i = 1: npt2;if v2f(i) > vs; v2fc = v2fc + 1; end; end; v(2,1) = v2fc / npt2;
for i = 1: npt2;if v2b(i) > vs; v2bc = v2bc + 1; end; end; v(2,2) = v2bc / npt2;
for i = 1: npt3;if v3f(i) > vs; v3fc = v3fc + 1; end; end; v(3,1) = v3fc / npt3;
for i = 1: npt3;if v3b(i) > vs; v3bc = v3bc + 1; end; end; v(3,2) = v3bc / npt3;
for i = 1: npt4;if v4f(i) > vs; v4fc = v4fc + 1; end; end; v(4,1) = v4fc / npt4;
for i = 1: npt4;if v4b(i) > vs; v4bc = v4bc + 1; end; end; v(4,2) = v4bc / npt4;
for i = 1: npt5;if v5f(i) > vs; v5fc = v5fc + 1; end; end; v(5,1) = v5fc / npt5;
for i = 1: npt5;if v5b(i) > vs; v5bc = v5bc + 1; end; end; v(5,2) = v5bc / npt5;
for i = 1: npt6;if v6f(i) > vs; v6fc = v6fc + 1; end; end; v(6,1) = v6fc / npt6;
for i = 1: npt6;if v6b(i) > vs; v6bc = v6bc + 1; end; end; v(6,2) = v6bc / npt6;
for i = 1: npt7;if v7f(i) > vs; v7fc = v7fc + 1; end; end; v(7,1) = v7fc / npt7;
for i = 1: npt7;if v7b(i) > vs; v7bc = v7bc + 1; end; end; v(7,2) = v7bc / npt7;
for i = 1: npt8;if v8f(i) > vs; v8fc = v8fc + 1; end; end; v(8,1) = v8fc / npt8;
for i = 1: npt8;if v8b(i) > vs; v8bc = v8bc + 1; end; end; v(8,2) = v8bc / npt8;
for i = 1: npt9;if v9f(i) > vs; v9fc = v9fc + 1; end; end; v(9,1) = v9fc / npt9;
for i = 1: npt9;if v9b(i) > vs; v9bc = v9bc + 1; end; end; v(9,2) = v9bc / npt9;
% Area ratios:
ar1=0.0941;ar2=0.1013;ar3=0.1165;ar4=0.1245;ar5=0.1271;ar6=0.1245;ar7=0.1165;ar8=0.1013;ar9=0.0941;
art=ar1+ar2+ar3+ar4+ar5+ar6+ar7+ar8+ar9;
%
alf1=v(1,1)*ar1+v(2,1)*ar2+v(3,1)*ar3+v(4,1)*ar4+v(5,1)*ar5+v(6,1)*ar6+v(7,1)*ar7+v(8,1)*ar8+v(9,1)*ar9;
alf2=v(1,2)*ar1+v(2,2)*ar2+v(3,2)*ar3+v(4,2)*ar4+v(5,2)*ar5+v(6,2)*ar6+v(7,2)*ar7+v(8,2)*ar8+v(9,2)*ar9;
alfav=(alf1+alf2)/2;
v
alf1
alf2
alfav
%
pos =0.8:0.2:0.8;
h1=plot (pos,v(:,1),'s',pos,v(:,2),'o');
set (h1,'markersize',5);
axis([-1,1,0,1]);
set(gca,'xtick',[-1:0.2:1],'fontname','helvetica','fontsize',10);
set(gca,'ytick',[0:0.1:1],'fontname','helvetica','fontsize',10);
orient tall;
%
title(['Void Fraction Distribution: Run # ',int2str(runumb), ' Regime: ' pattern],'fontname','helvetica','fontsize',10);
xlabel('Radial Position: r/R','fontname','helvetica','fontsize',9);
ylabel('Local Void Fraction','fontname','helvetica','fontsize',9);
h2=legend('Upstream Sensor','Downstream Sensor');
```

```

set (h2,'fontname','helvetica','fontsize',8);
axes(h2); axis off
%gtext(['Alfa1 =',num2str(alf1),'Alfa2 =',num2str(alf2),Alfav =',num2str(alfav)]];
h3 = gtext(['Alfa1 = ',num2str(alf1),' Alfa2 = ',num2str(alf2),' Alfav = ',num2str(alfav)]];
set (h3,'fontname','helvetica','fontsize',8);
%
set (h,'paperposition',[2 5 4.5 4.5]);
%
```

Appendix D

Tabulated Raw Data

Tabulated Raw Data

This Appendix contains the measured voltages and temperatures for each of the experimental test runs. The time traces of the optical probes and the recorded videotapes of the flow can be made available upon request.

For each test run, the measured voltages and recorded temperatures are presented in three lines with the following:

- (1) Zero readings of the AVT transducer, the paddle-wheel sensor, and the 4 pressure transducers.
- (2) Voltage output of the AVT transducer, the paddle-wheel sensor, the 4 pressure transducers, and the three thermocouples; *before* taking measurements of the local void fractions.
- (3) Voltage output of the AVT transducer, the paddle-wheel sensor, the 4 pressure transducers, and the three thermocouples; *after* taking measurements of the local void fractions.

Notes:

- (a) The zero readings of the instruments are checked at the end of each set of experimental test runs (about 10 to 15 test runs) to ensure that any drift of the zero readings is within an acceptable range, i.e., the expected uncertainty of the measurements.
- (b) Each reading represents the average of 5000 measurements sampled at about 1000 samples per second.
- (c) For each test run, the overall reading is taken as the average of the readings given in (2) and (3) above.

Test Run No.	Flow rates (Volts)		Wall Pressure (Volts)				Temperature (°C)		
	Air	Water	AVT	inlet	middle	exit	air	water	Mixture
45	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	0.054	-0.002	4.703	0.039	-0.043	-0.071	19.7	19.6	19.6
	0.053	-0.002	4.645	0.039	-0.045	-0.072	19.7	19.7	19.6
46	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	0.101	-0.002	4.631	0.043	-0.041	-0.068	19.9	19.7	19.6
	0.103	0.190	4.795	0.042	-0.049	-0.084	19.8	19.6	19.5
47	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	0.203	0.339	4.724	0.048	-0.053	-0.090	19.8	19.6	19.5
	0.200	0.223	4.672	0.067	-0.053	-0.084	19.7	19.6	19.4
48	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	0.290	0.215	4.774	0.096	-0.047	-0.071	19.5	19.6	19.4
	0.293	0.202	4.779	0.098	-0.051	-0.087	19.8	19.7	19.4
49	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	0.530	0.235	4.664	0.166	0.040	-0.061	19.8	19.6	19.1
	0.512	0.233	4.404	0.149	0.022	-0.058	19.4	19.7	19.1
50	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	0.701	0.233	4.655	0.262	0.111	-0.017	19.8	19.7	19.0
	0.697	0.234	4.700	0.264	0.108	-0.015	19.9	19.7	18.9
51	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	0.845	-0.002	4.583	0.346	0.173	0.028	19.8	19.7	18.8
	0.830	-0.002	4.556	0.345	0.182	0.038	19.8	19.7	18.8
52	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	1.093	0.078	4.465	0.523	0.320	0.137	19.9	19.8	18.7
	1.041	0.061	4.166	0.492	0.253	0.122	19.5	19.7	18.6
53	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	1.292	-0.002	3.843	0.694	0.499	0.234	19.5	19.7	18.3
	1.327	0.131	3.956	0.705	0.500	0.240	19.7	19.7	18.4
54	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	1.314	0.151	3.912	0.704	0.578	0.547	19.9	19.7	18.4
	1.272	0.033	3.822	0.706	0.522	0.552	19.7	19.7	18.4
55	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	1.244	0.831	3.790	1.455	1.135	0.668	19.8	19.7	19.1
	1.331	0.834	4.079	1.559	1.219	0.784	20.2	19.7	19.2
56	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	1.313	1.163	4.025	1.864	1.438	0.853	20.2	19.8	19.4
	1.291	1.127	3.909	1.764	1.396	0.832	20.3	19.8	19.5
57	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	1.332	1.754	4.009	2.419	2.094	1.004	20.3	19.7	19.4
	1.273	1.805	3.787	2.316	1.888	0.770	20.1	19.8	19.5
58	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	1.274	2.524	3.838	3.168	2.526	1.058	20.2	19.9	19.6
	1.315	2.426	4.021	3.186	2.449	0.932	20.4	19.8	19.6

Test Run No.	Flow rates (Volts)		Wall Pressure (Volts)				Temperature (°C)		
	Air	Water	AVT	inlet	middle	exit	air	water	Mixture
59	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	1.272	3.361	3.921	4.184	3.442	1.329	20.0	19.7	19.5
	1.242	3.430	3.784	4.060	3.023	1.240	20.1	19.8	19.6
60	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	0.466	3.683	4.527	2.103	1.528	0.653	20.2	19.7	19.6
	0.481	3.676	4.689	2.080	1.272	0.560	20.4	19.7	19.6
61	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	0.223	3.715	4.865	1.229	0.528	0.163	20.2	19.7	19.6
	0.218	3.730	4.670	1.139	0.822	0.143	20.1	19.7	19.6
62	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	0.180	3.793	4.545	1.077	0.652	0.181	20.0	19.7	19.6
	0.189	3.737	4.811	1.051	0.544	0.154	20.1	19.7	19.6
63	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	0.084	3.778	4.824	0.665	0.534	0.025	20.1	19.7	19.6
	0.083	3.812	4.673	0.657	0.536	0.021	19.9	19.7	19.6
64	0.021	-0.002	0.228	0.031	-0.050	-0.134			
	0.028	3.874	4.792	0.452	0.492	0.006	20.1	19.6	19.6
	0.031	3.820	4.619	0.447	0.490	0.001	19.8	19.7	19.7
66	0.000	0.000	0.000	0.000	0.000	0.000			
	0.221	-0.005	4.673	0.047	-0.039	-0.130	21.9	21.5	22.0
	0.387	-0.005	4.716	0.047	-0.039	-0.130	22.2	21.6	22.0
67	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	0.759	3.541	4.728	0.475	0.254	0.000	21.9	17.1	16.9
	0.756	3.543	4.726	0.476	0.240	-0.007	21.8	17.0	16.8
68	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	1.071	3.534	4.717	0.662	0.387	0.030	22.2	16.8	16.7
	1.067	3.534	4.726	0.647	0.382	0.019	22.3	16.8	16.6
69	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.539	3.559	4.729	0.406	0.193	0.004	21.8	16.7	16.6
	0.544	3.589	4.734	0.405	0.202	-0.007	21.7	16.7	16.6
70	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.542	3.829	4.737	0.438	0.225	0.007	21.7	16.7	16.6
	0.539	3.848	4.728	0.437	0.213	0.001	21.8	16.8	16.6
71	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	1.102	3.807	4.714	0.744	0.444	0.054	22.4	16.9	16.7
	1.088	3.810	4.707	0.742	0.431	0.050	22.5	17.0	16.9
72	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.733	1.090	4.710	0.126	0.024	-0.074	22.4	17.6	17.6
	0.739	1.113	4.722	0.127	-0.003	-0.041	22.2	17.5	17.5
73	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.727	0.606	4.711	0.076	-0.048	-0.095	22.2	17.6	17.6
	0.739	0.620	4.725	0.058	-0.032	-0.078	22.2	17.7	18.3
74	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.735	1.190	4.735	0.127	0.010	-0.071	22.3	17.6	17.5
	0.731	1.178	4.716	0.134	0.017	-0.034	22.2	17.4	17.3

Test Run No.	Flow rates (Volts)		Wall Pressure (Volts)				Temperature (°C)		
	Air	Water	AVT	inlet	middle	exit	air	water	Mixture
75	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.727	2.238	4.727	0.267	0.097	-0.042	22.2	17.3	17.1
	0.659	2.244	4.173	0.256	0.091	-0.037	22.1	17.2	17.0
76	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.720	3.885	4.492	0.525	0.288	0.009	22.1	17.1	16.9
	0.741	3.916	4.702	0.525	0.282	0.013	22.1	17.0	16.8
77	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	1.077	3.935	4.721	0.720	0.438	0.047	22.7	16.9	16.8
	1.056	3.878	4.723	0.742	0.424	0.062	22.7	16.8	16.6
78	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	1.243	3.902	4.717	0.887	0.550	0.097	22.6	16.7	16.5
	1.229	3.861	4.711	0.891	0.534	0.086	22.5	16.5	16.4
79	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	1.663	3.831	4.687	1.387	0.823	0.254	22.5	16.4	16.3
	1.663	3.844	4.675	1.331	0.694	0.317	22.1	16.1	16.0
80	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	2.247	3.641	4.563	2.045	1.275	0.606	21.7	15.8	15.7
	2.248	3.636	4.557	1.961	1.431	0.715	21.7	15.9	15.8
81	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.461	0.863	4.727	0.081	-0.053	-0.060	21.4	17.8	17.8
	0.464	0.845	4.724	0.085	-0.059	-0.076	21.3	17.7	17.8
82	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.465	2.012	4.731	0.177	-0.021	-0.066	21.3	17.4	17.3
	0.460	2.044	4.731	0.181	-0.011	-0.062	21.0	17.1	17.0
83	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.463	3.556	4.725	0.376	0.173	0.012	20.9	16.7	16.5
	0.464	3.577	4.706	0.373	0.157	0.000	20.9	16.2	16.0
84	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.470	3.887	4.724	0.440	0.226	0.020	20.8	16.0	15.9
	0.467	3.914	4.712	0.432	0.199	0.010	20.8	15.7	15.6
85	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.411	1.128	4.954	0.092	0.008	-0.011	21.3	16.6	17.2
	0.410	1.132	4.900	0.100	0.012	-0.012	21.2	16.4	17.1
86	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.415	2.777	4.965	0.240	0.105	-0.015	21.2	16.2	16.0
	0.410	2.816	4.880	0.243	0.102	-0.014	21.1	16.1	15.9
87	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.421	3.837	4.998	0.378	0.204	0.004	21.1	15.9	15.7
	0.419	3.797	4.937	0.379	0.205	0.006	21.0	16.0	15.7
88	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.424	4.502	4.997	0.489	0.274	0.022	21.1	15.9	15.7
	0.417	4.545	4.911	0.490	0.279	0.018	21.0	15.8	15.6
89	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.414	4.567	4.939	0.481	0.272	0.015	21.0	15.9	15.6
	0.399	4.602	4.982	0.468	0.265	0.017	21.0	15.8	15.6

Test Run No.	Flow rates (Volts)		Wall Pressure (Volts)				Temperature (°C)		
	Air	Water	AVT	inlet	middle	exit	air	water	Mixture
90	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.855	3.454	4.764	0.516	0.305	0.013	21.8	15.8	15.6
	0.880	3.451	4.999	0.511	0.304	0.014	21.9	15.7	15.5
91	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	1.063	3.412	4.979	0.636	0.396	0.029	21.9	15.7	15.5
	1.040	3.483	4.653	0.631	0.382	0.035	22.0	15.7	15.5
92	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	1.018	4.372	4.678	0.837	0.551	0.068	22.0	15.7	15.5
	1.029	4.339	4.665	0.840	0.532	0.063	22.0	15.8	15.6
93	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.693	3.794	4.661	0.537	0.309	0.017	21.5	15.8	15.6
	0.712	3.814	4.694	0.516	0.301	0.011	21.5	15.8	15.6
94	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.861	0.608	4.893	0.069	-0.029	-0.089	21.4	16.3	16.4
	0.792	0.653	4.815	0.091	-0.032	-0.085	21.4	16.5	16.5
95	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.807	0.931	4.686	0.126	0.038	-0.041	21.5	16.5	16.6
	0.849	0.948	4.878	0.130	0.051	-0.062	21.8	16.5	16.4
96	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	0.825	3.175	4.848	0.454	0.247	-0.003	21.8	16.3	16.2
	0.853	3.199	4.845	0.464	0.262	0.000	21.8	16.1	15.9
97	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	1.260	3.146	4.678	0.722	0.439	0.069	21.9	16.0	15.9
	1.290	3.165	4.904	0.759	0.484	0.114	22.0	16.0	15.9
98	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	1.606	3.184	4.942	1.020	0.674	0.206	22.1	16.0	15.8
	1.580	3.146	4.679	0.989	0.582	0.275	22.0	16.0	15.8
99	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	1.954	3.132	4.567	1.307	0.888	0.508	22.0	15.9	15.8
	1.977	3.117	4.917	1.370	1.224	0.364	22.2	15.8	15.6
100	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	2.435	3.399	4.861	2.275	1.733	1.014	22.2	15.8	15.6
	2.401	3.392	4.713	2.211	1.409	0.921	22.0	15.7	15.6
101	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	2.708	3.473	4.431	2.776	1.956	1.210	22.2	15.7	15.6
	2.740	3.437	4.558	2.857	2.054	1.116	22.5	15.8	15.6
102	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	1.003	0.958	4.867	0.160	0.016	-0.066	21.6	17.5	17.6
	0.981	0.969	4.878	0.152	0.077	-0.059	21.3	17.3	17.2
103	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	1.499	0.988	4.731	0.213	0.085	0.058	21.0	17.0	17.2
	1.490	1.066	4.631	0.203	0.072	0.069	20.8	16.7	16.7
104	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	1.802	0.972	4.634	0.328	0.339	0.046	20.8	16.7	16.7
	1.818	1.033	4.830	0.280	0.349	0.044	20.9	16.6	16.5

Test Run No.	Flow rates (Volts)		Wall Pressure (Volts)				Temperature (°C)		
	Air	Water	AVT	inlet	middle	exit	air	water	Mixture
105	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	2.524	0.991	4.825	0.668	0.432	0.308	20.8	16.6	16.5
	2.479	1.005	4.636	0.606	0.333	0.206	20.9	16.5	16.4
106	0.019	-0.005	0.020	0.047	-0.039	-0.130			
	2.485	1.870	4.503	1.136	0.820	0.484	21.0	16.3	16.2
	2.505	1.853	4.739	1.348	0.797	0.616	21.1	15.8	15.7
107	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	0.470	1.169	4.831	0.089	-0.023	0.001	19.9	18.3	18.3
	0.464	1.171	4.787	0.094	-0.023	0.001	19.8	18.3	18.3
108	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	0.957	1.180	4.963	0.165	0.013	0.001	20.2	18.3	18.3
	0.968	1.158	4.845	0.151	0.031	0.001	20.0	18.3	18.2
109	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	0.987	1.149	4.893	0.160	0.037	0.001	20.0	18.3	18.2
	0.963	1.172	4.785	0.161	0.013	0.001	20.0	18.3	18.2
110	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	0.948	2.131	4.786	0.308	0.115	0.001	20.0	18.3	18.2
	0.955	2.119	4.808	0.306	0.134	0.001	20.0	18.2	18.2
111	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	0.943	3.386	4.726	0.529	0.281	0.001	20.0	18.2	18.2
	0.975	3.395	4.991	0.537	0.274	0.001	20.2	18.1	18.1
112	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	0.966	4.049	4.747	0.661	0.375	0.001	20.0	18.1	18.0
	0.955	4.057	4.910	0.667	0.374	0.001	20.2	18.1	18.0
113	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	0.946	4.332	4.806	0.750	0.427	0.001	20.0	18.1	18.1
	0.972	4.411	4.997	0.750	0.428	0.001	20.2	18.1	18.1
114	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	0.944	4.794	4.722	0.835	0.488	0.001	20.0	18.1	18.1
	0.960	4.774	4.967	0.836	0.492	0.001	20.2	18.1	18.0
115	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	0.966	4.999	4.969	0.923	0.545	0.001	20.2	18.0	18.0
	0.960	4.999	4.865	0.937	0.554	0.001	20.0	18.0	18.0
116	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	0.967	4.999	4.932	0.967	0.585	0.001	20.1	18.0	18.0
	0.963	4.999	4.837	0.955	0.568	0.001	20.0	17.9	17.9
117	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	1.709	0.599	4.831	0.160	0.005	0.001	20.2	18.6	18.8
	1.693	0.601	4.729	0.181	0.011	0.001	20.2	18.5	18.6
118	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	1.687	0.839	4.742	0.202	0.063	0.001	20.2	18.4	18.4
	1.691	0.917	4.828	0.297	0.042	0.001	20.2	18.4	18.3
119	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	1.708	1.293	4.969	0.414	0.221	0.001	20.3	18.3	18.2
	1.689	1.354	4.846	0.406	0.223	0.001	20.2	18.3	18.2

Test Run No.	Flow rates (Volts)		Wall Pressure (Volts)				Temperature (°C)		
	Air	Water	AVT	inlet	middle	exit	air	water	Mixture
120	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	1.717	2.135	4.963	0.747	0.545	0.001	20.3	18.3	18.2
	1.705	2.141	4.844	0.725	0.446	0.001	20.2	18.3	18.1
121	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	1.693	2.991	4.902	1.031	0.736	0.001	20.2	18.2	18.1
	1.703	3.056	4.905	1.012	0.655	0.001	20.2	18.2	18.1
122	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	1.697	3.498	4.882	1.235	0.828	0.001	20.2	18.2	18.0
	1.688	3.520	4.778	1.178	0.682	0.001	20.2	18.2	18.0
123	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	1.687	4.549	4.809	1.534	1.094	0.001	20.2	18.1	18.0
	1.677	4.556	4.708	1.565	1.058	0.002	20.2	18.2	18.0
124	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	1.692	4.999	4.866	1.849	1.202	0.002	20.2	18.1	18.0
	1.684	4.999	4.770	1.792	1.308	0.001	20.2	18.1	18.0
125	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	2.307	4.955	4.886	3.006	1.947	0.002	20.3	18.0	17.9
	2.303	4.999	4.801	2.860	1.870	0.002	20.3	18.1	17.9
126	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	3.197	4.668	4.535	4.999	3.370	0.002	20.2	18.1	17.9
	3.188	4.645	4.409	4.999	3.241	0.002	20.1	18.1	17.9
127	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	3.532	4.487	3.017	4.999	4.028	0.002	19.1	18.0	17.8
	3.465	4.558	2.895	4.999	3.747	0.002	19.5	18.0	17.8
128	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	3.430	4.557	3.172	4.999	3.913	0.002	20.0	18.0	17.8
	3.438	4.560	3.137	4.999	4.126	0.002	19.9	18.0	17.8
129	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	3.465	3.089	3.138	4.612	3.467	0.002	20.0	18.0	17.7
	3.449	3.142	3.059	4.435	3.429	0.002	19.9	18.0	17.8
130	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	3.485	1.250	3.029	2.342	2.115	0.002	19.9	18.0	17.7
	3.455	1.243	2.934	2.274	2.126	0.002	19.8	18.0	17.7
131	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	3.434	0.002	2.885	0.844	0.906	0.001	19.7	18.1	17.6
	3.469	0.001	2.994	0.853	0.904	0.001	19.9	18.1	17.5
132	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	3.444	0.736	2.968	1.835	1.717	0.002	20.0	18.2	17.6
	3.468	0.732	2.999	1.806	1.688	0.002	20.1	18.1	17.7
133	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	3.422	1.507	2.875	2.509	2.299	0.002	20.0	18.2	17.9
	3.444	1.525	2.988	2.632	2.383	0.002	20.1	18.1	17.9
134	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	3.458	2.055	2.989	3.100	2.674	0.002	20.2	18.1	17.9
	3.421	2.037	2.895	3.109	2.810	0.002	20.1	18.1	17.9

Test Run No.	Flow rates (Volts)		Wall Pressure (Volts)				Temperature (°C)		
	Air	Water	AVT	inlet	middle	exit	air	water	Mixture
135	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	3.448	2.339	3.000	3.498	3.109	0.002	20.2	18.1	17.9
	3.419	2.371	2.906	3.419	2.988	0.002	20.1	18.1	17.9
136	0.019	-0.005	0.019	0.047	-0.039	-0.130			
	3.426	2.867	2.969	4.104	3.320	0.002	20.2	18.1	17.9
	3.400	2.878	2.966	4.164	3.349	0.002	20.2	18.1	17.9

Appendix E

Results

E.1 Gas and Liquid Superficial Velocities and Flow Patterns

E.2 Measured Void Fraction

E.3 Measured Frictional Pressure Gradients

Appendix E.1

Gas and Liquid Superficial Velocities and Flow Patterns

Legend

Observed flow patterns	
SW	Stratified wavy flow
P	plug flow
S	Slug flow
A	Annular flow

Run No.	J_0 (m/s)	J_1 (m/s)	Flow Pattern	Descriptions and Comments
45	0.373	0.002	SW	Slight waves, small amplitude, long wave-length
46	0.916	0.023	SW	Interface somewhat rough. Very occasional slugs
47	2.034	0.069	SW	Occasional slug. Surface wavier before slug
48	3.036	0.051	SW	Similar to # 47. Some droplets on surface above liquid
49	7.612	0.057	SW	Wavy rough surface. Occasional slug. Increased number of droplets on surface above liquid
50	12.865	0.057	SW	Wavy rough surface. No slugs. Longer droplets on upper surface.
51	15.344	0.002	SW	Similar to # 50. More and longer drops on upper surface
52	19.326	0.017	SW	Similar to # 51. Increased numbers of droplets on upper surface.
53	23.346	0.016	A	Similar to # 52. Increased number of droplets on upper surface. Approaching annular or droplet.
54	22.984	0.023	A	Droplets on surface.
55	21.258	0.202	A	Vertical and horizontal views have same appearance. Occasional roll waves. Droplets.
56	20.939	0.278	A	Annular flow. Similar to # 55.
57	19.981	0.431	A	Annular flow, similar to # 56.
58	18.613	0.599	A	Annular flow with occasional roll waves.
59	16.811	0.822	A	Annular flow with roll waves.
60	4.200	0.890	P	Medium to large gaps between plugs. Rough interface.
61	2.016	0.901	P	Similar to #60.
62	1.672	0.911	P	Similar to #61. Some waviness.
63	0.665	0.918	P	Similar to #62. High frequency of plugs.
64	0.095	0.931	P	High frequency scalloped plugs.
67	0.807	0.831	P	Long air plugs, high frequency. Some bubbles in water between slugs.
68	1.629	0.829	P	Similar to #68. Near Plug to slug transition.
69	0.457	0.839	P	Plugs scalloped, thin, variable length, have high frequency
70	0.454	0.901	P	Similar to #69.

Run No.	J_g (m/s)	J_l (m/s)	Flow Pattern	Descriptions and Comments
71	1.707	0.894	S	Bubbles in liquid slugs. Plug to Slug transition.
72	0.793	0.259	P	Smooth air plugs some very long. No bubbles between plugs (elongated bubbles).
73	0.792	0.145	P	Extremely long plugs, approaching smooth stratified. Long water gaps between slugs. Droplets left on upper surface after water gap passes. Stratified wavy to plug flow transition.
74	0.787	0.279	P	Like #73 but plugs shorter and gaps shorter. Occasional bubble in water gaps.
75	0.701	0.526	P	Plugs and gaps shorter. Interface level lower than for #74. Small scallops of plugs.
76	0.751	0.915	P	High frequency scalloped plugs, short gaps, no bubbles in gaps.
77	1.607	0.917	S	Bubbles in slugs. Rough interface. Slug to wavy transition.
78	2.220	0.911	S	Bubbles in slugs. Rough interface.
79	4.346	0.901	S	Bubbles in slugs. Rough interface.
80	8.781	0.854	S	Similar to # 79, slug to annular flow transition.
81	0.221	0.201	P	Plugs thin, occasionally quite long. Interface smooth. Small plugs occasionally break off tail of one plug, picked up by nose of next one.
82	0.218	0.477	P	Plugs more uniform & scalloped, no long ones, depth varies. Tails narrow & shallow.
83	0.214	0.837	P	Plugs shallow and highly scalloped and closely spaced.
84	0.215	0.915	P	Like # 83 except plugs longer & more frequent. Very occasionally long gap between slugs.
85	0.191	0.266	P	Plugs with small scallops near nose of elongated bubbles, interface smooth. Long gaps of variable length. Occasional small bubble in gaps.
86	0.189	0.657	P	Plugs shorter, higher frequency, more scalloped than in #85, with some long gaps
87	0.19	0.896	P	Like #86, with trends enhanced. Plugs shallow
88	0.188	1.061	P	Scalloped plugs approach separate bubbles, but still some grouping of bubbles
89	0.181	1.076	P	Same as # 88, scalloped flow.
90	1.048	0.810	P	Plugs much smoother and longer and deeper than in #88 & 89. Droplets on upper surface during plug passage.
91	1.566	0.809	P	Similar to #90, interface more wavy, more smaller bubbles between plugs. Approaching slug flow.
92	1.448	1.022	P	Similar to #91, with rougher interface. Approaching slug flow.
93	0.695	0.893	P	Long, deep scalloped plugs with rough interface. Few bubbles between plugs.

Run No.	J_0 (m/s)	J_i (m/s)	Flow Pattern	Descriptions and Comments
94	0.998	0.149	P	Very long, deep plugs with smooth interface. Small gaps with almost smooth stratified. Occasional droplets on upper surface in plug regions.
95	0.995	0.221	P	Long, deep plug with smooth interfaces. Some short plugs.
96	0.987	0.748	P	Shorter and deeper plugs @ higher frequency. More uniform plug length with scallops beginning and rougher interfaces. Some small bubbles between plugs.
97	2.417	0.741	S	Long distances between slugs, with low and rough interface. Aerated liquid slugs.
98	4.033	0.743	S	Long distances between slugs. Very rough, low interface, smooths out after slug passes. Approach to annular flow in slug flow.
99	6.611	0.733	S	Similar to #98. Approach to annular flow in slugs.
100	10.362	0.797	S	Long slugs, very bubbly in slugs. Rough interface between slugs. Closer to annular flow in slugs.
101	13.464	0.811	S	Long slugs, shorter gaps between slugs with rough interfaces. Approaching annular.
102	1.454	0.227	P	Plugs long and deep and very smooth. Short gaps with every few bubbles. Onset of slug flow.
103	3.740	0.242	S	Plugs long, almost stratified flow. Bubbles in intermittent water slugs. Interface rough after slug passes and then smooths out. Onset of slug flow.
104	5.950	0.236	S	Similar to #103. Plugs deeper. Interface rougher but still smooths out after slug passes. Onset of slug flow.
105	13.139	0.235	S	Slugs very bubbly. Low, rough interface between slugs. Plug to slug transition.
106	12.296	0.437	S	Slugs larger. Approaching annular flow?
107	0.224	0.275	P	Plugs long, smooth, and well defined. Plug length variable. Some scallops near nose, tail elongated towards top and sometimes asymmetrical in horizontal plane. Occasional small bubbles between plugs.
108	1.365	0.275	P	Like #107 but plugs longer & deeper and gaps shorter. Nose rougher than rest. More small bubbles in gaps.
109	1.402	0.273	P	Very similar to # 108. Plugs long.
110	1.313	0.499	P	Similar to #109 but higher frequency of gaps. Some droplets on top surface.
111	1.304	0.796	P	Similar to #110 but smaller, more frequent gaps. Bottom of plugs rougher, wavier. More bubbles in gaps. Approaching slug flow?
112	1.292	0.951	S	Continues trend of #108 to #111
113	1.278	1.026	S	Many bubbles in slugs, higher slug frequency. Scallops beginning in plug regions?
114	1.248	1.122	S	Similar to #113
115	1.266	1.173	S	Similar to #114.

Run No.	J_0 (m/s)	J_i (m/s)	Flow Pattern	Descriptions and Comments
116	1.268	1.173	S	Same as #115.
117	5.201	0.142	S	Infrequent slugs, almost stratified wavy. Interface rough after slug passes but smooths out.
118	5.062	0.207	S	Higher frequency of slugs, frequency somewhat random. Slugs are highly aerated.
119	5.054	0.311	S	Similar to #118. Sometimes two slugs close together. Interface smooth just before slug.
120	4.986	0.502	S	Slugs more frequent and random. Many bubbles in slugs. Interface smooth before slugs.
121	4.762	0.710	S	Slugs more frequent still, at higher speed, behavior otherwise similar to #120. Occasional long spaces between slugs.
122	4.646	0.824	S	Similar to #121. Low interface initially rough after slug, then smooths out.
123	4.434	1.068	S	Similar to #122. Slug velocity very high and interface always rough.
124	4.368	1.173	S	Similar to #123.
125	8.786	1.168	S	Slugs very fast and bubbly. Very rough interface becomes smoother before next slug. Liquid level between slugs high.
126	17.637	1.092	A	Still appears to have horizontal interface between slugs, but may be annular flow. Vertical and horizontal views similar. Slug (roll waves) very fast. Slug to Annular transition.
127	22.430	1.061	A	Vertical and horizontal views similar. Fast roll waves. Wispy annular?
128	21.349	1.070	A	Same as # 127
129	22.447	0.731	A	Same as #128.
130	26.810	0.293	A	Same as #129.
131	30.062	0.002	A	Vertical and horizontal views same, with waves evident.
132	27.645	0.173	A	Similar to #130.
133	25.525	0.357	A	Similar to #132.
134	24.593	0.481	A	Same as #133.
135	23.821	0.553	A	Same as #134
136	22.316	0.674	A	Same as #135.

Table E.2: Measured local void fraction using the upstream and downstream sensors.

Run	Upstream sensor (r/R = -0.8, -0.6, -0.4, -0.2, 0, 0.2, 0.4, 0.6, 0.8)									avg.	Overall avg.
	Downstream sensor (r/R = -0.8, -0.6, -0.4, -0.2, 0, 0.2, 0.4, 0.6, 0.8)									avg.	
Wavy flow regime											
45	0	0	0	0.363	1	1	1	1	1	0.609	0.648
	0	0	0	1	0.986	1	1	1	1	0.686	
46	0	0	0	0.8744	0.9788	1	1	1	1	0.670	0.678
	0	0	0	0.9937	1	1	1	1	1	0.687	
47	0	0	0.2892	0.9881	0.9949	0.9984	0.9978	0.9946	0.9935	0.718	0.723
	0	0	0.3714	0.9919	0.9975	0.9991	0.9988	0.9955	0.9942	0.729	
48	0	0.001	0.8503	0.9826	0.9923	0.9959	0.9984	0.9994	1	0.783	0.793
	0	0.0823	0.9428	0.987	0.9944	0.9989	0.9996	0.9999	1	0.804	
49	0.0004	0.663	0.9465	0.9917	0.999	1	1	1	1	0.864	0.884
	0.1962	0.8262	0.9743	0.9957	0.9996	1	1	1	1	0.903	
50	0.1837	0.8088	0.9846	0.9972	0.9986	1	1	1	1	0.901	0.928
	0.607	0.9175	0.9947	0.9998	0.9998	1	1	1	1	0.954	
51	0.4837	0.5356	1	1	1	1	1	1	1	0.904	0.933
	0.7969	0.8179	1	1	1	1	1	1	1	0.962	
52	0.4723	0.5338	1	1	1	1	1	1	1	0.903	0.943
	0.9259	0.8952	1	1	1	1	1	1	1	0.982	
Plug flow regime											
63	0	0	0.001	0.114	0.535	0.666	0.690	0.668	0.637	0.373	0.398
	0	0.001	0.002	0.341	0.595	0.667	0.694	0.697	0.733	0.422	
64	0	0	0	0	0.125	0.296	0.359	0.434	0.449	0.181	0.186
	0	0	0	0	0.170	0.339	0.364	0.461	0.417	0.192	
67	0	0	0	0.067	0.067	0.427	0.467	0.489	0.491	0.220	0.241
	0	0	0	0.215	0.215	0.454	0.472	0.490	0.484	0.261	
68	0	0.001	0.004	0.442	0.611	0.699	0.691	0.702	0.725	0.440	0.456
	0.001	0.001	0.114	0.537	0.661	0.700	0.692	0.708	0.732	0.473	
69	0	0	0	0.016	0.033	0.077	0.101	0.532	0.718	0.149	0.163
	0	0	0	0.040	0.047	0.094	0.159	0.674	0.726	0.178	
70	0	0	0	0	0.020	0.059	0.090	0.116	0.116	0.043	0.053
	0	0	0	0.001	0.030	0.073	0.107	0.189	0.189	0.062	
72	0.004	0.002	0.001	0.185	0.613	0.679	0.679	0.687	0.648	0.396	0.404
	0	0	0	0.385	0.547	0.685	0.680	0.692	0.648	0.413	
73	0.010	0.010	0.008	0.549	0.687	0.481	0.786	0.698	0.737	0.450	0.479
	0	0	0	0.756	0.702	0.775	0.770	0.682	0.725	0.507	
74	0	0	0	0	0.657	0.591	0.625	0.704	0.680	0.365	0.383
	0	0	0	0.218	0.657	0.654	0.629	0.708	0.680	0.401	
75	0	0	0	0.007	0.385	0.455	0.528	0.585	0.594	0.283	0.298
	0	0	0	0.144	0.428	0.474	0.548	0.599	0.596	0.312	
76	0	0	0	0	0.168	0.342	0.431	0.437	0.451	0.201	0.209
	0	0	0	0.010	0.230	0.396	0.440	0.440	0.449	0.218	
81	0	0	0	0	0.331	0.218	0.391	0.612	0.334	0.208	0.218
	0	0	0	0.120	0.310	0.336	0.296	0.307	0.713	0.229	
82	0	0	0	0.009	0.046	0.197	0.255	0.354	0.371	0.132	0.146
	0	0	0	0.094	0.161	0.205	0.324	0.344	0.317	0.160	
83	0	0	0	0.017	0.017	0.028	0.201	0.529	0.409	0.123	0.130
	0	0	0	0.021	0.020	0.059	0.286	0.543	0.377	0.136	

Run	Upstream sensor (r/R = -0.8, -0.6, -0.4, -0.2, 0, 0.2, 0.4, 0.6, 0.8)									avg.	Overall avg.
	Downstream sensor (r/R = -0.8, -0.6, -0.4, -0.2, 0, 0.2, 0.4, 0.6, 0.8)									avg.	
84	0	0	0	0	0	0	0.162	0.582	0.652	0.139	0.150
	0	0	0	0	0	0.001	0.323	0.621	0.641	0.161	
85	0	0	0	0	0	0	0.045	0.288	0.345	0.067	0.068
	0.005	0.005	0.005	0.005	0.005	0.011	0.073	0.303	0.282	0.070	
86	0	0	0	0	0	0	0.022	0.127	0.191	0.033	0.035
	0	0	0	0	0	0	0.050	0.124	0.187	0.036	
87	0	0	0	0	0	0	0	0.153	0.240	0.038	0.041
	0	0	0	0	0	0	0.017	0.186	0.245	0.044	
88	0	0	0	0	0	0	0.002	0.078	0.214	0.028	0.031
	0	0	0	0	0	0	0.011	0.105	0.230	0.034	
89	0	0	0	0	0	0	0.001	0.013	0.153	0.016	0.018
	0	0	0	0	0	0	0.003	0.036	0.182	0.021	
90	0	0	0	0.167	0.435	0.551	0.577	0.566	0.572	0.323	0.337
	0	0	0	0.344	0.475	0.553	0.578	0.567	0.580	0.352	
91	0	0.001	0.002	0.376	0.586	0.682	0.685	0.682	0.703	0.422	0.432
	0	0.001	0.040	0.498	0.633	0.639	0.679	0.685	0.717	0.443	
92	0	0	0.002	0.191	0.508	0.625	0.644	0.650	0.657	0.369	0.384
	0	0	0.019	0.363	0.557	0.608	0.641	0.656	0.669	0.398	
93	0	0	0	0.008	0.206	0.394	0.427	0.470	0.464	0.217	0.227
	0	0	0	0.093	0.255	0.434	0.407	0.476	0.461	0.237	
94	0	0	0.840	0.072	0.735	0.813	0.914	0.865	0.807	0.572	0.594
	0	0.237	0.900	0.533	0.532	0.809	0.914	0.691	0.806	0.616	
95	0	0.520	0.574	0.789	0.841	0.780	0.745	0.747	0.770	0.657	0.673
	0.066	0.662	0.747	0.737	0.831	0.784	0.748	0.749	0.758	0.689	
96	0	0	0.053	0.424	0.552	0.563	0.556	0.593	0.602	0.381	0.394
	0	0	0.265	0.450	0.579	0.565	0.557	0.597	0.540	0.407	
102	0	0	0.252	0.772	0.854	0.856	0.865	0.847	0.819	0.604	0.606
	0.001	0.001	0.491	0.805	0.836	0.854	0.849	0.757	0.657	0.608	
Annular flow regime											
53	0.538	0.989	1	1	1	1	1	1	1	0.955	0.970
	0.828	1	1	1	1	1	1	1	1	0.984	
54	0.538	0.989	1	1	1	1	1	1	1	0.955	0.970
	0.828	1	1	1	1	1	1	1	1	0.984	
55	0.160	1	0.905	1	1	0.969	0.973	1	0.986	0.902	0.930
	0.566	1	0.987	1	1	1	1	1	1	0.958	
56	0.204	0.127	1	1	1	1	0.968	1	1	0.833	0.842
	0.331	0.164	1	1	1	0.986	1	1	1	0.851	
57	0.118	0.209	0.514	0.969	1	1	0.997	0.976	0.964	0.770	0.774
	0.134	0.114	0.652	0.916	1	1	1	1	1	0.778	
58	0.293	0.100	1	1	1	1	1	1	1	0.842	0.854
	0.360	0.277	1	1	1	1	1	1	1	0.867	
59	0.371	0.153	0.451	1	1	1	1	1	1	0.791	0.807
	0.227	0.416	0.617	1	1	1	1	1	1	0.823	
Slug flow regime											
60	0.053	0.029	0.340	0.688	0.916	0.947	0.943	0.951	0.940	0.662	0.696
	0.100	0.069	0.644	0.835	0.929	0.952	0.954	0.960	0.949	0.730	

Run	Upstream sensor ($r/R = -0.8, -0.6, -0.4, -0.2, 0, 0.2, 0.4, 0.6, 0.8$)									avg.	Overall avg.
	Downstream sensor ($r/R = -0.8, -0.6, -0.4, -0.2, 0, 0.2, 0.4, 0.6, 0.8$)									avg.	
61	0.014	0.003	0.025	0.658	0.788	0.779	0.810	0.893	0.870	0.551	0.565
	0.020	0.013	0.046	0.771	0.808	0.820	0.832	0.903	0.872	0.580	
62	0.006	0.002	0.013	0.460	0.731	0.841	0.848	0.873	0.851	0.525	0.540
	0.011	0.006	0.043	0.624	0.770	0.845	0.851	0.878	0.856	0.556	
71	0	0	0.004	0.353	0.626	0.697	0.710	0.714	0.722	0.434	0.447
	0	0.002	0.038	0.487	0.670	0.697	0.708	0.717	0.727	0.461	
77	0	0	0.001	0.200	0.530	0.643	0.669	0.684	0.668	0.382	0.399
	0	0.001	0.005	0.404	0.579	0.652	0.666	0.683	0.666	0.415	
78	0.001	0.001	0.005	0.263	0.684	0.765	0.773	0.766	0.805	0.459	0.472
	0.001	0.002	0.008	0.459	0.730	0.759	0.758	0.761	0.793	0.486	
79	0.014	0.006	0.039	0.496	0.867	0.874	0.896	0.890	0.897	0.566	0.579
	0.017	0.010	0.092	0.695	0.865	0.858	0.884	0.877	0.882	0.591	
80	0.034	0.143	0.708	0.943	0.957	0.957	0.958	0.953	0.938	0.755	0.769
	0.060	0.283	0.843	0.953	0.957	0.949	0.948	0.940	0.916	0.783	
97	0.001	0.007	0.420	0.706	0.811	0.798	0.800	0.836	0.829	0.596	0.581
	0.004	0.014	0.568	0.680	0.630	0.756	0.773	0.835	0.676	0.565	
98	0.009	0.014	0.621	0.825	0.878	0.897	0.899	0.907	0.890	0.681	0.640
	0.011	0.051	0.764	0.787	0.605	0.695	0.684	0.847	0.810	0.598	
99	0.012	0.037	0.643	0.901	0.935	0.940	0.930	0.920	0.925	0.716	0.730
	0.027	0.119	0.774	0.926	0.938	0.938	0.920	0.914	0.921	0.743	
100	0.038	0.300	0.745	0.892	0.935	0.953	0.960	0.926	0.933	0.763	0.620
	0.063	0.500	0.731	0.658	0.463	0.422	0.376	0.449	0.569	0.478	
101	0.102	0.419	0.828	0.967	0.970	0.963	0.976	0.972	0.936	0.812	0.856
	0.345	0.744	0.969	0.990	0.994	0.985	0.986	0.988	0.975	0.900	
102	0	0	0.252	0.772	0.854	0.856	0.865	0.847	0.819	0.604	0.606
	0.001	0.001	0.491	0.805	0.836	0.854	0.849	0.757	0.657	0.608	
103	0	0.007	0.698	0.912	0.943	0.970	0.989	1.000	0.984	0.745	0.743
	0.001	0.012	0.745	0.962	0.946	0.964	0.918	0.936	0.964	0.741	
104	0.001	0.055	0.557	0.954	0.979	0.987	0.990	0.997	0.992	0.746	0.744
	0.004	0.316	0.794	0.968	0.984	0.894	0.897	0.958	0.625	0.742	
105	0.072	0.569	0.885	0.979	0.996	0.996	0.999	0.998	1.000	0.852	0.878
	0.330	0.753	0.942	0.994	0.999	0.999	1.000	1.000	0.995	0.904	
106	0.076	0.394	0.876	0.955	0.981	0.985	0.987	0.983	0.979	0.822	0.624
	0.169	0.470	0.675	0.732	0.268	0.408	0.312	0.298	0.446	0.427	

Note: As discussed in Section 4.5, void fraction measurements were taken for 60 experimental test runs only. Neither of the digital oscilloscopes was available to record the output signals of the optical probe for the remaining test runs (test runs numbered 107 to 136).

Table E.3: Measured Frictional Pressure Gradients.

Test	Flow	Measured Pressure Gradients			(dP/dx)
Run	Regime	P_1 and P_3	P_1 and P_2	P_2 and P_3	Average
#		kPa/m	kPa/m	kPa/m	kPa/m
45	SW	(*)	0.012	(*)	0.012*
46	SW	(*)	0.020	(*)	0.020*
47	SW	(*)	0.054	(*)	0.054*
48	SW	(*)	0.129	(*)	0.129*
49	SW	0.182	0.197	0.135	0.171
50	SW	0.335	0.352	0.281	0.323
51	SW	0.454	0.471	0.401	0.442
52	SW	0.685	0.717	0.585	0.663
53	A	0.961	0.955	0.983	0.966
54	A	0.937	0.932	0.954	0.941
55	A	2.119	2.098	2.183	2.133
56	A	2.566	2.545	2.632	2.581
57	A	3.389	3.255	3.807	3.484
58	A	4.589	4.523	4.795	4.636
59	A	5.973	5.893	6.221	6.029
60	S	3.006	3.104	2.698	2.936
61	S	1.695	1.794	1.385	1.625
62	S	1.514	1.609	1.217	1.447
63	P	0.926	0.855	1.149	0.977
64	P	0.612	0.467	1.066	0.715
67	P	0.621	0.647	0.540	0.602
68	P	0.882	0.904	0.815	0.867
69	P	0.517	0.543	0.436	0.499
70	P	0.564	0.591	0.479	0.545
71	S	1.010	1.041	0.913	0.988
72	P	0.111	0.123	0.072	0.102
73	P	(*)	0.039	(*)	0.039*
74	P	0.116	0.129	0.076	0.107
75	P	0.308	0.330	0.238	0.292
76	P	0.693	0.718	0.614	0.675
77	S	0.993	1.023	0.898	0.971
78	S	1.222	1.257	1.113	1.198
79	S	1.897	2.032	1.478	1.802
80	S	2.811	2.894	2.553	2.753
81	P	(*)	0.081	(*)	0.081*
82	P	0.189	0.243	0.019	0.150
83	P	0.470	0.504	0.365	0.446
84	P	0.560	0.592	0.460	0.538

Test	Flow	Measured Pressure Gradients			(dP/dx)
Run	Regime	P ₁ and P ₃	P ₁ and P ₂	P ₂ and P ₃	Average
#		kPa/m	kPa/m	kPa/m	kPa/m
85	P	0.060	0.063	0.051	0.058
86	P	0.276	0.285	0.247	0.269
87	P	0.476	0.485	0.448	0.470
88	P	0.639	0.654	0.592	0.628
89	P	0.617	0.630	0.576	0.608
90	P	0.675	0.683	0.652	0.670
91	P	0.851	0.860	0.820	0.844
92	P	1.151	1.159	1.123	1.145
93	P	0.694	0.707	0.653	0.685
94	P	(*)	0.059	(*)	0.059*
95	P	0.112	0.102	0.140	0.118
96	P	0.596	0.610	0.556	0.587
97	S	1.003	1.021	0.945	0.990
98	S	1.378	1.426	1.227	1.344
99	S	1.851	1.794	2.032	1.892
100	S	3.134	3.218	2.873	3.075
101	S	3.961	4.048	3.691	3.900
102	P	0.154	0.156	0.149	0.153
103	S	0.218	0.236	0.161	0.205
104	S	0.362	0.246	0.722	0.443
105	S	0.832	0.871	0.709	0.804
106	S	1.697	1.770	1.468	1.645
107	P	(*)	0.076	(*)	0.076*
108	P	0.150	0.176	0.071	0.132
109	P	0.155	0.180	0.077	0.137
110	P	0.371	0.398	0.284	0.351
111	P	0.706	0.739	0.602	0.682
112	S	0.900	0.931	0.803	0.878
113	S	1.027	1.063	0.914	1.001
114	S	1.153	1.188	1.043	1.128
115	S	1.293	1.333	1.168	1.265
116	S	1.338	1.375	1.224	1.313
117	S	0.169	0.210	0.041	0.140
118	S	0.285	0.334	0.134	0.251
119	S	0.524	0.536	0.486	0.515
120	S	1.005	0.989	1.055	1.016
121	S	1.428	1.414	1.472	1.438
122	S	1.702	1.736	1.596	1.678
123	S	2.209	2.191	2.263	2.221
124	S	2.610	2.602	2.635	2.616

Test	Flow	Measured Pressure Gradients			(dP/dx)
Run	Regime	P_1 and P_3	P_1 and P_2	P_2 and P_3	Average
#		kPa/m	kPa/m	kPa/m	kPa/m
125	S	4.256	4.340	3.995	4.197
126	A	7.313	7.444	6.902	7.219
127	A	7.313	7.057	8.112	7.494
128	A	7.313	6.969	8.386	7.556
129	A	6.609	6.421	7.198	6.743
130	A	3.331	2.977	4.436	3.582
131	A	1.172	0.937	1.907	1.339
132	A	2.610	2.304	3.567	2.827
133	A	3.720	3.344	4.895	3.986
134	A	4.510	4.120	5.729	4.786
135	A	5.033	4.606	6.367	5.335
136	A	6.033	5.736	6.961	6.243

Note: (*) As discussed in Section 6.3.1, readings of pressure transducer at the exit of the test section were discarded for these test runs. As a result, the average pressures were determined from the measured static pressures at Port 1 and Port 2, i.e., P_1 and P_2 .