

**QUANTIFYING THE INFLUENCE OF AIR ON
THE CAPACITY OF LARGE DIAMETER WATER
PIPELINES AND DEVELOPING PROVISIONAL
GUIDELINES FOR EFFECTIVE DE-AERATION**

VOLUME 1

**QUANTIFYING THE INFLUENCE OF AIR ON THE
CAPACITY OF LARGE DIAMETER WATER
PIPELINES**

SJ van Vuuren • M van Dijk • JN Steenkamp

WRC Report No. 1177/1/04



Water Research Commission



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Volume 1

**Quantifying the influence of air on the
Capacity of large diameter water pipelines**

**Report to the
WATER RESEARCH COMMISSION**

by

S J van Vuuren, M van Dijk and J N Steenkamp

**WRC Report No 1177/1/04
ISBN No 1-77005-154-6
ISBN Set No 1-77005-153-8**

MARCH 2004

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EXECUTIVE SUMMARY

1. Background and motivation

The geographically mismatch of the water demand centers and the water resources necessitate the transport of water over long distances and high elevation differences. On average water is transported about 350 km in South Africa (Basson, Van Niekerk & Van Rooyen, 1997). High energy costs and the increasing demand requires that the water transfer infrastructure should function optimally. To ensure operational effectiveness of pipelines they must be effectively de-aerated.

The influence of air on the capacity of pipelines has long been a cause for concern. The quantification thereof was, however, unknown and required investigation. The investigation of air in pipelines required a review of available literature, experimental work, field investigations and the development of provisional guidelines for optimal design to limit the negative effects of air.

2. Aims of this study

The aims of this study were to:

- Provide a literature review with regard to aspects related to air in water pipelines
- Conduct experimental and field work to establish to influence of air in pipelines and
- Compile a user guide for the design of effective de-aerated pipelines.

3. Research Findings

An extensive literature review has been conducted from which valuable information have been gathered and used in the design and set-up of experimental and field investigations.

The literature confirmed that flow velocity; air bubble size and the down slope of the pipeline are the main contributing factors, which determine whether the air can be removed hydraulically.

In the experimental work different air bubble sizes at different velocities and different pipe angles were tested to evaluate the available relationships predicting the required flow velocity to hydraulically transport air. New relationships were determined for different bubble sizes and operational conditions that can be used to determine the required positions where mechanical removal of air is required.

The findings are included in the documentation in a simple relationship with respect to the pipe diameter and pipe slope that can be used to determine if air will be transported.

Figure i is an example of the relationship derived for a pipe with a diameter of 110 mm.

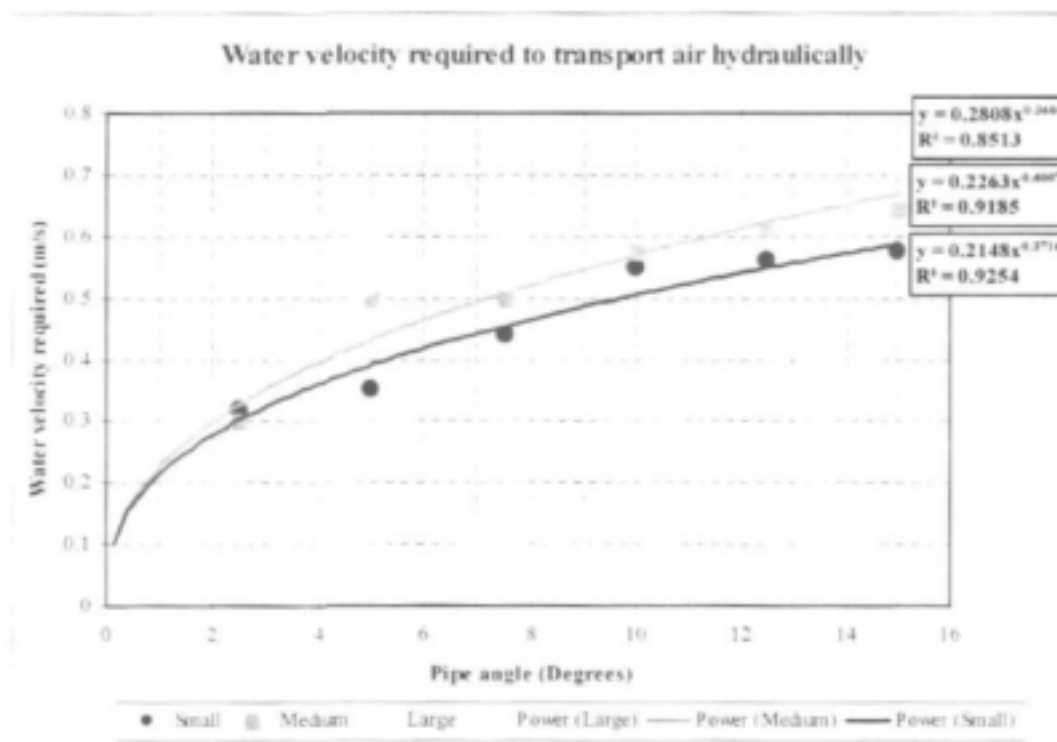


Figure i: Flow velocities determined from an experimental set-up for a 110 mm diameter pipe to transport different sizes of air bubbles hydraulically

The proposed relationship that was obtained is of the following form:

$$V_{min} = a\sqrt{gD} \theta^b$$

Where:

V_{min} = Minimum velocity required to transport air (m/s)

g = Gravitational acceleration (m/s^2)

D = Internal pipe diameter (m)

a = Constant value (experimentally determined = 0.2178)

- θ = Absolute value of slope of the pipe measured from the horizontal (degrees)
- b = Constant value (experimentally determined = 0.4007)

Due to the erratic flow path of air bubbles that are transported along a pipeline it was necessary to derive a relationship for the required discontinuity that should be created to capture the air for mechanical removal from the pipeline. Experimental work and numerical analyses (CFD Modelling) were used to compare the effectiveness of a specific size of discontinuity to capture the air that is transported along the pipe. The required size for effective de-aeration is dependent on the flow velocity, size of air bubble and the pipe slope. This is an important finding because it has been reported that although systems have been equipped with sufficient air valves air was still trapped in the pipeline. **Figure ii** reflects the relationship of the efficiency to capture the air against the relative size for the off-take for different air bubble sizes.

These experimental results reflected that to effectively de-aerate a pipeline the size of the discontinuity should be at least 35% of the pipe diameter. The required discontinuity is a function of flow velocity, pipe diameter and slope and driven by economic considerations. The suggested discontinuity has been set as a function of pipe diameter; see Section 8 in Volume 2.

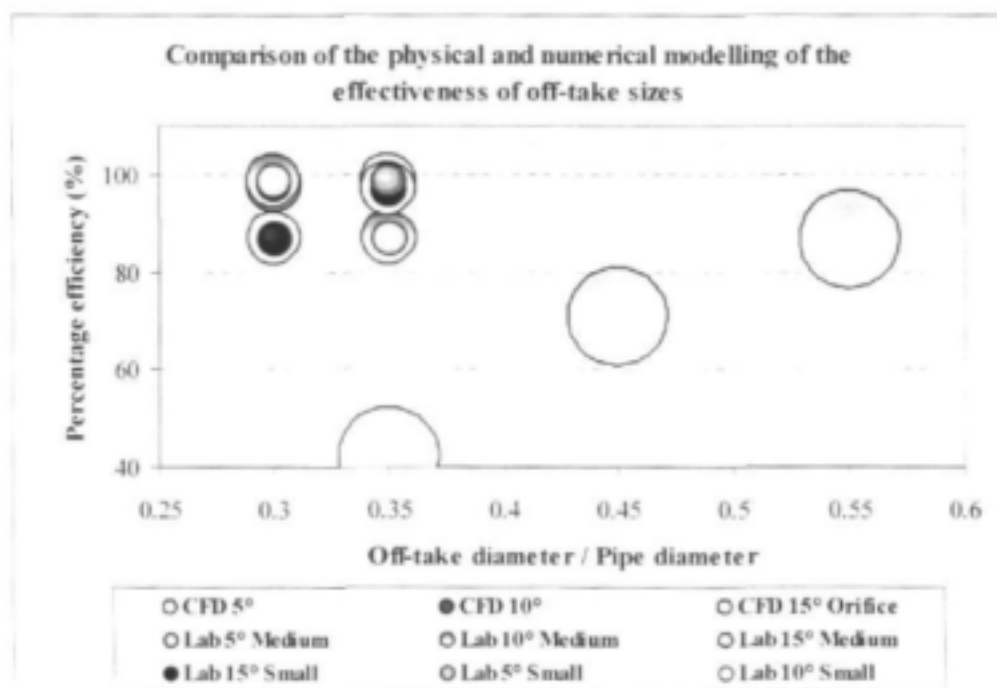


Figure ii: Relationship of the efficiency to capture the air against the relative size of the off-take for different air bubble sizes

A practical problem for pipeline operators remains to establish if the pipeline has been effectively de-aerated. The procedure of "Inverse Transients" (movement of negative pressure transients waves) were investigated to determine if it could be used as a means to determine if there are unknown leaks in a pipe system. This procedure requires the induction of a pressure wave and the accurate recording of the movement and change of the induced pressure wave as it propagates through the system. An experimental set-up with a 200 mm steel pipe was used with a fast acting valve installed at the end of the line, to create the negative wave. Recordings of the pressure variation were conducted from which the influence of the leak on the pressure wave was determined. The experimental results reflected that the procedure could be used if the leak is more than about 5 % of the flow in the pipeline. **Figure iii** reflects the results and suggests that the procedure has potential and needs to be developed further as a means to detect leaks and air bubbles in a pipeline.

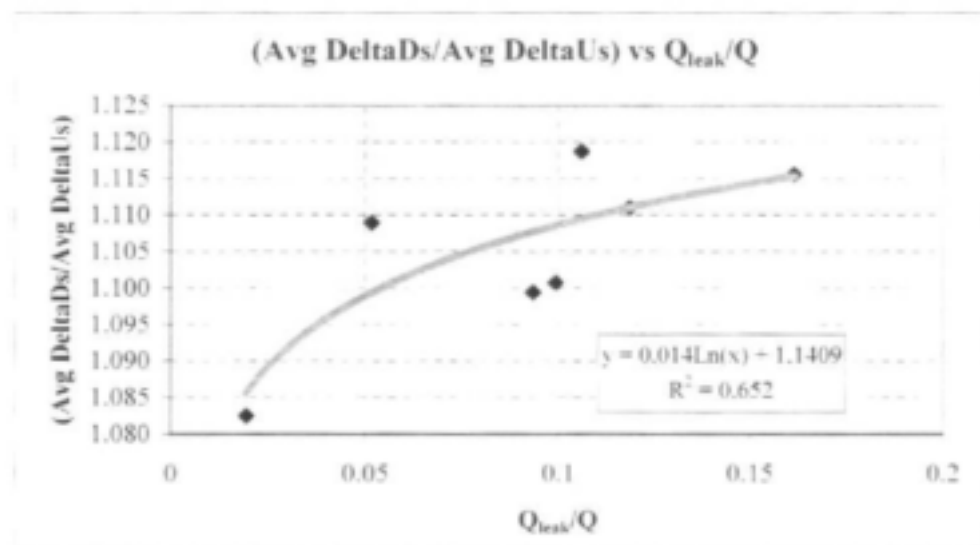


Figure iii: Inverse pressure fluctuation as a function of the size of the leakage

The energy loss associated with free air that is stuck at an apex point, influence the hydraulic capacity of the system and experimental work have been conducted to define the losses. An apex was constructed out of transparent pipe with a 160 mm diameter, which was used for a number of experiments. A relationship was first determined for the specific layout of the relationship for the secondary losses for the bend. Various bubble sizes were then injected at the apex point and the energy loss resulting from the air bubbles was measured. The difference in the secondary loss between the two tests was assigned to the energy loss originated from the bubble at the apex point. **Figure iv** reflects the pressure drop that results from the captured bubbles of different sizes at an apex with an angle of 11.25° . The pressure

drop was used to calculate a secondary loss coefficient, k (Figure v) and it was found for a specific size air bubble, the secondary loss coefficient, k changed due to the change in the form of the bubble. The secondary loss coefficient decreases for a specific bubble size as the velocity increases indicating that the influence of the air bubble on the flow area is more significant than the length of the air bubble.

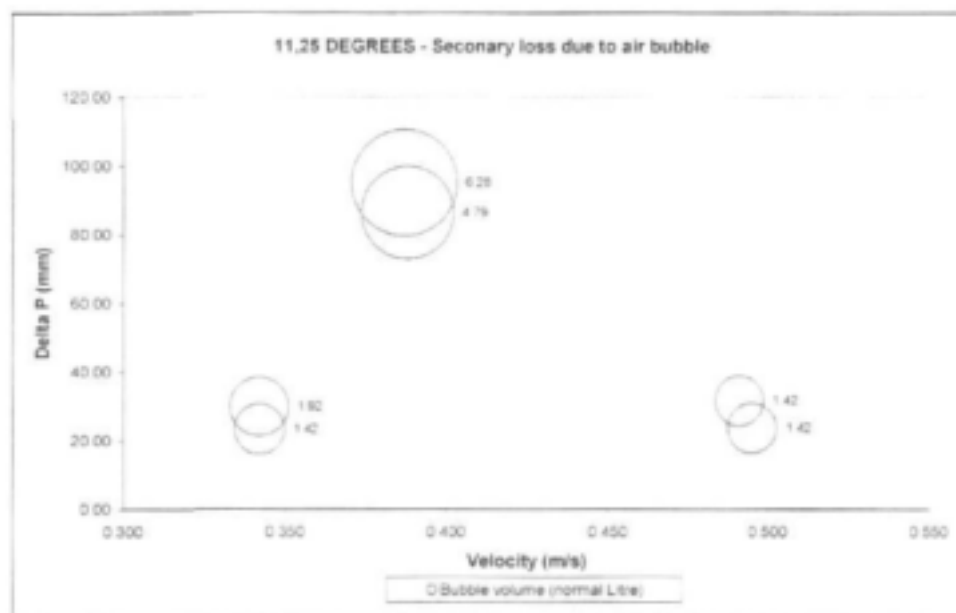


Figure iv: Energy loss associated with different air bubble sizes at an apex point (11,25°)

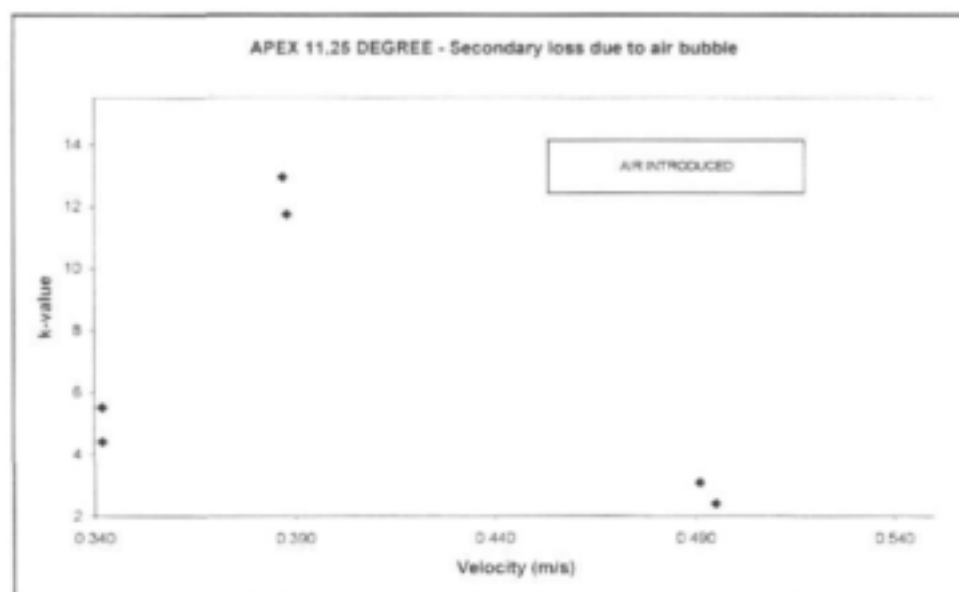


Figure v: Calculated secondary loss coefficient, k due to the energy loss associated with different air bubble sizes at an apex point (11,25°)

The experimental work was backed by field investigations that were performed on pipelines of Umgeni Water, Rand Water and the Buffalo River Raw water supply line.

The Buffalo River Raw Water supply line was recently constructed and major problems were experienced. These problems were related to air in the line. Visual inspection as well as computer analysis confirmed that air was the reason for the loss in capacity and unstable flow. The origin of the air was the negative pressures on the suction line that intruded air into the pumps for which remedial measures were suggested for implementation.

The pump line from the Hazelmere/LaMercy bifurcation to Avondale reservoir was investigated to determine the transportability of air, the effective capturing of the air and the influence of an air bubble on the propagation of a transient wave. Air was introduced into the line and tests were performed to establish if it could be transported down the descending section of the line to the next air valve. Figure vi is a schematic profile of the section that was evaluated. Air was introduced at an air valve at point A (chainage 8 531m) while the air valve at point C (chainage 8 936 m) was isolated and the release from the next air valve at chainage 9 728 m (point E) was monitored to determine if the air was being transported downstream.

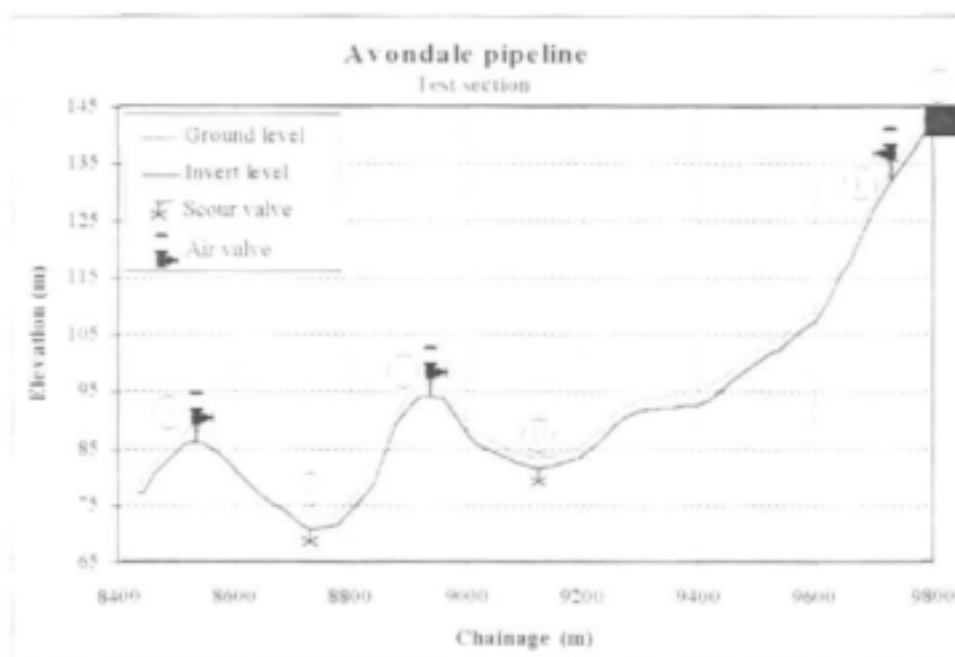


Figure vi: Section used for the test on the Hazelmere - Ballito (Avondale) pipeline

No air that was introduced at the induction point (point A) was transported down the line towards the observation point (point E). This result was significant because it reflected that with a flow velocity of 0.75 m/s in the 700 mm pipeline, it was not possible to transport the air down a slope of 14.3% (the steepest section).

Analyses of the entire Hazelmere - Ballito (Avondale) pipeline reflected that a total length of approximately 3.2 km of the 9.8 km long pipeline cannot transport the air hydraulically at a flow velocity of 0.75 m/s. This was determined with the newly developed relationship for transportability of air; see Section 4.2 of this report. The design of this pipeline however provided equal-T off takes to the accumulators underneath the air valves from where the air is released in a controlled manner.

A quantitative assessment was made of the air valve installation chambers, where equal T-pieces had been used and sufficient accumulation volume underneath the air valve had been provided, led to the conclusion that the de-aeration capacity of the pipeline is effective. This is a conservative design and if the accumulator is big enough, all the air will be captured and released.

To establish if air was stuck in the pipeline, air was introduced at chainage 8 528.50 m (Figure vi, point A) and a fast acting valve was installed on the scour valve at chainage 9 120.22 m (point D). Pressure recordings were taken at points A and E, upstream and downstream respectively from the point where the surge wave was created. One set of captured data, however, indicated that free air in the system did influence the wave speed celerity between point D and A. During the course of the project, difficulties were experienced with regard to the introduction of air into the pipeline, as well as the capturing and measuring of air released from existing pipelines under operating conditions and that for the bigger pipelines of Rand Water the field investigations had to be changed. Numerical modelling was included to establish if it could be used as an alternative to the physical field tests.

Rand Water upgraded the B8 pipeline by installing the B16 pipeline in parallel to the B8 pipeline. This created the opportunity to investigate the influence of air on the capacity of the B8 pipeline, which was operating approximately 15% below its designed capacity. The survey of the pipeline, constructed during 1976, showed serious shortcomings in its de-aeration features (**Figure vii**). The lack of accumulators below the air valves and insufficient discontinuity reduced the capacity of the existing air valves considerably. Air valves with accumulators will be placed on critical positions to improve the de-aeration of the line B8 pipeline. Problems with the completion of the B16 pipeline made access to the B8 impossible and field tests had to be delayed beyond the time scope of this project. The tests will, however, be conducted with Rand Water and the results should be available in future.



Figure vii: Typical 100 mm off-take (stub) underneath the air valve

Based on the laboratory results and the field tests, provisional guidelines were prepared, describing the influence of air on the capacity of large diameter pipes and providing details to ensure effective de-aeration. **Figure viii** reflects the proposed layout for effective de-aeration for pipelines. Furthermore an air valve sizing and positioning (ASAP) procedure has been developed and incorporated into utility software for the determination of air valve sizes and locations. These programs are freely available and a link to the suppliers could be established through the following web site:

<http://www.up.ac.za/academic/civil/divisions/water/airvalves.html>

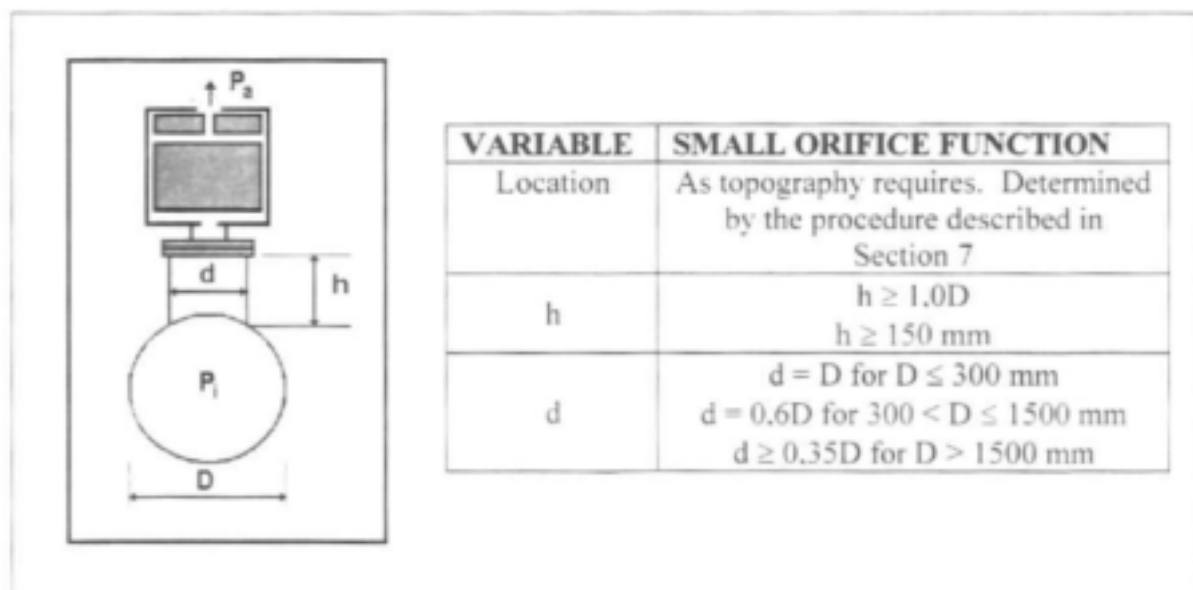


Figure viii: Proposed layout for effective de-aeration

At the end of the research it was felt that efforts should be made to further develop the procedure of inverse transients as a tool to determine the position of leaks and air in pipe systems. The results from this research will be made available through the Guidelines, which is contained in Volume 2: "Provisional Guidelines for the effective de-aeration of large diameter water pipelines". Feedback for further improvements to the procedures will be forthcoming from the use of the Provisional Guidelines.

4. Further research

It is recommended that in light of the aging water infrastructure and increasing losses:

- The use of inverse transients should be further developed as a tool to establish the operational characteristics of the system. The procedure should be developed to such a level that it can be used to evaluate a pipeline, mainly with the objective of determining the effectiveness of de-aeration equipment as well as the identification of leakages.
- Due to the difficulty that was experienced with the field tests additional numerical modelling be conducted on the transportation and capturing of air at the discontinuities that are provided for air valves. This will indicate if any scale effects might have to be considered for the application of normative guidelines on large diameter pipelines.

**QUANTIFYING THE INFLUENCE OF AIR ON THE CAPACITY OF LARGE
DIAMETER WATER PIPELINES AND DEVELOPING PROVISIONAL GUIDELINES
FOR EFFECTIVE DE-AERATION
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LIST OF SYMBOLS

a	$=$	Constant value (experimentally determined)
A	$=$	Area of the pipe (m^2)
A_a	$=$	Area on the air-inlet boundary (m^2)
A_p	$=$	Area of the pipe (m^2)
A_i	$=$	Flow area for air inflow (m^2)
$A_{L,1}$	$=$	Liquid area at location 1 (m^2)
A_o	$=$	Flow area for air outflow (m^2)
$A_{rupture}$	$=$	Area of the rupture (m^2)
b	$=$	Bubble size parameter
b	$=$	Constant value (experimentally determined)
Bt	$=$	Bubble time (s)
c	$=$	Wave celerity (m/s)
C_B	$=$	Velocity of the back of the slug (m/s)
C_D	$=$	Discharge coefficient
C_{d_i}	$=$	Discharge coefficient for the inlet area of the air valve
C_{d_o}	$=$	Discharge coefficient for the inlet area of the air valve
C_F	$=$	Velocity of the front of the slug moving downstream (m/s)
C_T	$=$	Velocity of the tail of the slug (m/s)
D	$=$	Inside diameter of pipe (m)
D	$=$	Distance travelled by the bubble in a specific time (m)
d_s	$=$	Maximum air bubble size, (m)
D_{outlet}	$=$	Inside diameter of outlet pipe (m)
F_{RB}	$=$	Bubble Froude number
g	$=$	Gravitational acceleration (m/s^2)
H	$=$	Hydrostatic head (m)
H_L	$=$	Liquid hold-up (m)
L_B	$=$	Length of the bubble (m)
k	$=$	Isentropic constant
L_B	$=$	Length of the bubble (m)
L_S	$=$	Bubble length (m)
m	$=$	Mass of water (kg)
$\dot{m}$	$=$	Mass flow rate (kg/s)
P	$=$	Absolute pressure (kPa)

p	=	Absolute pressure head (m)
P_a	=	Atmospheric pressure (m)
p_a	=	Absolute pressure head in the air valve (m)
P_1	=	Pressure of air at point 1
P_2	=	Pressure of air at point 2
Q	=	Flow rate (m^3/s)
Q_{burst}	=	Flow rate due to burst (m^3/s)
Q_c	=	critical discharge (m^3/s)
Q_{drain}	=	Flow rate due to controlled drainage (m^3/s)
Q_L	=	Required flow rate (m^3/s)
R	=	Gas constant (J/kg/K)
R	=	Radius of pipe (m)
Re_L	=	Reynolds number
s	=	Distance between the irregularity and the measuring point (m)
S_0	=	Slope of pipe (m/m)
T	=	Temperature ($^{\circ}\text{K}$)
t	=	Time interval (s)
t	=	Time taken for the bubble to travel a distance (s)
V	=	Volume of water (m^3)
V_1	=	Volume of air at point 1
V_2	=	Volume of air at point 2
V_{Bubble}	=	Calculated bubble velocity (m/s)
V_c	=	Clearing velocity (m/s)
V_{C1}	=	Velocity in the liquid film ahead of the slug (m/s)
V_{fe}	=	Velocity at the end of the liquid film (m/s)
V_g	=	Gas velocity in stagnant liquid (m/s)
V_g	=	Drift velocity at stagnant liquid conditions (m/s)
V_{g0}	=	Bubble velocity in stagnant liquid (m/s)
V_{GS}	=	superficial velocity of the gas (m/s)
V_L	=	Superficial liquid velocity (m/s)
V_L	=	Average liquid velocity (m/s)
V_L	=	Critical velocity required for the removal of air (m/s)
V_{L1}	=	Liquid velocity at section 1 (m/s).
$V_{L(\text{end slug})}$	=	velocity of the liquid at the end (upstream) of the slug (m/s)
V_{LS}	=	Superficial velocity of the liquid (m/s)

V_m	=	Velocity of the mixture (m/s)
V_{min}	=	Minimum velocity required to transport air (m/s)
V_s	=	Sweeping velocity (m/s)
V_T	=	Translational velocity (m/s)
Vol	=	Volume of the air pocket
We	=	Webber number
We_{crit}	=	Critical Weber number
σ	=	Inter-facial surface tension (N/m)
Δh	=	Available energy between peak and drainage point (m)
Δh	=	Change in pressure (m)
ΔP	=	Energy loss (mm)
λ	=	Friction factor
ϕ	=	Slope of the pipe (degrees)
θ	=	Pipe slope (degrees)
ξ	=	Slope factor
ρ_a	=	Density of the air in the air valve under the line head conditions (kg/m ³)
ρ_g	=	Gas density (kg/m ³ (dispersed phase))
ρ_L	=	Liquid density (kg/m ³ (continuous phase))
μ_L	=	Liquid viscosity (kg/ms (continuous phase))
ν_L	=	Kinematic liquid viscosity
τ	=	Stress at the bubble surface (N/m)
τ_w	=	Average resisting stress at the wall (Pa)

ACKNOWLEDGEMENTS

The research described in this report emanated from a project funded by the Water Research Commission entitled:

QUANTIFYING THE INFLUENCE OF AIR ON THE CAPACITY OF LARGE DIAMETER WATER PIPELINES AND DEVELOPING PROVISIONAL GUIDELINES FOR EFFECTIVE DE-AERATION

The research findings presented in this report emanated from a project funded by the Water Research Commission (WRC), whose support is acknowledged with gratitude. The Steering Committee made an important contribution in steering and guiding this project to obtain the format and results reflected in this report. The support of the Chairman and Manager of this project, Mr Jay Baghwan, as well as the staff of WRC, is greatly appreciated.

The Steering Committee and Project Team responsible for this project consisted of the following persons:

Mr JN Bhagwan	:	Water Research Commission – Chairman
Ms U Wium	:	Water Research Commission (Committee Secretary)
Mr B Mokgonyana	:	Water Research Commission (Committee Secretary)
Mr R Boekhoud	:	Vent-O-Mat (Pty) Ltd
Mr T de Ridder	:	Lepelle Northern Water
Mr AR Kockott	:	Umgeni Water
Prof GGS Pegram	:	University of Natal
Dr MJ Shand	:	Ninham Shand (Pty) Ltd
Mr N Trebicki	:	Rand Water
Mr DJ Winzer	:	Rand Water
Prof SJ van Vuuren	:	University of Pretoria
Mr M van Dijk	:	University of Pretoria
Mr JN Steenkamp	:	Ninham Shand (Pty) Ltd

This project was made possible with the co-operation of many individuals and institutions. The authors therefore wish to record their sincere thanks to the following:

Mr Gerrit Otto (Department Water Affairs and Forestry), whose support to the students at the DWAF laboratories is highly valued.

Mr Don Hogkinson (Umgeni Water), who provided access to portions of their distribution system for tests.

Mr Alan Kockott (Umgeni Water), who assisted with the fieldwork and mobilized a number of their operational staff who assisted and gave valuable input.

Mr Nic Trebicki (Rand Water), who shared valuable information with us and assisted in the field tests that were attempted on lines of Rand Water.

Messrs Kevin Richards and Allistair Balitto from Vent-O-Mat, who supplied equipment used for the field work and shared valuable experience with the project team.

The project team, whom were prepared to explore new ground and in the process endeavour some unsuccessful options, was greatly appreciated. The contributions of the members of the project team are acknowledged. They are:

Messrs Marco van Dijk and Jaco Steenkamp, as well as a number of students from the University of Pretoria, who undertook the experimental work and assisted in the experimental and field work are namely:

Messrs Thomas Sinden, Org van Rensburg, Eben Beetge, Francois Lewis, Andrew Lea, Hilton Guerra and Hoffman Munro, as well as Ms Mari du Toit and Ms Annemie Kotze.

QUANTIFYING THE INFLUENCE OF AIR ON THE CAPACITY OF LARGE DIAMETER WATER PIPELINES AND DEVELOPING PROVISIONAL GUIDELINES FOR EFFECTIVE DE-AERATION

1. INTRODUCTION

1.1 Aim of the study

The aim of the study was:

- To quantify the influence of air on the capacity of large diameter water pipelines, and
- To develop provisional guidelines for effective de-aeration.

1.2 Methodology

The approach that was used was grouped under three main activities relating to the investigation:

- Literature review
- Experimental work and
- Field investigations.

These aspects of the research are highlighted below.

1.2.1 Literature study

The literature review consisted of a scan of the latest trends and developments at international and national level, regarding:

- Hydraulic transportation of air in pipelines.
- The stability of dissolved air in water under varying temperature, pressure and turbulence.
- The influence on transient waves by localized air bubbles in a pipeline.

The review provided an advantage in understanding the behavior of air in pipes, as well as in the planning and design of the field and experimental investigations of this study.

1.2.2 Field investigation

Pipelines of Rand Water and Umgeni Water were identified for the field investigations.

The hydraulic capacity for the pipelines were firstly obtained from operational data, designs documentation or measurements. On the longitudinal profile the critical sections with large negative slopes were identified and it was evaluated whether the operational flow rate was sufficient to hydraulically transport the air.

The Hazelmere – Ballito (Umgeni Water) pump line (700 mm diameter steel pipeline, with Copon Epoxy lining) was investigated to determine whether the design made sufficient provision for the capturing of the air at the installed accumulators. The operational experience showed that the pipeline is operating at a capacity below the design capacity, raising the question whether air in the pipeline has a possible influence.

The Rand Water B8 pipeline (2100 mm Steel pipeline with a cement mortar lining), constructed in 1976, was selected as the second pipeline to be evaluated during this research. The availability of this line for internal inspection was due to the completion of the B16 pipeline, as this runs parallel to the last section (about 30 km) of the B8 pipeline. The B8 pipeline was designed in accordance with outdated procedures with regard to the positioning and location of air valves.

Lack of sufficient and accurate as-built information necessitated a detailed investigation of all the air valve chambers on this pipeline. The characteristics and installation information were recorded.

The delay in the testing and commissioning of the B16 pipeline, delayed the installation of pressure tappings and field measurements on the B8 pipeline. The initial programme included a measurement of the B8 line prior to the rehabilitation and refurbishment of the pipeline followed by a re-evaluation of the pipeline after refurbishment. Time constraints then forced the project team to include the B16 pipeline that runs parallel to the B8 line and has a similar longitudinal profile. It was argued that the inclusion of the B16 and B8 lines and a comparison of the B16 results with those of the B8 pipeline would provide similar results to take measurements on the B8 pipeline, pre and post-refurbishment.

Details of all the measurements, results and conclusions are discussed in Section 3 of this report.

1.2.3 Experimental work

Laboratory tests were set up and designed to obtain information and to establish relationships for the following aspects:

- Movement of various size air bubbles to establish the relationship between the pipe slope and the required flow velocity that will hydraulically transport the air.
- The discontinuity required in the crown of the pipe to release the air from the pipeline under operating conditions and capture the air, was determined.
- The influence of air bubbles on transient waves was used to establish whether inverse transients could be used to detect leaks and to determine the presence of free air in pipelines.
- The energy loss associated with a localized air bubble was evaluated.

Details of all the experimental work are discussed in Section 4 of this report.

1.2.4 Development of Provisional Guidelines

Results from the experimental work and field measurements were used to develop provisional guidelines with regard to the effective de-aeration of pipelines. The provisional guidelines also include utility design software for the sizing and location of air valves. This software is freely available from air valve manufacturers.

1.3 Layout of the report

This Report was divided into two volumes, each consisting of a number of chapters and where applicable, supporting Appendices containing data captured during the research. A Provisional Guideline for effective de-aeration of pipelines (Volume 2) was compiled in such a way that it can be read as a stand-alone document. This approach resulted in the compilation of the following two volumes, which reflect the findings of this research:

- Volume 1: Quantifying the influence of air on the capacity of large diameter water pipelines
- Volume 2: Provisional guidelines for the effective de-aeration of large diameter water pipelines

The contents of Volume 1 (This report - Main Report) comprises:

- Introduction
- Influence of air on pipelines
- Field investigations
- Experimental work
- Numerical modeling of air release from pipelines (CFD)
- Effective de-aeration of pipelines
- Further developments.

Volume 2, includes the following sections:

- Introduction
- Intrusion of air into pipelines
- The consequences of air in pipelines
- Hydraulic transport of air
- Air valves
- Implementation of a new pipeline
- Sizing and positioning of air valves
- Typical installation details of air valves
- User comments

All the collected data, video clips, photos and analyses have been compiled on a CD, which is available from the project leader, Prof SJ van Vuuren, or on the following web site:

<http://www.up.ac.za/academic/civil/divisions/water/airvalves.html>

2. INFLUENCE OF AIR IN PIPELINES

2.1 Introduction

The objective of Section 2 was to review existing literature related to the effects of air in water pipelines. Although the flow in water mains is supposed to be single phased, wherever sufficient free air is available in pipelines, it is possible that it will function as a two phase system. This necessitates reference to two phased flow phenomena (water-air).

The effect of air is summarized under different headings reflecting the following aspects:

- Ways in which free air can enter water systems – Section 2.2
- Solubility of air in water – Section 2.3
- Influence of air on the wave-speed – Section 2.4
- Influence of air on hydraulic capacity – Section 2.5
- Transportation of air bubbles in water pipelines – Section 2.6
- Induced pressures resulting from air release – Section 2.7
- Options to effectively de-aerate a pipeline – Section 2.8
- Positioning of air-valves – Section 2.9
- Determination of the intake capacity of air valves – Section 2.10
- Utility software for determining the sizing and location of air valves – Section 2.11
- Procedures for the sizing and location of air valves – Section 2.12
- Detection of slug flow in pipelines – Section 2.13
- Bubble characteristics – Section 2.14
- Pressure drop across an air bubble – Section 2.15
- Bubble dynamics in closed conduit flow – Section 2.16
- Economics of ineffective de-aeration – Section 2.17

In the following sections the original notation used for variables have, where possible, been changed to provide consistency in this document.

2.2 Ways in which free air can enter water systems

It is commonly found in practice that little or no reference is made to the presence of air when dealing with water distribution systems. Although it is known that there will always be contact between water and air, either in the storage reservoirs or even in the pipeline, where unwanted air accumulated or was allowed into the system through air valves or vacuum breakers installed to prevent negative pressures. Other ways of air intrusion have also been identified (Van Vuuren, 1990), for example: air intrusion through pump or reservoir intakes, ineffective filling of the pipeline etc. Some of these circumstances can greatly be eliminated if sound engineering practice is applied.

Under normal operating conditions air and water are two extremely different fluids, one being a gas, the other a liquid. Air has a density of 0.12% that of water and can be viewed as an ideal gas, which is compressible, while water is incompressible. In general, whenever a mixture of water and air is present, based on the premise that water is incompressible, the mixture (air and water) has distinctly different characteristics compared to that of water alone, especially in terms of bulk modules. It has been documented (Wylie and Streeter, 1978) that it is not unusual for water to hold up to 3%, by volume, of air in solution. This dissolved air will be released when the pressure drops or when the temperature rises, and will be present as air bubbles. Dissolving the air bubbles requires a substantial increase in the head and agitation (increasing the surface/volume ratio) through which the process is driven. The rate of dissolving the air under normal operating conditions and even under surge pressures, is much slower than the rate at which the air can be released (Zlecka and Keller, 1990) from the mixture.

2.3 Solubility of air in water

The authors (Delfos et al. 2001) investigated the stability of a Taylor bubble (large air bubble transported at about the mean flow velocity in the pipe) in a vertical tube with a diameter of 100 mm. The research objective was to obtain insight of the physical mechanisms of gas entrainment and re-coalescence.

Three different Taylor bubble Froude numbers were investigated:

$$Fr_B = \frac{V_L}{V_g} \dots\dots\dots (2.1)$$

Where:

V_L = superficial liquid velocity

V_g = gas velocity in stagnant liquid

Froude numbers, $Fr_B = 1.0$; 1.2; and 1.4 were investigated.

From the observations, at high flow velocities, the frequency observed at the bottom of the Taylor bubble was similar to the results produced by Polonsky, Barnea and Shemer (1999) and was about 3Hz.

It was found that there was almost a linear relationship between the gas loss and the length of the Taylor bubble. This relationship is referred to as the gas loss curve, $V_{SG(Z)}$. It was also found that beyond a certain Taylor bubble length the gas loss decreased and then stabilized at a value (cm/s), about 10% lower than the maximum value.

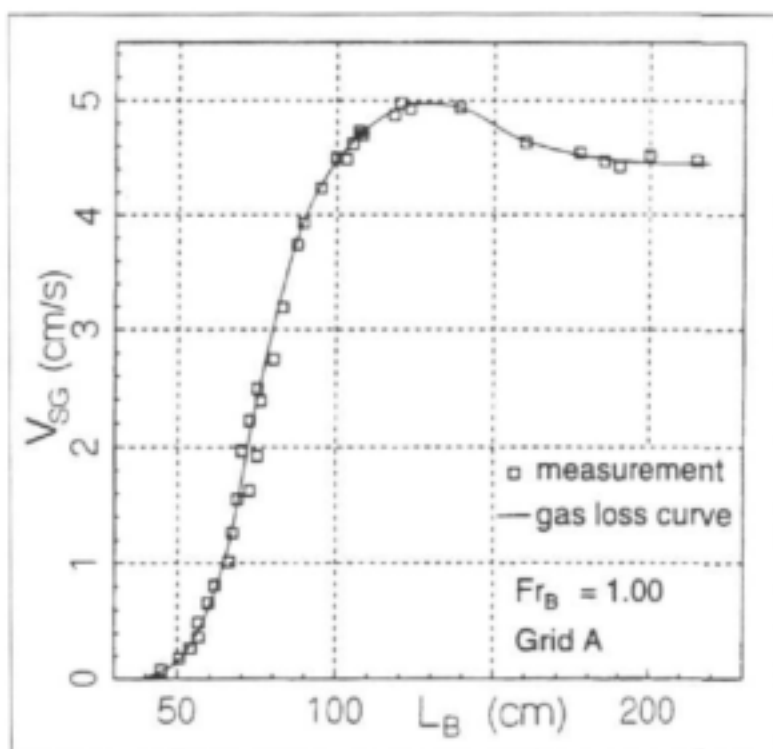


Figure 2.1: Air/Gas loss relationship for a Taylor type bubble

The void fraction in the liquid column showed the same relationship as the air/gas loss curve. For higher Taylor bubble Froude numbers, $Fr_B = \frac{V_L^2}{g L_B}$ it was established that the air loss increases, although the void ratio for a given bubble length, L_B , varying from 25 to 200 mm, is constant.

The researchers concluded that:

- gas loss from a Taylor bubble increases with increasing liquid flow rate.
- the void fraction in the liquid column below the bubble only slightly depends upon Fr_B

2.4 Influence of air on the wave-speed

Due to the compressibility of air the reduction of the wave speed of water/air mixture is dramatically reduced by the increase of free air. **Figure 2.2** (Wylie and Streeter, 1978) reflects the wave speed in a pipe for an air-water mixture and $(K/E)(D/e) = 0,263$.

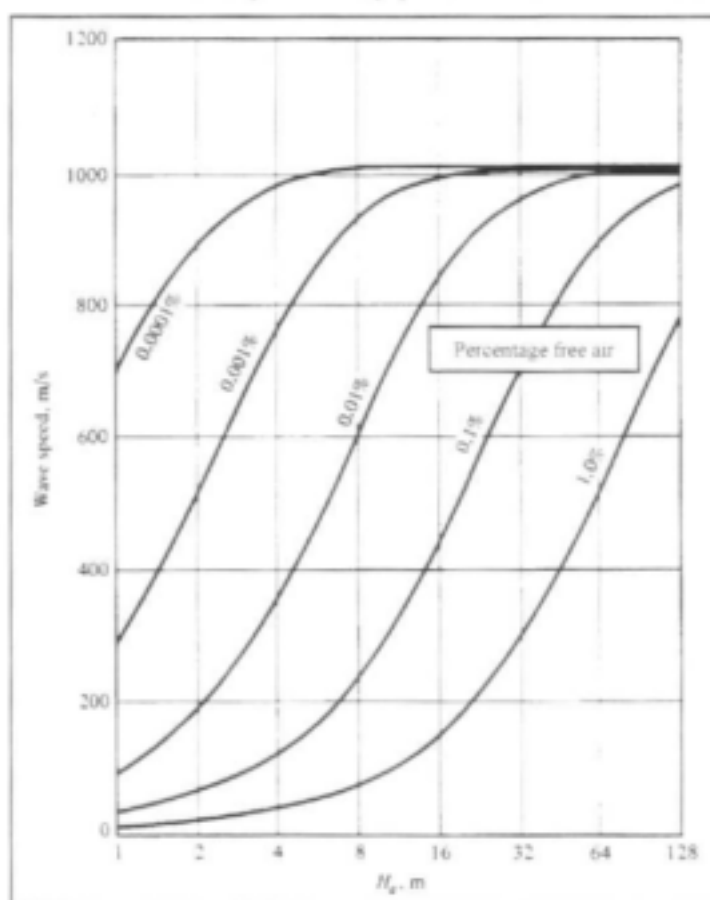


Figure 2.2: Wave speed for a for an air-water mixture

2.5 Influence of air on the hydraulic capacity of pipelines

Now, if it is accepted that free air is present in water distribution networks, it becomes necessary to define the influence of free air on the performance of the networks.

Distinction has to be made between normal operating conditions and surge conditions (high and sub-atmospheric pressures) when evaluating the influence of air in water systems. During normal operation, the presence of free air will have a negative effect on the discharge capacity of the system and the objective therefore is to ensure that all the free air can be removed from the system, either mechanically or hydraulically (Van Vuuren, 1990). During transient conditions, however, the presence of free air is an advantage, because it decreases the wave celerity (Wylie and Streeter, 1978). This ability of free air to reduce induced pressure spikes in a confined system by compressing the air (deformation energy is required) is implemented in the design of air vessels to reduce induced pressures. In concept, it can therefore be assumed that if large quantities of air are present in a pipeline, the surge pressures will be reduced. It is also true however that the large quantities of air, utilized to dampen the surge pressures, should be removed from the pipeline at a certain stage. The most appropriate positions for the release of free air would be at the apex points along the pipeline where air valves are normally situated. Some air valve manufacturers realized the need to discharge large volumes of air from pipelines during the filling of the pipeline and re-designed their air valves, in some cases including special devices like "dynamic shields" to prevent the valve from "blow shut". This, however, resulted in high-induced pressures, which can be associated with the deceleration of the water column(s) when all the air has been expelled or when the valve eventually closes. The misconception of having a valve able to handle high discharge rates, led to confusion and negatively influenced the optimal use of air release valves (Van Vuuren, 1992).

Given a specified air volume and internal pressure conditions, research has indicated that it is possible to define an opening size through which air can be released, while at the same time the induced pressure can be restricted to a predetermined maximum value (van Vuuren, 1989).

The anti-shock type (3-Stage) air valve utilizes these phenomena to provide an air-vacuum breaker valve, which will reduce the induced pressure surges by restricting the discharge rate of air from the pipeline when a predetermined differential pressure has been reached. This is obtained by switching to a smaller (intermediate or the second release) orifice while air is being released from the system. An air cushion is thus created in the pipeline, which will dampen the pressure wave. The size of the second orifice is such that it will limit the induced pressure, associated with the closure of the air valve when all the air has been expelled, to acceptable levels

A mathematical model, for the assessment of pressure head maxima that air pockets in a pipeline can originate on start-up, has been developed by Fuertes et al. (1999).

The authors (Fuertes et al. 1999) selected an irregular profile with a number of entrapped air pockets. By assuming a vertical face between the air and water, and treating the water as a rigid column, relationships for the pressure increase was derived. The Suter curves for pumps were used to obtain a generalized pump behaviour during start-up and rundown of the pumps.

Differential equations for the movement of the inter-face between the water and air, the velocity of the water columns and the pressure were derived. The profile will define the number of air pockets (n) and the number of water blocking columns (m). For a simplified system of two entrapped air columns ($n=2$), the maxima pressure and velocity was calculated.

The author concluded that pipelines with entrapped air pockets, must be filled very slowly and hence the flow supplied by the pump should be restricted.

2.6 Transportation of air bubbles in water pipelines

The movement of large air pockets in pipelines has been subject to a number of investigations falling into the following categories:

- Rising velocity of an air pocket, V_{g0} , in closed conduit when the water is stationary or moving for a range of different conduit inclinations.
- The water velocity, V_L to sweep the air pocket downstream with the flow, for various pipe inclinations.

Although surface tension, inertia forces viscosity and buoyancy forces influence the movement of air in water carrying conduits, it has been shown that in the civil engineering context, only the inertia forces and buoyancy forces are significant. **Wisner et al. (1975) have indicated that viscous effects were dictated up to Reynolds numbers, (VD/ν) of up to 10^5 .** For the case where buoyancy and inertia forces are dominant it follows from the balance of forces.

$$\frac{1}{2} \rho_L C_d A_p V_L^2 = (\rho_L - \rho_g) g Vol \dots\dots\dots (2.2)$$

Where:

- C_d = Discharge coefficient
- A_p = Area of the pipe (m^2)
- V_L = Velocity of the water (m/s)
- Vol = Volume of the air bubble
- $\rho_L - \rho_g \approx \rho_L$

The air pocket and A_p is proportional to the square of the diameter of the bubble, d_p^2 and the volume, Vol , is proportional to d_p^3 .

The water velocity to keep the air pocket in equilibrium is:

$$V_L = K_1 \sqrt{g d_p} \dots\dots\dots (2.3)$$

Where K_1 is a constant.

If it is assumed that the size of the air pocket is proportional to the conduit diameter, then the rise velocity of the air pocket would be:

$$V_{g0} = K \sqrt{g D} \dots\dots\dots (2.4)$$

The rise velocity of air in a vertical conduit has been determined by a number of researchers. (Davies and Taylor, 1936). (Nicklen, 1962). If it is assumed that:

$$V_{g0} = 0.35 \frac{(\rho_l - \rho_g)}{\rho_l} \sqrt{gD} \dots\dots\dots (2.5)$$

with $\frac{(\rho_l - \rho_g)}{\rho_l} \approx 1$, then it follows that:

$$V_{g0} = 0.3 \sqrt{gD} \dots\dots\dots (2.6)$$

In the case of upward moving water in a vertical pipe the air bubble velocity was determined by Martin, (1976).

$$V_g = C_0 \left[\frac{Q_g + Q_l}{A} \right] + C_1 \sqrt{gD} \dots\dots\dots (2.7)$$

For stationary conditions $C_1 = 0.35$ but it increased to 0.495 if the air pocket is ascending in a moving water column. C_0 was found to be approximately equal to 1.2. With β representing the air/water ratio it follows that:

$$V_g = 1.2 V_l (1 + \beta) + C_1 \sqrt{gD} \dots\dots\dots (2.8)$$

According to Martin (1976), maintaining the air bubble, the minimum downward velocity of the water is when:

$$1.2 V_l (1 + \beta) = 0.35 \sqrt{gD} \dots\dots\dots (2.9)$$

$$\therefore \frac{V_l}{\sqrt{gD}} \geq \frac{0.35}{1.2(1 + \beta)}$$

The following was shown by Mechler (1966), that near the air pocket on the downstream side of the pocket.

$$\frac{V_l}{\sqrt{gD}} > 0.4 \text{ to } 0.6 \dots\dots\dots (2.10)$$

Research on the rising velocity of air in an inclined conduit, has been carried out by Zukoski (1966), Bonnicore et al (1971) Runge and Wallis 1965) and Wisner (1975).

It was indicated that the maximum rise velocity usually corresponds with an angle of between 30° to 60°. The maximum velocity was found to be about 60% more than the velocity in a vertical pipe, and that the maximum velocity was experienced at 40°.

This phenomenon can be explained in terms of the balance of buoyancy, drag and surface tension forces on the air pocket. In an inclined pipe the air pocket changed to a more streamlined body, reducing the drag on the body. Although the buoyancy force also decreases, the net balance of forces results in an increased flow velocity of the pocket.

Bonnicare et al (1971) deduced the following relationship.

$$\frac{V_L}{0.35\sqrt{gD}} = \sqrt{3.43 \cos \phi + \sin \phi} \dots\dots\dots(2.11)$$

Where:

ϕ = angle with the horizontal (degrees)

The clearing (sweeping) velocity to remove an air pocket has been investigated by several authors (Mechler (1966), Kent (1952), Ahmed et al (1984)). When their findings are plotted in a non-dimensional form of $V_L/\sqrt{gD}$ against the angle ϕ it can be noted:

The clearing velocity to remove air pockets is generally larger than the rising velocity of air pockets in stationary water. The clearing velocity increases with relatively larger air pockets, denoted in **Figure 2.3** by **n**, which is the ratio of the air pocket volume to a representative conduit volume $\pi D^3/4$. V_L is independent of n, for values of $n > 0.8$.

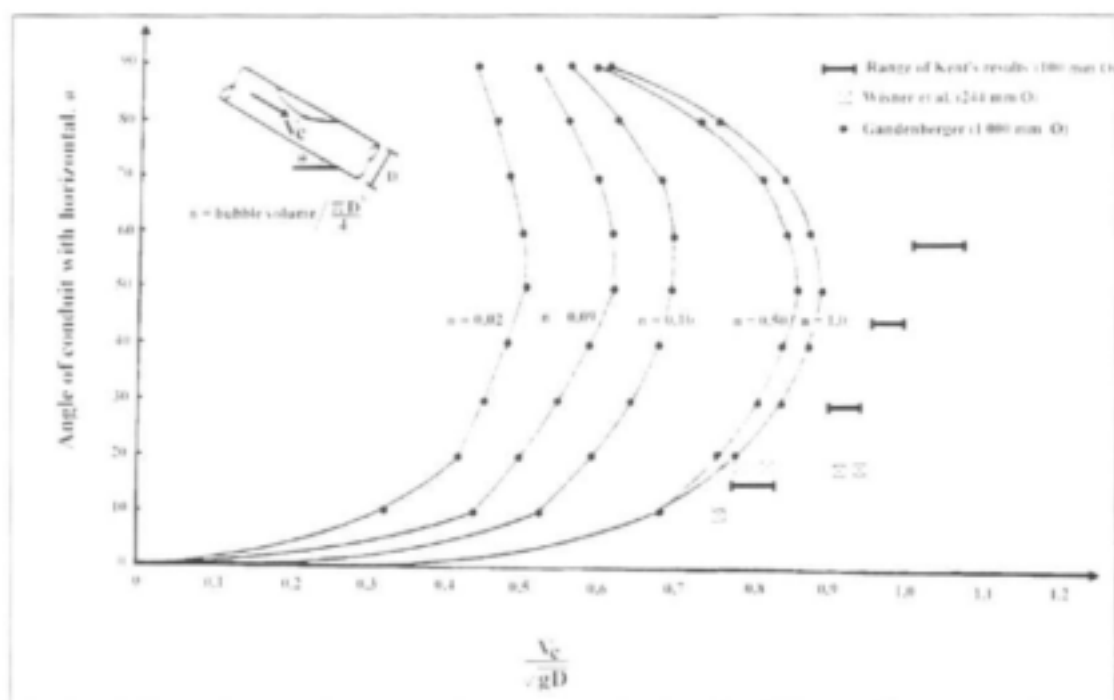


Figure 2.3: Clearing velocity for the removal of air from a pipeline

Gandenberger's results indicate a maximum clearing velocity required at a conduit angle of $50^\circ - 60^\circ$. Kent's results do not indicate any such maximum value, probably due to the fact that the value of θ was not increased beyond 60° . Kent proposed a relationship for clearing water velocities in the form.

$$V_L = 1.62 \sqrt{\xi} \sqrt{gD \sin \phi} \dots\dots\dots (2.12)$$

where the value of ξ is 0.58 when the pocket reaches a certain size ($L_p/D > 1.5$)

It can therefore be concluded that for a given air pocket size (denoted by n), the clearing velocity is again a Froude scaled phenomenon.

Referring to Wisner's work (Wisner et al. 1975) it can be stated that the water velocity required to clear an air pocket can be given by:

$$\frac{V_L}{\sqrt{gD}} = f(\theta, L_p/D, n, R_f, W_c) \dots\dots\dots (2.13)$$

Where θ , L_b/D and n can be identical in model and prototype structures. Re , the Reynolds number, may have an influence up to a value of 10^5 and We , the Weber number, is only significant in pipes under 150 mm diameter.

When the water velocity under the air pocket becomes supercritical, a weak hydraulic jump is formed on the downstream tail of the air pocket. If this is combined with an average velocity greater than 0.8 m/s in the pipe, entrainment of air bubbles occur and are transported downstream, thus reducing the air bubble. The reducing air bubble might be swept along the line. Ahmed et al. (1984) researched this for angles between 10° and 90° and found that the velocity upstream from the bubble must be more than 1.5 m/s to transport the air. To reduce the air pocket, the velocity should be more than 0.9 m/s. It was also found that the rate of air removal from a longer pocket was velocity dependent and not Froude dependent, although the rising velocity, V_{g0} and clearing/sweeping velocities, V_L are Froude dependent.

Relationships for the sweeping velocity can therefore be scaled using Froude uniformity but the velocity should also be considered when the entrainment and transport of smaller bubbles from the main pocket are evaluated.

Hesketh et al (1991) evaluated the rate of bubble breakage in a control volume. Air bubbles of a specific size were introduced through a nozzle into flowing water in a pipeline.

The maximum stable bubble size measured was in agreement with the prediction by Hesketh et al. (1987).

$$d_s = 1.38(W_{crit})^{0.6} \left(\frac{\sigma^{0.6}}{\rho_L^{0.3} \rho_g^{0.2} \mu_L^{0.1}} \right) \left(\frac{D^{0.5}}{V_L^{1.1}} \right) \dots\dots\dots (2.14)$$

Where $W_{crit} = 1.1$

$$W_{crit} = \text{Critical Weber number} \left(\left(\frac{\tau}{\sigma} \right) / (d(\rho_g / \rho_L)^{\frac{1}{3}}) \right)$$

$$\sigma = \text{Inter-facial surface tension (N/m (721 x 10}^{-3} \text{ N/m))}$$

ρ_l	=	Liquid density (kg/m ³ (continuous phase))
ρ_g	=	Gas density (kg/m ³ (dispersed phase))
μ_l	=	Liquid viscosity (kg/ms (continuous phase))
V_L	=	Superficial velocity of water (m/s)
τ	=	Stress at the bubble surface (N/m)
D	=	Pipe diameter (m)
d_b	=	Maximum air bubble size, (m)

The velocity distribution in the water face underneath the air needs to be defined, because it influences the calculation of the losses.

Sharma et al (1998) conducted experiments to define the velocity distribution at a number of positions relative to the nose and tail of the air bubble, as well as in the interface near the nose and wake of the slug. The liquid velocity in the experimental set-up varied between 1.1 to 2.2 m/s and the Froude number in these tests varied between 1.52 and 3.05. The maximum mean velocity in front of the nose was recorded at a position lower than the centre of the pipe, while in the wake of the slug, the maximum average velocity measured immediately after the bubble, was located above the centreline of the pipeline.

Further behind the wake the maximum mean velocity was again measured below the centre line.

The Movement of an elongated (Taylor) bubble was studied in a vertical pipe (Polousky et al, 1999). The shape, bubble velocity, and the velocity variation due to the hydrostatic pressure gradient were quantified. The oscillation of the bottom of the bubble was studied.

Since previous research focussed on the flow around the nose of the bubble, and because limited information was available pertaining to the flow structure at the wake of the bubble, a detailed experimental investigation of the elongated bubble was undertaken.

The experimental set-up consisted of a 25 mm diameter Perspex pipe with a length of 4.3 (170 x diameter). Four video cameras were used to capture the movement and form of the bubble. The liquid flow velocity, V_L varied from 0 to 0.3 m/s (vertically upward). The bubble length L , varied from $L/D = 1$ to $L/D = 20$.

It was found that the nose of the bubble was in the form of a spherical cup and the bubble was symmetrical. The bottom of the bubble exhibits oscillations, and the bottom varies as the bubble propagates along the pipe. The oscillations are strongly dependent on the bubble length.

For higher flow velocities the liquid film thickness at the nose increased, resulting in a shorter nose.

The velocity of the bubble nose appears to be dependent on the bubble length, due to the effect of gas compressibility. In general it is accepted that the translational velocity, V_t be expressed as:

$$V_t = CV_L + V_{g0} \dots \dots \dots (2.15)$$

Where:

V_{g0} = Drift velocity at stagnant liquid conditions (m/s)

V_L = Liquid velocity (m/s)

It was found that $C \approx 1.2$ for turbulent flow and $C = 2$ for fully developed laminar flow, and hence C is dependent on the Reynolds number, $R_e = \frac{V_L D}{\nu_L}$. The drift velocity, V_{g0} approaches $0.351\sqrt{gd}$, with d = diameter of the air bubble.

The bubble velocity is influenced by the compressibility, and this effect was investigated by observing the relative movement of the bubble nose and the bubble bottom. The following relationship was derived for the bubble **nose velocity**, V_{gn} .

$$V_{gn}(h) = (CV_L + V_{g0}) \left[1 - C \frac{\int_0^h r^2(x) dx}{R^2 [P_a / (\rho_L g) + h]} \right] \dots \dots \dots (2.16)$$

Where:

- R = Radius of the pipe (m)
 P_a = Atmospheric pressure (m)
 H = Hydrostatic head (m)

For the tail (bottom of the bubble) it was shown that the bottom velocity, V_{gT} could be defined by:

$$V_{gT} = V_{gv} - \frac{\partial L}{\partial t} \dots \dots \dots (2.17)$$

$$V_{gT} = V_{gv} \left(1 - \frac{\int_0^L r^2(x) dx}{r^2(L) [Pa/(\rho_l g) + h]} \right) \dots \dots \dots (2.18)$$

The oscillation of the bubble reflects a dominant frequency of 4 Hz, and did not exceed 5 Hz.

Ruder et al (1989) defined the different interfacial flow configurations that have been characterized, starting with stratified/wavy flow, where the gas fills the top section of the conduit and liquid the bottom section. If the gas flow velocity is maintained, while the liquid velocity increased, the flow will change to one of two "intermittent" patterns, plug flow or slug flow. The difference between plug and slug flow is made by referring to the gas movement. In plug flow, the gas velocity is low and the gas appears as elongated gas bubbles, which move along the top of the pipe. In slug flow intermittent appearances are found of high velocity liquid slugs, which can be highly aerated, that bridge the whole pipe.

The authors (Ruder et al, 1989) indicated that an un-aerated slug will exist in a circular pipe if the ratio of the difference between the slug velocity and liquid velocity in front of the slug to the square root of the product of the gravitational constant and the pipe diameter, $(C_F - V_{L1})/(g D)^{0.5}$ must have a value greater than those indicated in **Table 2.1** for different values of $\frac{h_{l1}}{D}$. C_F is the slug velocity and V_{L1} the water velocity (m/s).

Table 2.1: Required velocities to maintain a stable slug with a specific size

h_{L1}/D	$(C_F - V_{L1})/(g D)^{0.5}$
0.1	0.9459
0.2	0.9200
0.3	0.9090
0.4	0.9108
0.5	0.9213
0.6	0.9440
0.7	0.9937
0.8	1.0778

As the tail of the slug is studied and a relationship is derived for a circular tube, it was found that:

$$\left(\frac{4A_{L1}}{\pi D^2} \right) \left(\frac{C_F - V_L}{\sqrt{gD}} \right) = 0.542 \quad \dots\dots\dots (2.19)$$

From this relationship, the liquid thickness in front of air slug for steady-state inviscid flow can be determined. In the case of an unsteady state inviscid flow; the thickness of the liquid in front of the slug is defined by:

$$\left(\frac{C_F - V_L}{\sqrt{gD}} \right) = \frac{\pi D^2}{4A_{L1}} \left(\frac{\pi}{4} \right)^{0.5} \quad \dots\dots\dots (2.20)$$

Figure 2.4 reflects the conditions for a stable slug.

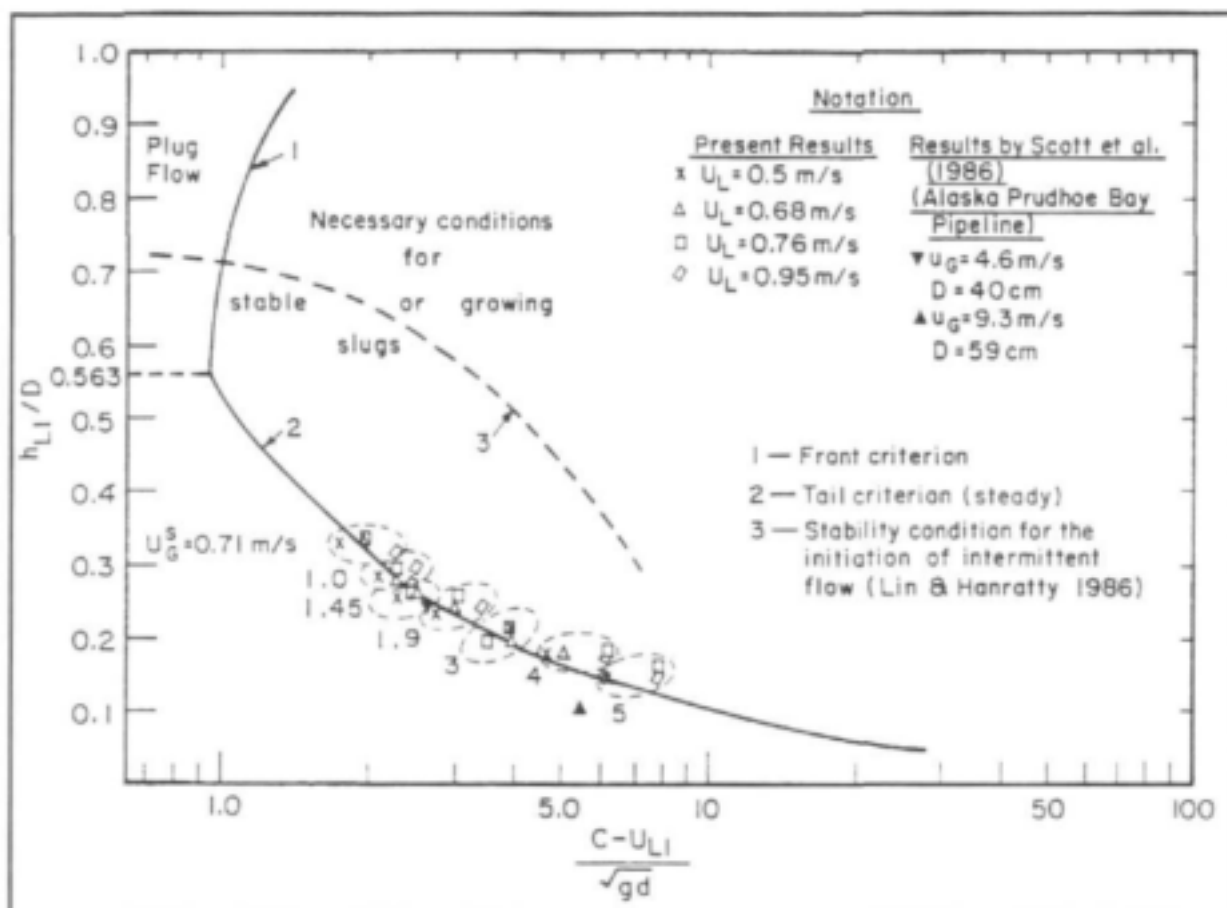


Figure 2.4: Conditions for a stable slug ($V_L = U_L$ = liquid velocity and u_G = gas velocity)

The authors concluded that for the development of a stable slug:

- The front of the slug should be defined as a hydraulic jump, which gives a gas velocity beyond which a stable slug would not exist.
- The propagation velocity of the tail, that defines a minimum height of the liquid layer over which stable slugs can propagate, should exceed the minimum value.

The propagation of long bubbles through stagnant liquids has been studied (Bendiksen (1989)), and it was shown (Davies and Taylor (1939)) from dimensional analysis that when $\rho_G \ll \rho_L$ the bubble velocity is proportional to $(gD)^{0.5}$. When the liquid is moving through the pipe and the Reynolds number ranges from $(8 \text{ to } 50) \times 10^3$ the velocity of the air bubble, V_g , is correlated by:

$$V_g = C_O V_L + V_{g0} \dots \dots \dots (2.21)$$

Where:

- V_{g0} = Bubble velocity in stagnant liquid (m/s)
 V_L = Velocity of the liquid phase (m/s)
 C_0 = 1.20

For inclined pipes these relationships differ significantly.

In the experimental work, the slope of the pipe was changed from horizontal to vertical. Different pipe lengths were used in the experiments and the diameter varied from 2.42 cm to 5.0 cm. The water was fed through the pipeline from the bottom of the system, i.e. upwards along the slope. For each inclination, 50 tests were conducted.

The non-dimensional bubble propagation rate V_{g0}^* in stagnant liquid ($V_L = 0$) yields:

$$V_{g0}^* = \frac{V_{g0}}{\sqrt{gD}} \dots\dots\dots (2.22)$$

For $V_L > 0$ the following features were observed:

- In the lowest liquid velocity interval, the LSQ values of V_{g0} are in good agreement with those from direct measurements in stagnant liquid for all $\theta \geq 0$.
- For high liquid velocities, $V_L > V_L^1(\theta)$, and $0 \leq \theta < 90^\circ$, V_{g0} decreases approximately as V_{g0} approaches $V_{g0}^* \sin \theta$, where V_{g0}^* is the value of V_{g0} for $\theta = 90^\circ$.
- For $-30^\circ \leq \theta < 0$ (liquid downflow) $v_a \approx -|v_a(+\theta)|$ for average liquid velocities less than a critical value $v_L^c(\theta)$, and $v_a > 0$ for $v_L \geq v_L^c(\theta)$.
- For all the measurements of upward flow ($\theta > 0$), C_0 has a value of 1.19 to 1.2.

When $\theta < 0$ the situation becomes more complex and a number of bubble movement phenomena were identified (Bendiksen, 1984).

The authors (Wisner et al, 1975) defined the two ways in which air can be removed from a pipeline: mechanically (air relief valves, traps, etc) and hydraulically (motion of the moving water).

During hydraulic transport of the air bubble it can either be 'swept' (removed as a whole) or entrained into the solution. The water velocity required to transport the air is referred to as the clearing velocity.

Research by Kalinske and Bliss (1943) to determine the required discharge to transport the air was related to the slope of the pipeline.

$$\frac{Q_c}{gD^3} = 0.707 \tan \phi \dots\dots\dots (2.23)$$

- ϕ = Slope angle of the pipe (flow is down the slope) (degrees)
- Q_c = Critical discharge (m³/s)
- D = Diameter of the pipe, m

Kent (1952) recommended the following semi-empirical formula:

$$V_{min} = 1.62 \sqrt{\frac{L_B}{D}} \sqrt{gD \sin \phi} \dots\dots\dots (2.24)$$

Where:

- $\frac{L_B}{D}$ = Slope factor ($\frac{L_B}{D}$)
- L_B = Length of the bubble (m)

By applying dimensional analysis, it was shown that:

$$\frac{V_s}{\sqrt{gD}} = f\left(\frac{L_B}{D}, R_{el}, \phi\right) \dots\dots\dots (2.25)$$

- V_s = Sweeping velocity (m/s)
- R_{el} = Reynolds number
- ϕ = Slope of the pipe (degrees)
- L_B = Length of the bubble (m)

A dimensionless pocket size parameter, was defined:

$$u = \frac{Vol}{\left(\frac{\pi}{4} D^3\right)} \dots\dots\dots (2.26)$$

Where:

- Vol = Volume of the air pocket

For different air pocket sizes the critical velocity to transport the air was determined. The maximum bubble size was approximately 50% of the diameter of the pipe. The results were presented in a graphical format, with $\frac{V_c}{\sqrt{gD}}$ plotted on the vertical scale and $\sqrt{\sin \phi}$ plotted on the horizontal axis.

All the values were enclosed in an envelope, which reflected the follow relationship.

$$\frac{V_c}{\sqrt{gD}} = 0.25 \sqrt{\sin \phi} + 0.825 \dots\dots\dots(2.27)$$

The author found that for the same Froude number, air in larger pipes clear faster than in smaller pipes. This can probably be explained by the fact that the size of generated bubbles does not vary significantly in the model and prototype.

The authors (Ervine and Himmo, 1961) investigated the dynamics of air bubbles at the entrance of a tunnel connected to a drop shaft.

2.7 Induced pressures resulting from air release

It is known that air valves tend to blow-shut under a relatively low differential pressure. Air Valve Manufacturers then introduce features like the dynamic shield to prevent valve closure. This however led to high induced pressures that originate from the deceleration of the approaching water columns. **Figure 2.5** depicts a typical relationship between the differential pressure and the discharge for a two-stage large orifice air valve. Air valve manufacturers have practically investigated designing the system to introduce the "switch" at a predetermined differential pressure and switching to a "specific smaller orifice" that creates acceptable pressure surges.

Recent developments (3 stage air valves), utilises the blow-shut phenomenon to step down the area of the large orifice, and thereby limits the over-pressures to acceptable limits. This staged closure of the large orifice can be done in such a way that the rate of discharge can be limited to a predetermined capacity by adjusting the size of the orifice for the different stages.

Taitel et al, (1989) indicated that, by the addition of the liquid momentum to the quasi-steady state, the analysis procedures for two phase transient analysis flow simplifies.

In the analysis, distinction was made between, stratified, annular, slug and bubbly flow.

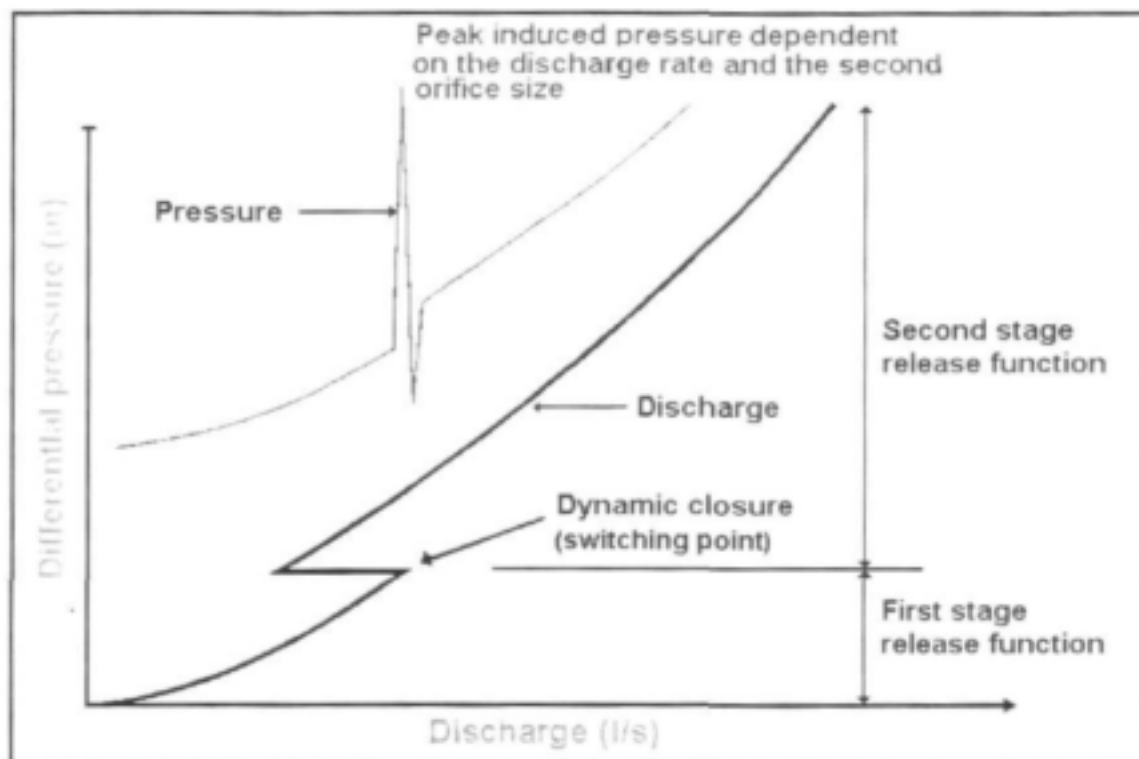


Figure 2.5: Differential pressure - discharge relationship for a two-stage large orifice air valve.

2.8 Options to effectively de-aerate pipelines

Since it is difficult to prevent the intrusion of free air into pipe systems, procedures have been developed to de-aerate pipelines. Hydraulic and mechanical methods of de-aeration are distinguished and discussed below

Hydraulic de-aeration of pipelines

Research has determined the "threshold" velocity for a given pipe at a certain slope where the air bubble becomes unstable and is transported through the pipeline Kalinske and Bliss (1943) Equation 2.28 and Wisner (1975) Equation 2.29. The most commonly used equations that indicate the velocity required to transport the air are:

$$\frac{Q_L^2}{gD^5} = 0.707 \tan \phi \dots\dots\dots (2.28)$$

Where:

Q_L = Required flow rate (m^3/s)

D = Pipe diameter (m)

ϕ = Slope of pipe (degrees)

and

$$V_L = \sqrt{gD}(0.25\sqrt{\sin\phi} + 0.825) \dots\dots\dots (2.29)$$

Where:

V_L = Critical velocity required for the removal of air (m/s)

Figure 2.6 provides a graphic comparison of the results from the two equations for a 0.5 m diameter pipe. The two equations provide contradictory results. The reason for the differences are mainly due to the way in which Equation 2.29 was derived. The relationship of Equation 2.29 is based on observation of large air bubbles ($d/D > 0.2$, where d = the bubble diameter and D the internal pipe diameter), and therefore, in accordance with Equation 2.29, high flow velocities are required to transport air downstream through the pipeline. **Figure 2.6** also contains a proposed relationship, Equation 2.30, which can be used for assessing the hydraulic ability to transport free air in water pipelines (Van Vuuren, 1994).

$$V_L = \sqrt{gD}(0.25\sqrt{\sin\phi} + 0.4) \dots\dots\dots (2.30)$$

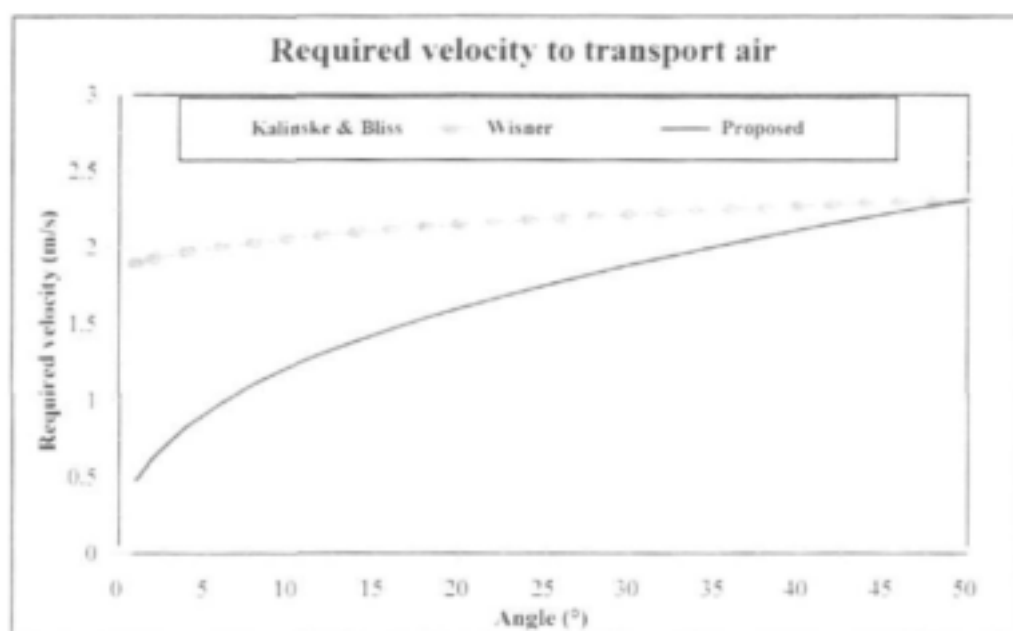


Figure 2.6: Required flow velocity to remove air hydraulically

Mechanical removal of air

In normal water distribution systems, sufficient outlets are available to expel the air, or to dissolve it due to the higher static pressures experienced during low demands.

In other water infrastructure components, it remains important to consider the effect of free air in the system.

Contradictory statements were made in the past regarding the need for air valves (Van Vuuren, 1991) and certain functional tests of these valves have not yet been included in the specifications of water supply authorities.

Details on the sizing and positioning of air valves are readily available, and therefore only certain aspects are highlighted here.

- **Large orifice air valve - sizing**

These valves are mostly sized for intake. The differential pressure for intake should preferably be limited to 40 kPa. With this as a guideline, **Appendix 2.1** can be used to determine the required flow rate, and following from this, the size of the large orifice air valve can be determined.

Large orifice air valves are also used to discharge large volumes of air when pipelines are filled. This specific function, however, has been misrepresented, mainly because valve manufacturers have focused on the discharge capacity, employing kinetic shields and other facilities to maintain the high discharge rate. In most of these developments the consequences that follow after blow-shut has occurred, or when the valve closes after all the air has been discharged, were ignored. Details are available describing the maximum permissible filling rate to prevent induced surge pressures resulting from the closure of the large orifice (Van Vuuren, 1993).

The implementation of the staged closure of the large orifice reduces the requirement to restrict the discharge velocity of the air through the valve to 30 m/s, as was suggested previously (Van Vuuren, 1991).

- **Small orifice air valve – sizing**

A small orifice provides the facility to de-aerate the system under normal operating pressures and the selection therefore should be a function of the operating pressure. Drop tests provide the designer with proof that the small orifice will function under the operating pressures. The sizing of the small orifice is dictated by the operating pressure, atmospheric pressure and the characteristics (mass and buoyancy) of the float. In recent developments, a slit opening was employed as the small orifice and the float virtually peels the seal from the opening when the float drops.

Some manufacturers use the name *combination air valve* for a valve where the small and large orifice function is combined in a single valve.

2.9 Positioning of air valves

The sizing of air valves was discussed briefly and some attention is now paid to the positioning of air-release and vacuum-break valves.

Although it might be necessary to install both large and small orifice air valves at specific points along a pipeline (double orifice or combination air valve) the functioning of these valves is so distinct that their positioning is discussed separately.

The positioning of large orifice air valves as vacuum breakers

Sub-atmospheric pressures in a pipeline result from pump stoppage or valve closure, pipe failure or drainage of sections of the line.

The points along the longitudinal section of the pipeline closest to the hydraulic grade line (this does not necessarily constitute topographic high points) will reach sub-atmospheric pressures first (if the pressure drops that low), leading to the activation of the large orifice air valve.

- A large orifice air valve, when used as an alternative for a one-way surge tank (or discharge tank), should be placed as near as possible to, and just downstream from, the pump check valve.
- Depending on the longitudinal profile and the positioning of the shut-off valves or non-return valves in the line, a large orifice air valve should be provided to prevent sub-atmospheric pressures in the line when it is drained. To ensure the flexible and effective operation of the pipeline (to isolate sections), shut-off valves are normally provided at intervals of between 1 500 and 3 000 m (or more). The result is that the spacing of large orifice air valves is more or less the same, due to the requirement that one should be able to drain any section between shut-off valves.
- Where multiple peaks occur along a section of pipeline between the shut-off valves, a number of scour valves must be supplied to drain the pipeline. In this instance, the operation of each scour valve and any possible associated negative pressures should be taken into consideration in order to determine the positioning of the air valves.
- The results of surge analyses indicate where sub-atmospheric pressures might occur, and it is suggested that large orifice air valves should be provided wherever the pressure drops to below 6.0 m absolute.

Visualising the operation of the system remains an integral part of assessing the problem to obtain the optimal solution, rather than to follow normative design criteria.

The positioning of large orifice air valves to release high volumes of air

Large orifice air valves release air only under low differential pressures, and this operational feature is used when the line is filled.

If the requirements are met to prevent low pressures in the pipeline, as discussed in the previous paragraph, most of the requirements concerning the positioning of air release valves will also have been met. Other points where large orifice air valves are required depend upon the way in which the line is filled, how many isolated sections are available, and how the filling rate can be controlled.

The positioning of small orifice air valves

During operation, air can be released only through the small orifice air valve. For this to occur, the air should be "stationary" at the valve, although the pipeline simultaneously continues to handle the design flow. This emphasises the need to provide a collection chamber underneath the valve, where the air can be captured temporarily and then released through the air valve. **Although water can contain 2-3% per volume free air, it is not necessary to install small orifice air valves to provide such a discharge rate!** It is suggested that the following approach be used to determine the position of small orifice air-release valves:

1. Determine whether the available free air in the pipeline is hydraulically transported through the pipeline along the downward sloping-sections. Equations 2.28, 2.29 or 2.30 can be used to determine whether the air can be hydraulically transported. Work through all the sections and identify those sections where air is not transported hydraulically.
2. The points identified in 1 above, where free air is not transported, potentially constitute positions where small orifice air valves are required.
3. Define the points where small orifice air valves should be installed by evaluating all the potential points under 2, working downstream along the line. The valves can be of use to de-aerate the system only if it is possible for the air to be captured

at the small orifice air valve. The effectiveness of all air valves upstream of the point under consideration therefore influences the motivation for the downstream small orifice air valve. If the distance between small orifice air valves is short (say 250 m), it is not necessary to provide a small orifice air valve at all the identified points, provided that the line is effectively filled and that the upstream small orifice air valve effectively releases all the air that reaches it.

Ensuring the effective operation of the small orifice air valve necessitates the provision of a collector chamber to capture all the air. As a guideline, it was suggested that the small orifice air valve should be installed on a T-piece, the diameter of which should be at least 50% that of the pipe. In the case of a double air valve, it was recommended that the T-piece on which the air valve is installed, should be equal to the diameter of the line and that the length of the riser pipe should be at least 50% the diameter of the pipe, but not less than 150 mm. **The experimental results has been included and a guideline has been compiled, see Section 4.**

2.10 Determination of the intake capacity of air valves

The intake capacity of air valves has always been a point of contention, but unfortunately it is also the most important operational characteristic when determining the air valve size. The insufficient capacity of vacuum pumps, resulted in currently used standards allowing manufacturers of air valves to determine the intake capacity of air valves by pressurizing the outlet and forcing pressurized high density air flow in reverse through the air valve. One such standard is the European Standard prEN 1074-4 Part 4 Final Draft (February 2000). The test is not required for valves larger than DN100.

The incorrectness of this procedure (referred to as Procedure 1) can be calculated, and is shown, in **Figure 2.7**, for different differential pressures across the valve. The two governing mass-flow equations for air through an orifice distinguish between the sub-sonic and sonic flow conditions.

The mass flow rate for sub-sonic flow conditions (unchoked conditions where $p/p_a > 0.528$ for intake capacity and $p_a/p \leq 0.528$ for release capacity) can be determined by the following relationships. Refer to this calculation procedure to determine the intake capacity by applying the mass continuity equation and the orifice equation is Procedure 2.

Air intrusion through the air valve ($p_a > p$)

Un-choked conditions ($p/p_a > 0.528$)

$$\dot{m} = Cd_i A_i \left\{ 2 p_a \rho_a \left(\frac{k}{k-1} \right) \left[\left(\frac{p}{p_a} \right)^{2/k} - \left(\frac{p}{p_a} \right)^{(k+1)/k} \right] \right\}^{0.5} \dots\dots\dots (2.31)$$

Choked conditions ($p/p_a \leq 0.528$)

$$\dot{m} = Cd_i A_i \left\{ k p_a \rho_a \left[\frac{2}{k+1} \right]^{(k+1)/(k-1)} \right\}^{0.5} \dots\dots\dots (2.32)$$

Where:

- A_i = Flow area for air inflow (m^2)
- A_o = Flow area for air outflow (m^2)
- Cd_i = Discharge coefficient for the inlet area of the air valve
- Cd_o = Discharge coefficient for the inlet area of the air valve
- k = Isentropic constant, $k=1.4$
- $\dot{m}$ = Mass flow rate (kg/s)
- p = Absolute pressure head (m)
- p_a = Absolute pressure head in the air valve (m)
- ρ_a = Density of the air in the air valve under the line head conditions (kg/m^3)

Air release through the air valve ($p > p_a$)

Un-choked conditions ($p_a/p > 0.528$)

$$\dot{m} = Cd_o A_o \left\{ 2 p \left(\frac{k}{k-1} \right) \left[\left(\frac{p_a}{p} \right)^{2/k} - \left(\frac{p_a}{p} \right)^{(k+1)/k} \right] \right\}^{0.5} \dots\dots\dots (2.33)$$

Choked conditions ($p_a/p \leq 0,528$)

$$\dot{m} = C d_0 A_0 \left\{ k p \rho \left[\frac{2}{k+1} \right]^{(k-1)/(k+1)} \right\}^{0.5} \quad \text{and} \quad \frac{p_a}{p} = \left(\frac{\rho_a}{\rho} \right)^{1/k} \dots\dots\dots (2.34)$$

Figure 2.7 reflects the discrepancy between the two procedures for the determination of the intake capacity of an air valve. It is suggested that the intake capacity characteristics of air valves obtained through any test that pressurise the outlet should be used with great care (and suspicion). The end results is that when air valve manufacturers are using the pressurized outlet procedure (procedure 1), the performance of the valve will be overestimated and the intake capacity under atmospheric conditions will be far less. This will lead to unwanted vacuum condition in the pipeline.

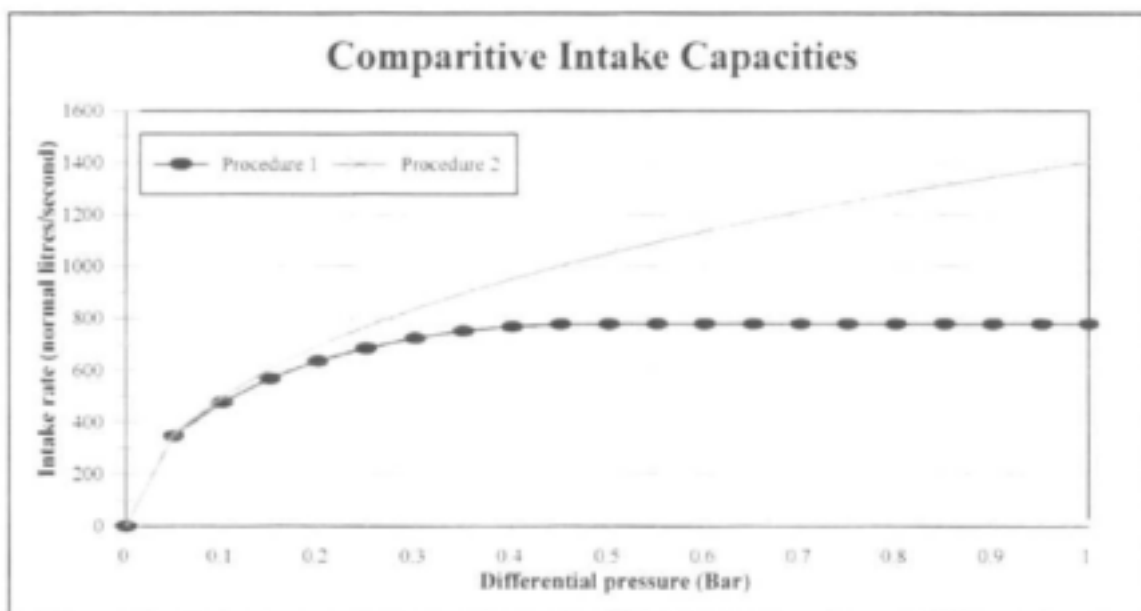


Figure 2.7: Comparative Intake Capacities

2.11 Utility software for the determining the sizing and location of air valves

Air valve sizing and location software was developed in South Africa, and is currently used by pipeline designers in Africa, Europe, America, Australia and Asia to optimally locate the air valves. These software packages are freely available from the air valve suppliers or from the Web site of WRC. The procedures used in the programs are discussed in Section 2.12. The interrelationships of the operating condition as well as the location of scour-, non-return and isolation valves on the location of air-valves are reflected in the program.

2.12 Procedures for the sizing and location of air valves

The following procedures were used for the sizing and location of air valves along a pipeline. The objective of the procedure is to determine the size and the position of the air valves on a new pipeline to:

- prevent vacuum conditions in the pipeline.
- have sufficient discharge capacity and
- release air under operating pressures.

The procedure firstly focuses on the intake capacity of the valves since atmospheric pressure is the only available energy to produce flow to balance the pressure difference. The following steps are followed:

- Obtain the flow characteristics, pipe characteristics, longitudinal profile of the pipeline and other relevant information.
- Define the position of all the other valves, such as isolating, drainage and non-return valves that will be used to control the filling and draining of the pipeline or divide the pipeline into isolatable sections. Also define off-takes along the pipeline where water will be abstracted.
- For *Burst analysis* using a modified Darcy Weisbach formula, the intake capacity for each pipeline section is calculated. The rupture size (percentage of pipe diameter) reduces the size of the air valve in terms of intake requirements.

$$Q_{burst} = \sqrt{\frac{g\pi^2 D^5 S_0}{8\lambda}} \frac{A_{rupture}}{A} \dots\dots\dots (2.35)$$

Where:

Q_{burst}	=	Flow rate due to burst (m^3/s)
g	=	gravitational acceleration (m/s^2)
D	=	Inside diameter of pipe (m)
S_0	=	Slope of pipe (m/m)
λ	=	Friction factor
$A_{rupture}$	=	Area of the rupture (m^2)
A	=	Area of the pipe (m^2)

- In the *Draining analysis* the potential maximum flow rate at which the section can be drained/scoured is determined by the following relationship:

$$Q_{drain} = C_D \sqrt{2g\Delta h} \left(\frac{\pi D_{outlet}^2}{4} \right) \dots\dots\dots (2.36)$$

Where:

Q_{drain}	=	Flow rate due to controlled drainage (m^3/s)
C_D	=	Discharge coefficient for an orifice (0.6)
g	=	Gravitational acceleration (m/s^2)
Δh	=	Available energy between peak and drainage point (m)
D_{outlet}	=	Inside diameter of outlet pipe (m)

- Based on the results the required intake capacities can be determined at all the apex points and other points where slopes change (since a large enough change in energy warrants the placing of an air valve).
- The calculated required intake capacity is then compared with the intake capacities of the available air valves and suitable valves are selected.
- The position of the air valves to release air under operating pressures is then determined. Determining the ability of the fluid to transport the air hydraulically using the given relationships (Equation 2.28 – 2.30).
- The distance preferred by the designer for the spacing of air valves (distance between air valves releasing air under operating conditions).
- Review of special points, for instance at the pump station or non-return valves.
- The last step is to combine the results and determining the correct air valve (type and size) to be installed.

2.13 Detection of slug flow in pipelines

Gas and liquid (water) flow show a number of interfacial configurations called *flow patterns*. Slug flow refers to the flow pattern, where sufficient free air is available, resulting in slugish flow of the liquid. A liquid slug occurs when the liquid fills the whole pipe and is propagated intermittently down the pipe. Under high gas flow conditions the slugs become highly aerated, and the slugs become shorter, making it difficult to identify slug flow.

The techniques employed to detect slug flow were summarized by Hewitt (1978) and Jones (1979):

The x-ray absorption measurements, used by Jones & Zuber (1974), have the disadvantages of high cost and the need for safeguards. Barnea et al. (1980), Reimann et al. (1981) and others have used conductance probes, which detect the presence of a particular phase by making use of the large difference in electrical conductivity between the gas phase and the liquid phase. Jones & Zuber (1978) made use of a hot-film anemometer, but its fragility and short lifetime limited its acceptance. Hubbard & Dukler (1966) who first recognized the value of instrumental techniques, used the power spectra of the fluctuations in the local pressure to characterize different flow patterns. However, it is not at all certain that the frequency domain is an appropriate basis to distinguish between slugs and large amplitude waves that close off the pipe momentarily. Weisman et al. (1979) used the real-time differential pressure signal. They suggested a scheme that uses the amplitude and frequency of the signal to characterize a particular pattern. Tutu (1982) proposed using the probability density of the fluctuations in the pressure difference for vertical systems.

Experimental work confirmed that the slug velocity was approximately equal to the gas velocity for full slug flow. In the case of pseudo slug flow, the velocity of the pseudo slug was less than 2.0 m per second for the superficial gas velocity that varied from 4 to 36 m/s. The presence of slug flow can be established by using pressure transducer to determine the time difference of the pressure fluctuations from which the velocity of the slug can be determined. Based on the knowledge of the slug velocity, the presence of slugs can be qualified.

Nelsen et al. (Iowa Institute of Hydraulic Research (unpublished report) demonstrated that for flows with a low concentration of air bubbles it is possible to separate the ADV velocities measured on water from those of the air bubbles, based on the ADV signal strength. The extent of the signal strength variation is proportional to the amount of air bubbles.

2.14 Bubble characteristics in two phase flow

Netto et al (1999) investigated the movement of large air bubbles in pipelines under conditions of slug and plug flow. It was found that the bubble shape under these two flow conditions is different, and that the transition between plug and slug flow is dependent on the Froude number, Fr , and the bubble length, L_B . The distinction between slug and plug flow is demonstrated in **Figure 2.8**.

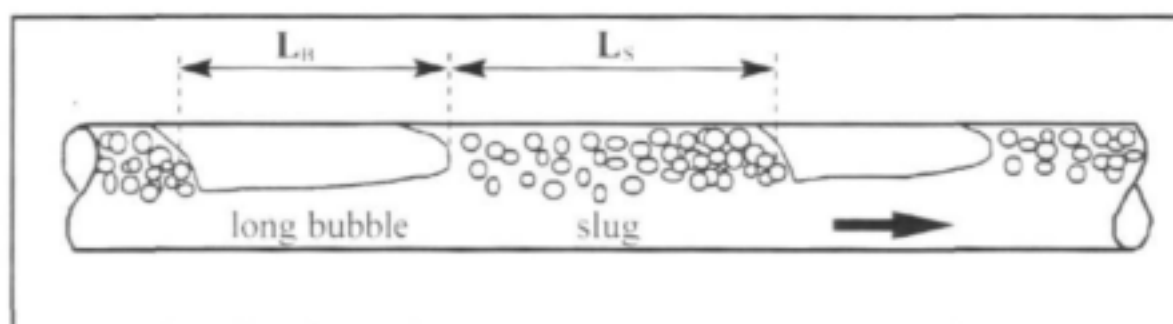


Figure 2.8: Differentiation between slug and plug flow

The authors previously used the statistical cell unit to describe the flow in order to be able to predict important flow parameters such as pressure gradient, mean phase velocities and mean volume fractions. This model lacked the ability to predict the statistics of bubble and slug lengths and their evolution along the pipe.

Two models are used to describe the gas – liquid slug flow. The widely employed “slug – tracking model” predicts the volume and position of each individual bubble and in the calculations the slug is followed “step-by-step” along the pipe. Nydal et al (1995) used this model to describe the movement of slug flow evolution in a pipe.

The second model is based on the transport equations for the distribution of bubble and slug flow. The assumption that all bubbles are fully developed leads to large errors when the bubble length is smaller than $100D$. Furthermore, the assumption that the liquid interface underneath the bubble is motionless in a horizontal flow leads to errors of at least 5% when the liquid fraction is calculated. In describing bubble movement, distinction is made between the nose, body, hydraulic jump and the tail. **Figure 2.9** reflects the different bubble regions. The shape of the bubble nose is dependent on the flow ahead of the bubble and independent of the bubble length.

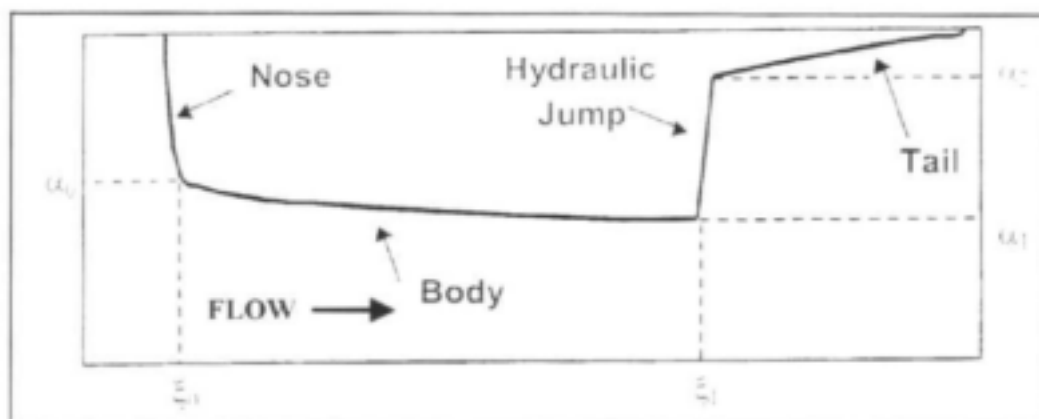


Figure 2.9: Definition of regions in a bubble

The form of the nose (**Figure 2.10**) is directly proportional to the Froude number ($V/(gD)^{0.5}$) in the pipeline. It was also established that for Froude numbers in excess of 2, the tail at the back of the hydraulic jump disappear and that small bubbles are entrained in the wake of the long bubble (**Figure 2.11**).

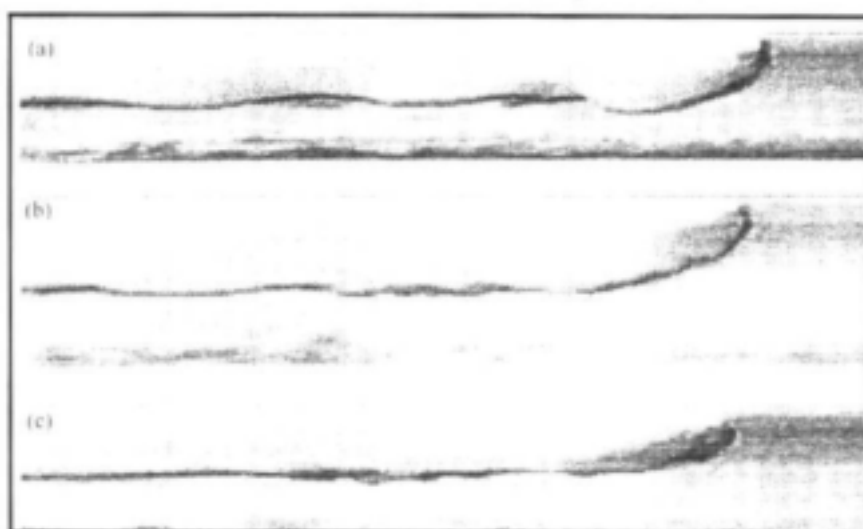


Figure 2.10: Nose of the air bubble – (a): $v = 0,6$; (b): $v = 1,0$; (c): $v = 1,5$ m/s (Flow direction is from right to left)

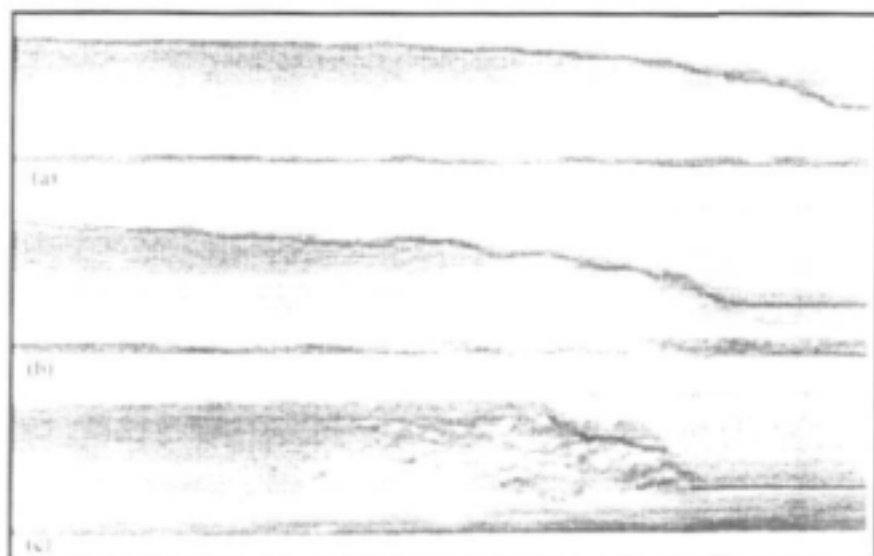


Figure 2.11: Tail of the air bubble – (a): $v = 0,6$; (b): $v = 1,0$; (c): $v = 1,5$ m/s (*Flow direction is from right to left*)

Figure 2.12 reflects the form of the air bubble for velocities of 0.5 m/s (a) and 1.8 m/s (b).

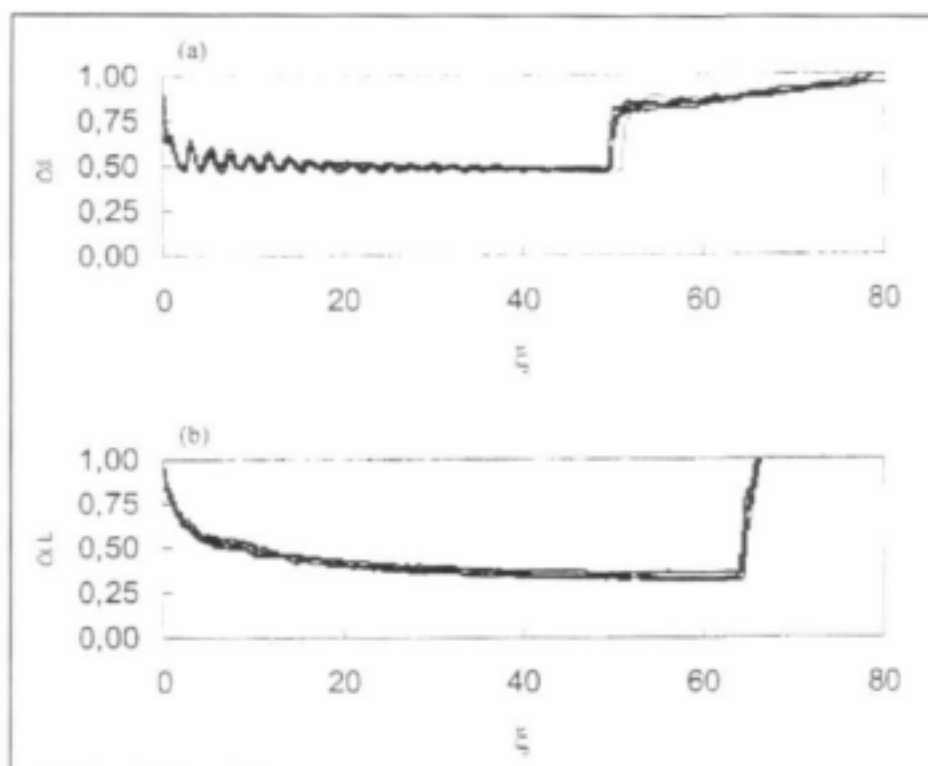


Figure 2.12: Form of the air bubble for velocities of 0,5 m/s (a) and 1,8 m/s (b)

α_L = liquid volume fraction and ε = the distance from the bubble nose as a factor of the pipe diameter. The authors found a high correlation between the model predictions and the experimental data for flow velocities varying from 0.5 to 1.8 m/s. Furthermore, it was established that for a Froude ≤ 2.0 , the bubble velocity could be determined by:

$$V_{bubble} = V_L + 0.48(gD)^{0.5} \dots\dots\dots (2.37)$$

For higher Froude numbers ($Fr > 2.0$) the velocity of the bubble is determined by:

$$V_{bubble} = 1.2V_L \dots\dots\dots (2.38)$$

Where:

- V_{Bubble} = Calculated bubble velocity (m/s)
- V_L = Average liquid velocity (m/s)
- D = Diameter of the pipe (m)

It has further been determined that the bubble length is a function of the pipe diameter, velocity of the liquid, bubble velocity and the bubble volume.

2.15 Pressure drop across an air bubble

An experimental system of a horizontal pipe was constructed by Fan et al (1993). The pressure was measured by the use of a piezoresistive pressure transducer mounted to the top of the pipe, while the velocity, length and void fraction was determined by two pairs of conductance wires.

In a horizontal pipe the pressure drop over the slug consists of the following three parts:

- The pressure drop associated with the hydraulic jump in front of the slug, ΔP_h
- A friction loss in the body of the slug (water), ΔP_f and
- A pressure variation associated with the velocity change at the rear of the slug, ΔP_r

The total pressure change, ΔP_T can then be determined by:

$$\Delta P_T = \Delta P_h + \Delta P_f + \Delta P_r \dots\dots\dots (2.39)$$

The researchers made a distinction between a stable and unstable slug and derived from the continuity of mass and momentum principles, the following relationships for the rear of the slug.

For a stable slug (volume is constant):

$$\Delta P_r = \frac{1}{2} \rho_L (C_F - V_{L(end\ slug)})^2 \dots\dots\dots (2.40)$$

Where:

C_F = Velocity of the front of the slug moving downstream (m/s)

$V_{L(end\ slug)}$ = Velocity of the liquid at the end (upstream) of the slug (m/s)

$$\Delta P_r = \tau_w \pi \frac{DL_s}{A_3} \dots\dots\dots (2.41)$$

For an unstable slug (rear of slug):

Distinction is made between a growing ($C_F > C_T$) and a decaying slug ($C_F < C_T$). In the case of a growing slug $C_F > C_T$, which indicates that the tail (T) will move slower than the front (F). The pressure drop at the rear of the slug in the case of a growing slug is:

$$\Delta P_r = \frac{1}{2} \rho_L [(C_F - V_{L(front\ slug)})^2 - (C_F - C_T)^2] \dots\dots\dots (2.42)$$

Because $(C_F - V_{L(front\ slug)})$ is small and $(C_F - C_T)$ is larger than $(C_F - V_{L(front\ slug)})$ the pressure drop at the rear of the slug is negative.

In the case of a decaying slug $C_F < C_T$ and with reference to **Figure 2.13**, the following relationship was derived:

$$\Delta P_r = \frac{1}{2} \rho_L \left[\frac{1}{(1 - \Sigma_3)} \left(\frac{A_{31}}{A_3} \right) (C_T - V_{L1}) - (C_F - C_T) \right]^2 \dots\dots\dots (2.43)$$

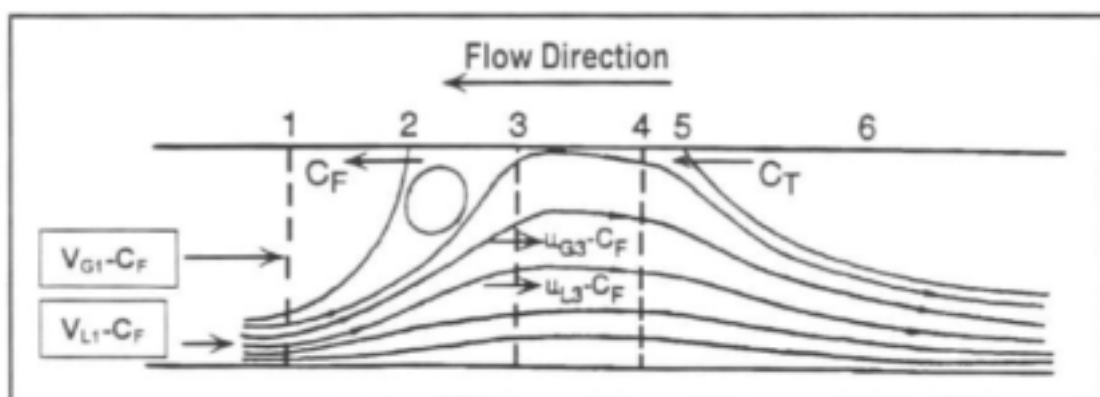


Figure 2.13: Reference to the flow regions

Where: C_T = Velocity of the tail of the slug (m/s) and

V_{L1} = Liquid velocity at section 1 (m/s).

The pressure drop associated with the frictional drag has been defined as:

$$\Delta P_f = \tau_w \pi \frac{DL_s}{A_3} \dots\dots\dots (2.44)$$

Where:

τ_w = Average resisting stress at the wall (Pa)

L_s = Bubble length (m)

The wall stress is approximated by the Blasius equation:

$$\tau_w = 0.046 \left(\frac{D}{\nu_L} \right)^{-0.2} \frac{\rho_L}{2} V_{L3}^{1.8} \dots\dots\dots (2.45)$$

With:

ν_L = Kinematic liquid viscosity

Through experimental work, the following relationships were defined for the value of V_{L3} .

For stable slug flow:

$$V_{L3} = C_F - \frac{1}{(1-\epsilon_3)} \frac{A_{L1}}{A_3} (C_F - V_{L1}) \dots\dots\dots (2.46)$$

For growing slugs:

$$V_{L3} = 2C_F - \frac{1}{(1-\epsilon_3)} \frac{A_{L1}}{A_3} (C_F - V_{L1}) - C_T \dots\dots\dots (2.47)$$

And for decaying slugs:

$$V_{L3} = C_F - \frac{1}{(1-\epsilon_3)} \frac{A_{L1}}{A_3} (C_F - V_{L1}) \dots\dots\dots (2.48)$$

The experimental data supports these relationships, although it was indicated by the author's that some of the simplifications could lead to overestimation of the pressure drop ΔP_h .

Another method that was followed was to use the classical relationship of Darcy-Weisbach and to modify it for two phase flow (P Boeriu, IAHR). The value of Lambda,

λ_{mix} for the two-phase flow is dependent on the Reynolds and Froude number, the absolute roughness (k) and the proportional air flow rate factor C_m .

$$C_m = \frac{Q_{air}}{Q_{air} + Q_{water}} \quad (2.49)$$

The relationship was tested on pipes with diameters between 100 and 400 mm. Based on the experimental data, the following relationship was derived:

$$\frac{\lambda_{mix}}{\lambda_{water}} - 1 = 0.00083 C_m^{1.7} \quad (2.50)$$

Figure 2.14 below indicates the change in the value of Lambda for the mixture, λ_{mix} with reference to the Lambda calculated for a single phase.

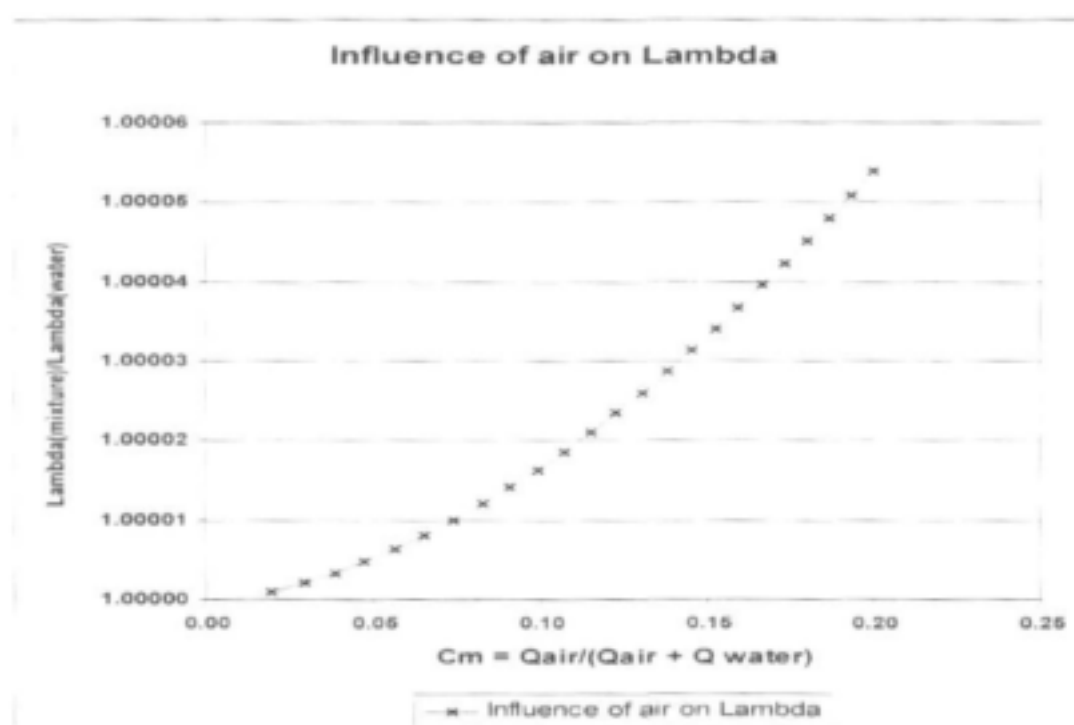


Figure 2.14: Influence of air on the value of Lambda

A comprehensive treatment of all sources of the pressure drop within intermittent gas-liquid flows was undertaken (Cook and Behnia, 2000). The pressure drop associated with the viscous dissipation within the slug was calculated, and the presence of dispersed bubble in a slug is accounted for, without using the assumption of

homogenous flow. Experimental work was conducted on a 32 and a 50 mm pipe with the slope varied between 0° and 10° . It was found that the observed pressure drops were more than the loss predicted by the present models for intermittent flows. This finding supports the work by Andreussi, Minervini and Pagliati (1993) who indicated the under prediction of the friction loss of up to a 30%. The physical model of the intermittent flow is presented in **Figure 2.15**.

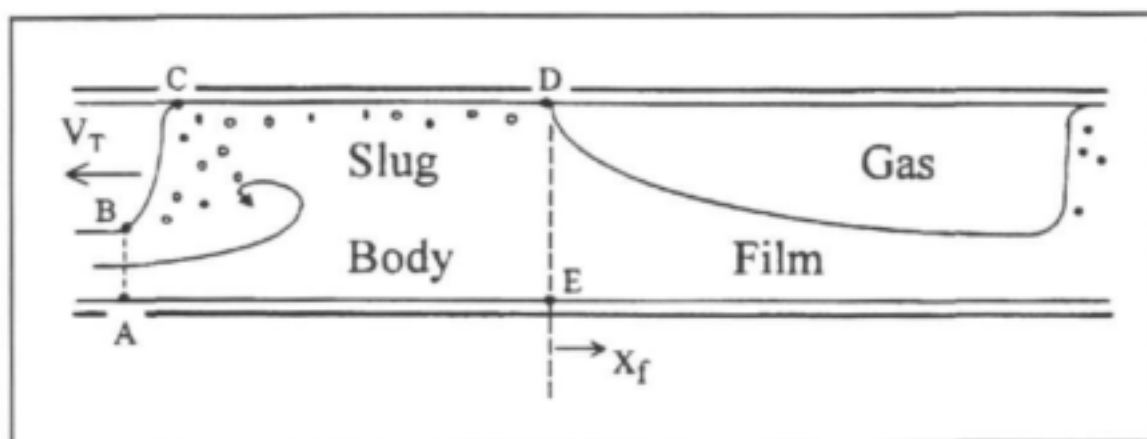


Figure 2.15: Bubble distribution in a horizontal pipe

Three separate sections are defined:

- A highly turbulent mixing zone, which is EXTREMELY aerated at moderate and high flow rates.
- The slug body which has a significant greater liquid hold-up and
- A trailing (tail) section with the bubble on the liquid film.

The pressure drop for intermittent flow is the sum of:

- Hydrostatic head loss in an inclined pipe
- The frictional loss caused by wall shear
- Viscous losses associated with eddy motion in the mixing vortex and
- The acceleration pressure loss due to gas expansion

These losses are presented by the following relationship:

$$\frac{dP}{dx} = \left(\frac{\partial P}{\partial x} \right)_H + \left(\frac{\partial P}{\partial x} \right)_F + \left(\frac{\partial P}{\partial x} \right)_V + \left(\frac{\partial P}{\partial x} \right)_A \dots \dots \dots (2.51)$$

With subscripts H, F, v and A respectively referring to the contribution of hydrostatic, friction, viscosity and the area to the total loss. Determining the friction loss with Colebrook - White relationship it is proposed that an effective Reynolds number should be used.

$$Re_{eff} = \frac{\rho_m V_m D}{\mu_{eff}} \dots \dots \dots (2.52)$$

This results in the following relationship:

$$h_f = \frac{\lambda L V^2}{2gD} \dots \dots \dots (2.53)$$

With:

$$\frac{1}{\sqrt{\lambda}} = -2 \log \left(\frac{k}{3.7D} + \frac{2.51}{Re_{eff} \sqrt{\lambda}} \right) \dots \dots \dots (2.54)$$

The viscous losses were determined by recognizing the similarity between the mixing vortex and an abrupt expansion in normal flow. The authors proposed the following relationship:

$$\Delta P = \frac{k \rho_L}{2} (V_m - V_n)^2 \dots \dots \dots (2.55)$$

Where:

- k = 0.3
- V_m = Velocity of the mixture (m/s)
- V_{te} = Velocity at the end of the liquid film (m/s)
- ρ_L = Liquid density (kg/m³)

The pressure drop associated by the acceleration was written as:

$$-\left(\frac{\partial P}{\partial x}\right)_A + z \frac{\partial P}{\partial x}$$

With

$$z = \frac{1.2 V_{LS} V_{GS}}{P V_T^2} \left[\frac{\rho_g V_{GS}^2}{(1-H_L)^2} - \frac{\rho_L V_{LS}^2}{H_L^2} \right] - \frac{V_{GS}^2}{RT(1-H_L)} \dots \dots \dots (2.56)$$

Where:

- V_{LS} = Superficial velocity of the liquid (m/s)
- V_{GS} = Superficial velocity of the gas (m/s)

- V_T = Translational velocity (m/s)
 H_L = Liquid hold-up
 R = Gas constant ($287 \text{ J kg}^{-1} \text{ K}^{-1}$)
 T = Temperature ($^{\circ}\text{K}$)

These components can be combined into the following relationship:

$$\frac{\partial p}{\partial x} = \frac{\left(\frac{\partial p}{\partial x}\right)_F + \left(\frac{\partial p}{\partial x}\right)_H + \left(\frac{\partial p}{\partial x}\right)_V}{(1 + Z)} \dots\dots\dots (2.57)$$

The hold up is determined by:

$$\frac{\partial H_L}{\partial x} = 1.2 \frac{V_{LS} V_{GS}}{P V_T^2} \frac{\partial p}{\partial x} \dots\dots\dots (2.58)$$

Comparing the predicted and measured pressure drop, the authors found the following surprising results:

- The effect of the slug length on the pressure drop is small. This can be ascribed to the fact that if L_s/D increased the ratio L_f/L_s decreases, thus increasing the frictional pressure gradient.
- The acceleration term accounts for up to 5% of the total loss, while the viscous losses comprise an important 25% of the total loss.

The flow through a 7.63 cm diameter horizontal pipeline, 20 m long, was investigated (Woods and Hanratty, 1999). The authors found that for a stable slug the following relationship holds:

$$\frac{A_{L1}}{A} = \frac{(0.542 (gD)^{0.5})}{C_B - U_1} \dots\dots\dots (2.59)$$

Where:

- A_{L1} = Liquid area at location 1 (m^2)
 A = Cross sectional area of the pipe (m^2)
 D = Diameter of the pipe (m)
 C_B = Velocity of the back of the slug (m/s)
 V_{C1} = Velocity in the liquid film ahead of the slug (m/s)

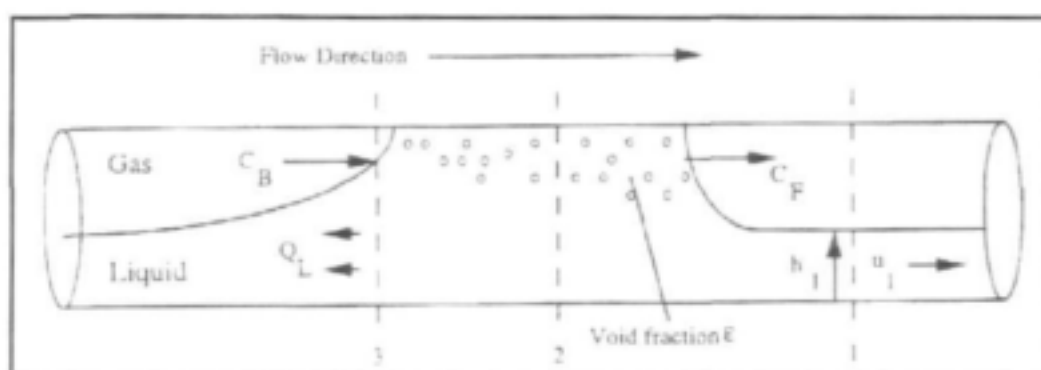


Figure 2.16: Pictorial presentation of the slug

Pressures were recorded in the system to detect the occurrence of slug flow. Presenting the superficial liquid velocity, U_{SL} with that of the superficial gas velocity, V_{SG} on a logarithmic plot, resulted in the following three regimes, **Table 2.2**.

Table 2.2: Slug flow regimes

Regime	Flow type	V_{SL}	Fr	Slugs form at L/D
I	Transition to slug	< 0.1	< 1	> 40
II	VII	$0.2 < 0.3$	> 1	$= 40$
III	Slug flow	> 0.2	> 1	-

In the petroleum industry, gas and oil are often transported simultaneously (De Henau et al. 1995). Terrain-induced slug flow results from the phase separation in hilly terrain. The liquid phase being heavier than the gas accumulated in the valleys, forming liquid bridges that are blown out by the increased gas pressure. This results in complex flow transients. Models to calculate the transient flows are based on the one-dimensional mass and momentum conservation in each phase and by assuming isothermal conditions. The wall friction factors in the liquid slug and in the film regions are evaluated using:

$$f_k = \frac{16}{Re_{hk}} \quad Re_{hk} \leq 2000 \dots \dots \dots (2.60)$$

$$f_k = \left[-3.6 \log \left(\frac{6.9}{Re_{hk}} + \left(\frac{e/D_{hk}}{3.7} \right)^{1.11} \right) \right]^2 \quad Re_{hk} \geq 3000 \dots \dots \dots (2.61)$$

$$f_k = \left(3 - \frac{6000}{Re_{hk}} \right) (f_{turb} - f_{lam}) + f_{lam} \quad 2000 < Re_{hk} < 3000 \dots \dots \dots (2.62)$$

Where e is the pipe wall roughness and D_{hk} is the hydraulic diameter for phase k . Re_{hk} is the Reynolds number based on the hydraulic diameter and is given by:

$$Re_{hk} = \frac{\rho_k u_k D_{hk}}{\eta_k} \dots\dots\dots (2.63)$$

f_{lam} is f_k for $Re_{hk} = 2000$ and f_{turb} is f_k for $Re_{hk} = 3000$. The expression for the turbulent wall friction factor follows the recommendation of White (1986) for a single-phase turbulent pipe flow. The liquid slug region is treated as a homogeneous and isotropically distributed mixture, for which case the hydraulic diameters for the gas and the liquid phases are:

$$D_{HL} = \frac{4A_L}{S_L} = D \qquad D_{HG} = \frac{4A_G}{S_G} = D \dots\dots\dots (2.64)$$

The film region is treated as a stratified flow and:

$$D_{HL} = \frac{4A_L}{S_L} = D \qquad D_{HG} = \frac{4A_G}{S_G + S_L} = D \dots\dots\dots (2.65)$$

The interface friction factor in the film region, f_i , is evaluated using the correlation of Miya et al. (1971) for wavy stratified flow:

$$f_i = 0.008 + 2.0 \times 10^{-5} Re_L^* \dots\dots\dots (2.66)$$

with $Re_L^* = m_L / \eta_L S_L$ is the liquid mass flow in the film and η_L is the liquid viscosity.

The authors (de Henau and Raithby, 1995) compared the measured and calculated pressure drops in slug flow. The experimental set-up consisted of a PVC pipe with an internal diameter of 53 mm and a length of 89.9 m. The pressures were recorded at 5 positions along the line. In the case of a steady-state air-water slug flow in a horizontal pipe the computed and measured gradients fall within 15% of each other. The results also coincided with the results obtained by Theron (1989)(Cited in (Henan et al, 1995)). In the model, there were slugs positioned at about 1.0m intervals. Based on the graphical presentation of the results the following relationship was derived for the slug flow:

$$\frac{DP}{dx} = 265V_L \left(\frac{Pa}{m} \right) \dots\dots\dots (2.67)$$

With:

$$V_L = \text{Liquid velocity (m/s)}$$

2.16 Bubble dynamics in closed conduit flow

Experimental work was conducted (Cook et al. 2000) to determine the slug lengths in a 55mm diameter tube at a positive slope of 5°. Although slug flow does not commonly occur in pipelines conveying water it remains important to include these findings to reflect the existing models in order to determine the pressure drop. Ducker and Hubbard (1975) indicated that an estimated average slug length has to be obtained to enable the researchers to determine the pressure drop and hold-up. Furthermore, designers are interested to establishing the slug size distribution and the maximum slug length. In all the cases the minimum slug length was close to 10 diameters in length.

The authors (Coleman and Garimella, 1999) investigated the influence of the pipe diameter and shape on the transition between different phases. The experiments were conducted with air and water superficial velocities from 0.1 to 100 m/s and 0.01 to 10.0 m/s respectively.

The quantification of the flow regime is important since the pressure drop in two-phased flow is much greater than in one phased flow. The definitions used to describe different flow conditions are mutually contradicting, and it was essential that the flow regimes be defined. **Table 2.3** defines the four difficult flow regimes used in their study.

Table 2.3: Flow regime classification

Flow regime classification		Superficial velocities	
Major Region	Flow pattern	V_L (m/s)	V_g (m/s)
Stratified	Stratified smooth	< 0.1	$2 < V_g < 12$
	Stratified wavy		
Intermittent	Elongated bubble (plug)	< 2.0	< 8.0
	Slug flow		
Annular	Wavy annular	< 1.0	$4.0 < V_g < 11$
	Annular		$V_g > 11$
Dispersed	Bubble	> 1.0 – 2	< 1.2
	Dispersed		> 2.0

The following paragraphs describe these flow regimes.

Stratified flow regime

The stratified flow regime is characterized by a complete separation of the liquid and gas phases. When both the liquid and gas flows are laminar and no fluctuations at the flow interface can be detected, the flow pattern is called stratified (stratified smooth). As the gas mass flow rate is increased, instabilities form at the liquid-gas interface, due to the interfacial velocity differential (Kelvin-Helmholtz instability). This flow pattern is called wavy flow (stratified wavy) and is characterized by the formation of small interfacial waves. In larger diameter tubes, these waves can amplify and crest. The waves are easier to detect in large diameter tubes and the wave height can be large enough to allow the waves to break. In small diameter tubes, such as the ones used for this study, large breaking waves were not observed.

Intermittent flow regime

The intermittent flow regime is characterized by discontinuities in the liquid and gas flows. Elongated bubble flow (plug flow) is characterized by a continuous stream of vapor plugs flowing in the liquid. A thin film of liquid coats the tube wall and surrounds the vapor plug. Small disturbances may exist before and after the bubbles, but as a whole the plugs remain intact and uniform. As the gas mass flow rate is increased, these disturbances amplify until the tail portion of the plug breaks apart into smaller bubbles. At this point, the flow pattern becomes slug flow. These smaller bubbles become trapped in the liquid flow, and impact the front of the following slug causing disturbances in the front flow profile.

Annular flow regime

The annular flow regime consists of a nearly complete separation of the liquid and gas along the circumference of the tube wall. The first form of annular flow occurs when the surfaces of waves in wavy flow amplify to the extent that they touch the top of the tube wall. This flow pattern is known as wavy-annular flow (pseudo-slug flow). As the mass flow rate is increased, the liquid is pushed up around the circumference of the tube wall by the increase in the gas momentum and falls downward under gravity in the form of annular waves.

When the liquid completely coats the tube wall (forming an annular ring of the liquid phase) and the gas flows through the core of the tube, the flow pattern is known as annular flow. In annular flow there could also be small droplets entrained in the gas core. This flow is also known as annular mist flow.

Dispersed flow regime

Dispersed flow occurs when the liquid flow is turbulent and the gas phase is in laminar or turbulent flow. When the gas flow is laminar, small bubbles are driven by buoyancy forces and flow primarily in the top half of the tube. This pattern is known as bubble flow. As the Reynolds number of the gas increases, keeping other variables constant, the bubble size decreases and the bubbles begin to disperse across the entire tube cross section. This flow pattern is known as dispersed bubble or dispersed flow.

The following relationships, developed by Weisman et al (1979) reflect the transition between the different flow regimes.

The transition from stratified flow to intermittent flow is given by:

$$\frac{V_g^{1.5}}{(gD)^{1/2}} = 0.25 \left(\frac{V_g^{1.5}}{V_L^{1.5}} \right)^{1.1} \dots\dots\dots (2.68)$$

This correlation appears to imply that liquid density, liquid viscosity, and surface tension have no effect on this transition line.

The transition to annular flow is given by:

$$1.9 \left(\frac{V_g^{1.5}}{V_L^{1.5}} \right)^{1.8} = \left(\frac{V_g^{1.5} \rho_G^{1.2}}{[g(\rho_L - \rho_g)\sigma]^{1.4}} \right)^{0.2} \left(\frac{V_g^{1.2}}{(gD)} \right)^{0.18} \dots\dots\dots (2.69)$$

This correlation appears to imply that liquid viscosity has no effect and liquid density and surface tension has only minor effects on this transition line.

The transition to dispersed flow is given by:

$$\left[\frac{dP/dx}{(\rho_L - \rho_g)g} \right]^{1/2} \left[\frac{\sigma}{(\rho_L - \rho_g)D_g^2} \right]^{0.25} = 9.7 \dots\dots\dots (2.70)$$

Where:

$$(dP/dx)_L^* = \frac{f}{2} \frac{G_L^2}{g\rho_L D} \dots\dots\dots (2.71)$$

Coleman et al (1999) confirmed these relationships.

2.17 Economics of ineffective de-aeration

Design engineers or operators of water distribution systems often refer to the problems associated with air valves such as leakage that occur at the valves as well as the cost associated with the installation of air valves. These might be valid arguments but it is outweighed by the capital and operational cost associated with failures, which can be attributed to the lack in capacity of effective de-aeration and the prevention of vapour conditions.

The experimental findings with regard to the energy loss (secondary losses) can be used to reflect the economic influence of ineffectively de-aerated systems on the hydraulic capacity and hence the potential income.

Experiences have been documented where pipelines under performed due to the presence of air in the system.

Based on the hydraulic transportability of air, all the apex points in a pipeline can be evaluated. At these points where the air will not be transported the air will result in a secondary energy loss. Assuming that the loss will be factor of the kinetic energy, the power requirement in pumping pipelines will increase or the capacity in a gravity pipeline will decrease. If the life cycle cost of this variation is calculated, based on the selling prize of water and the cost of capital, a monitory value can be set to the presence of air. This aspect was incorporated in the air valve utility software that was partially developed under this project.

3. FIELD INVESTIGATIONS

3.1 Introduction

At the onset of the project it was planned to include a medium as well as a large diameter pipe in the field investigation. With the cooperation of Umgeni Water and Rand Water, the Hazelmere – Ballito (Avondale) pipeline (700 mm steel), constructed in 1995, and the B8 – Zuikerbosch to Mapleton pipeline (2100 mm diameter steel with cement mortar lining) were identified respectively identified. Both these pipelines are pumping mains.

The intention with the field tests were to:

- Determine the effectiveness of de-aeration of the pipelines.
- Investigate the movement of air in pipelines, and
- Establish the influence of air on surge wave propagation in order to determine a relationship, which could be used to identify the presence of air in the system.

The available information for these pipelines was obtained and evaluated and surveys were conducted to provide unknown information.

During the course of the project, the project team was informed about a problem that the City of East London was experiencing on the Buffalo River Raw Water Supply Line. These problems were related to air in the system and hence the system was added to the fieldwork program.

The above-mentioned fieldwork was selectively conducted on the different pipelines. The data acquisition, testing and results from these investigations are discussed separately in the paragraphs that follow.

The following aspects, where applicable, are included in the discussion:

- Description of the system
- Data acquisition
- Procedures for the field tests
- Ability to de-aerate effectively
- Capacity to transport the air
- Identification of the presence of air in the pipeline by using inverse transients
- Conclusion

3.2 Field investigations: Hazelmere – Ballito (Avondale) pipeline (Umgeni Water)

3.2.1 Description of the system

Information on the Hazelmere – Ballito (Avondale) pipeline was obtained from the As-built drawings. The pipeline is about 9.8 km long and runs over undulating terrain along the N4 highway in a northerly direction. Along the route there are a number of small off-takes for domestic consumption by farmers. The water is fed from the Hazelmere purification plant through 2 BPK 42-4 4-stage pumps to the Ballito/La Mercy bifurcation point. From this point the pipeline is known as the Avondale pipeline. **Figure 3.2.1** depicts the location of the pipeline relative to the surrounding features. The longitudinal profile of the pipeline is reflected in **Figure 3.2.2**.

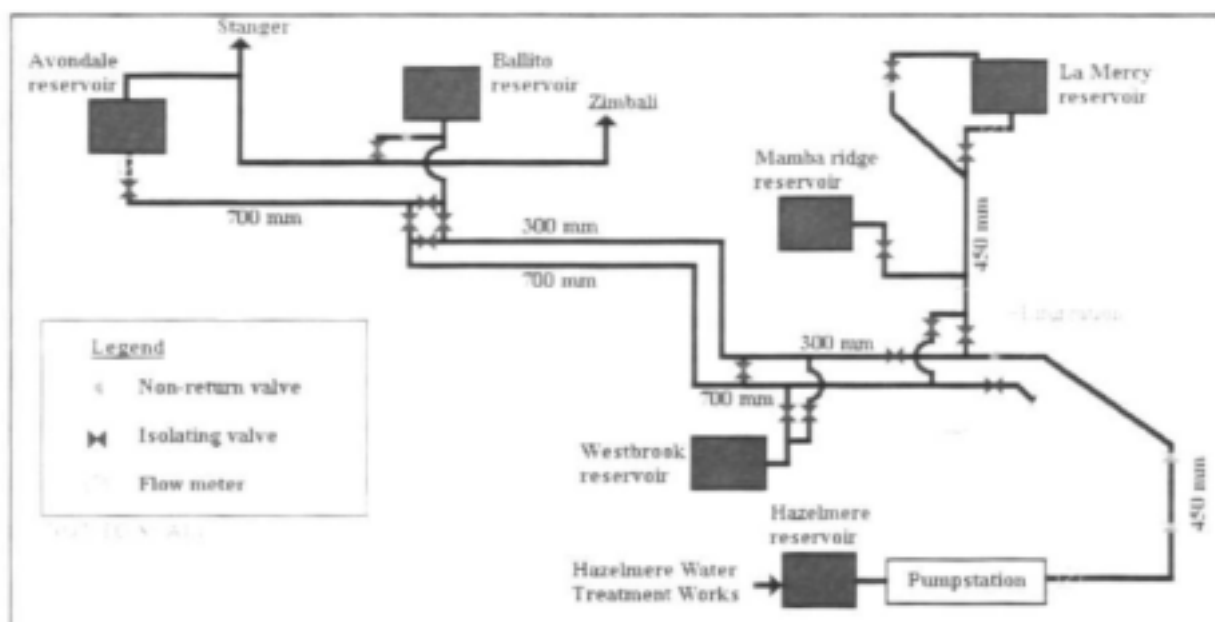


Figure 3.2.1: System layout



Figure 3.2.2: Hazelmerre – Ballito (Avondale) longitudinal profile

3.2.2 Data acquisition

The Avondale pipeline was inspected and data of all the air valve chambers and scour valves were recorded and compared to the As-built drawings. **Figures 3.2.3 and 3.2.4** contains typical details of the scour valves and the air valve chambers along this pipeline.

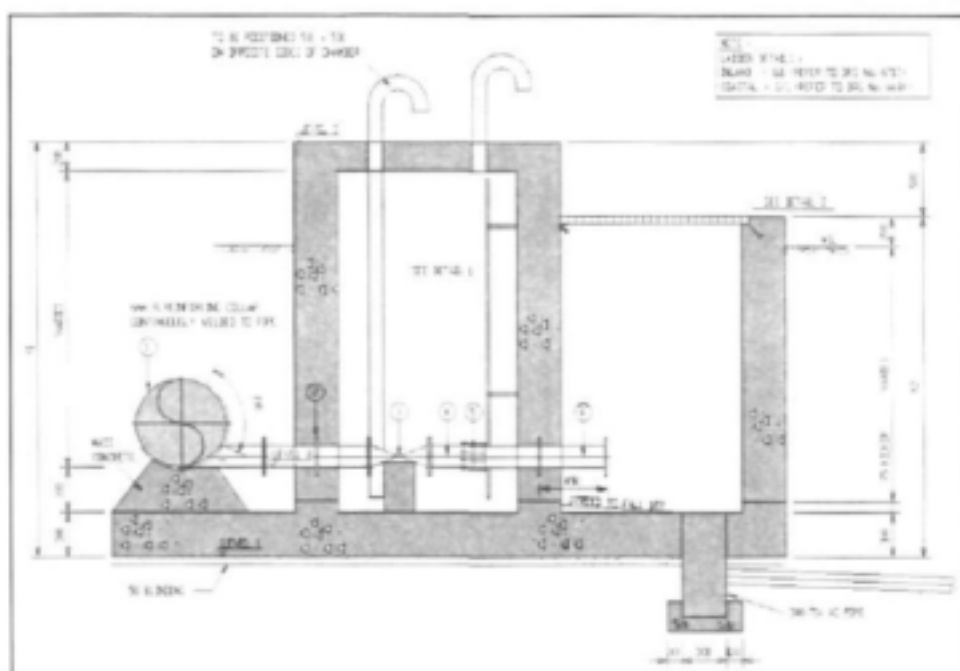


Figure 3.2.3: Typical scour chamber details of the Avondale pipeline

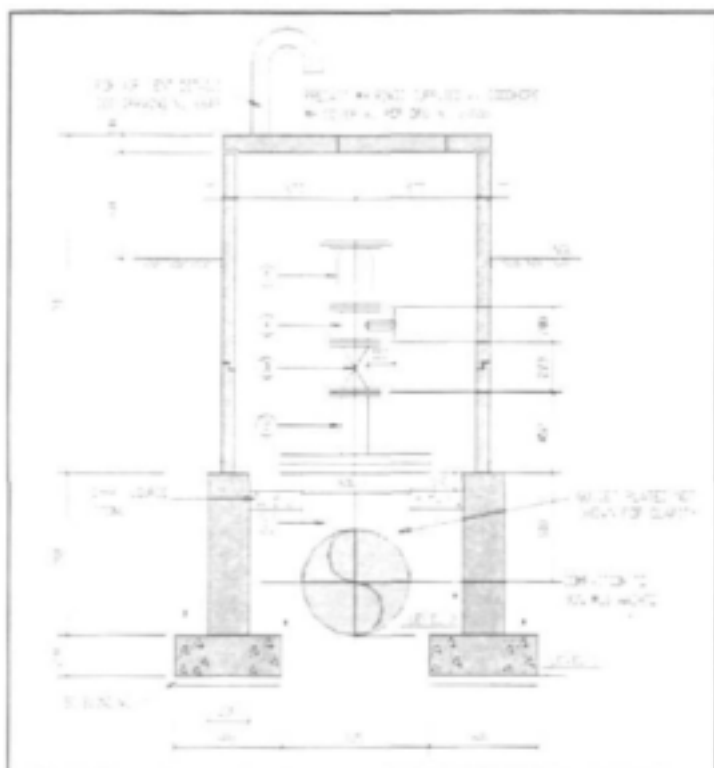


Figure 3.2.4: Typical air valve chamber details of the Avondale pipeline

The available data and description of the operation of the system was used in the preparations for the field investigations. The objectives of the fieldwork on the Avondale pipeline were to:

- Determine the capacity of the pipeline to transport the air.
- Establish the ability to de-aerate effectively by intercepting the air at the air valves, and
- Identification of the presence of air in the pipeline by using inverse transients.

These tests required selection of equipment and data acquisition systems to record the data effectively. The equipment required for these tests are reflected in **Table 3.2.1**.



Figure 3.2.5: Air induction facility (compressor)

Table 3.2.1: Equipment used for the field work on the Avondale pipeline

Function	Items	Description	Technical specification	Figure
Transport on site	Truck	Supplied by Umgeni Water and project team	4x4	
Air induction facility	Compressor with power generator	Air supply to be introduced to the pipeline. Hired by Umgeni Water	Operating capacity required was 20 m ³ /s at a pressure of 10 Bar	3.2.5
Controlling the air volume introduced to the pipeline	Air Induction Box	High-accuracy regulator with fast acting solenoid valves controlled by a programmable PLC to size the bubble and control the pressure	Details are described in Section 4	3.2.6
Communication	Cell phones and two-way radios	Supplied by Project team and Umgeni Water		
Data acquisition hardware	Mercer and HP laptops with supplementary power packs for long recordings	Capturing of data at 1 000 Hz	128 RAM minimum and 1GB Hard drive	3.2.6
Pressure measurement	Pressure transducers	Various pressure transducers were supplied by the project team	10 to 25 Bar	
Data acquisition software	3 x Quatech PCMCIA cards with software	PCMCIA card to be able to record 10 volts	Capable of recording at 1 000 Hz DAQP-208	
Air valve (measurement of air release)	Controlled air valve release orifice with temperature and pressure tapings	Constructed by Vent-O-Mat	Connector flange to connect onto the air valve stub with known characteristics of the small orifice release	3.2.7
Creation of the outflow	Fast Acting valve	The 150 mm butterfly valve with an extended spindle could be used to generate flow at scour valves (supplied by Vent-O-Mat)	Pressure rating of the valve was 10 Bar	3.2.8
Air capturing facility	Box with flange to connect to the grid of the air valve and connection to connect an air balloon to capture the air	Manufactured by the project team	Air capturing device with a large orifice to prevent negative pressures. The bias mechanism will be used to protect the pipeline against negative pressures	3.2.9



Figure 3.2.6: Air Induction Box used in field setup

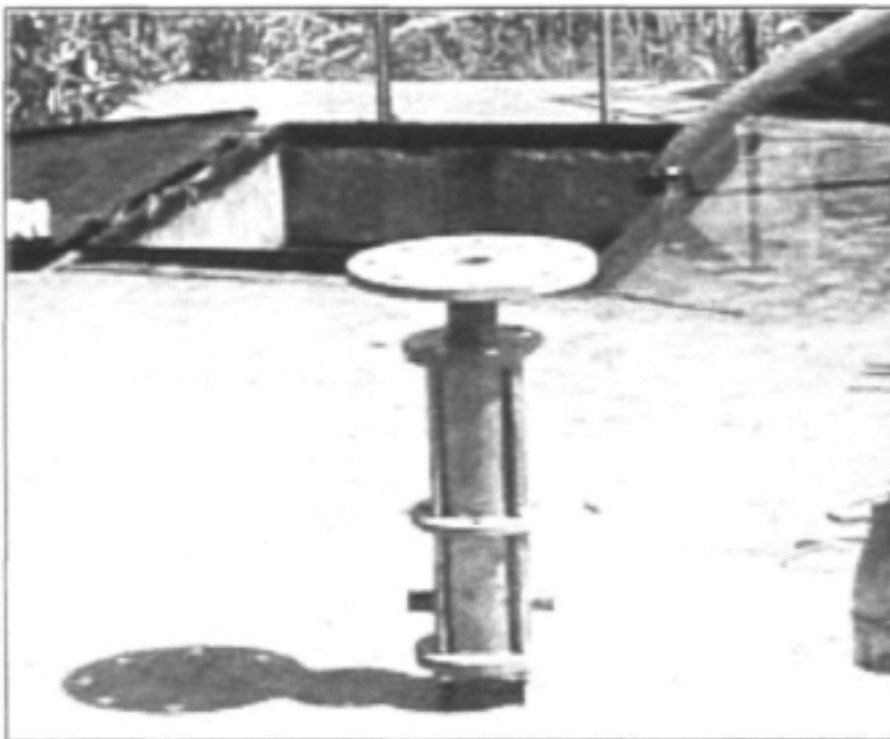


Figure 3.2.7: Air release measuring apparatus

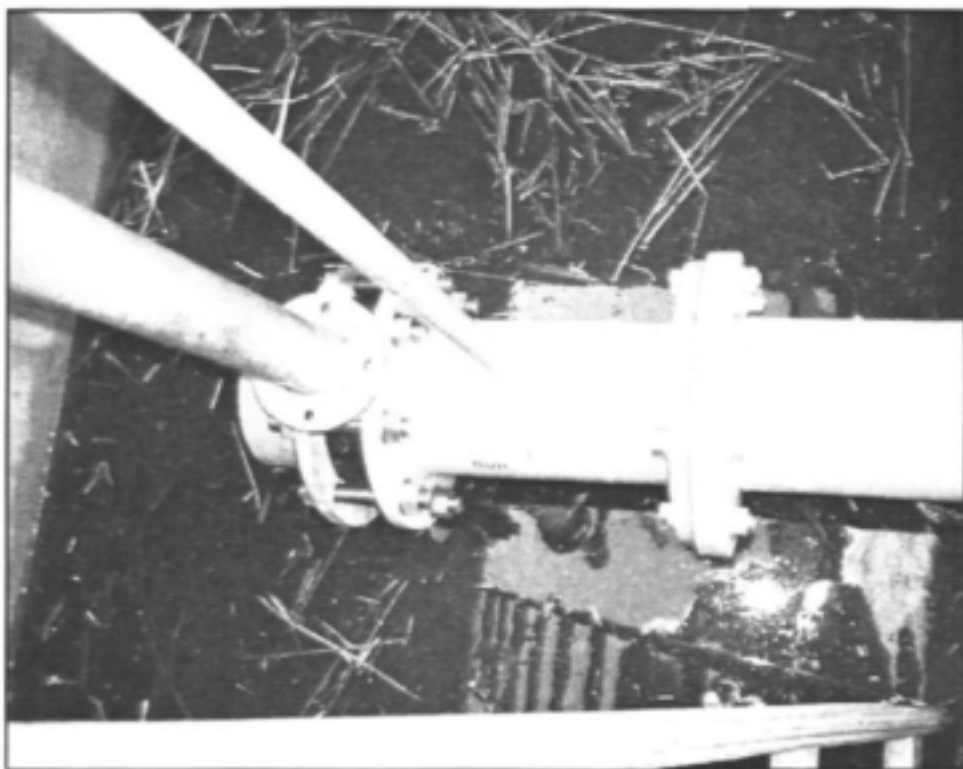


Figure 3.2.8: Fast acting valve for creating negative transient

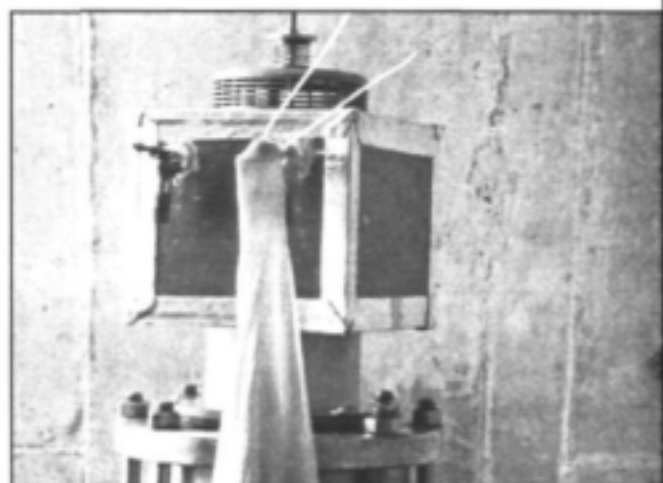
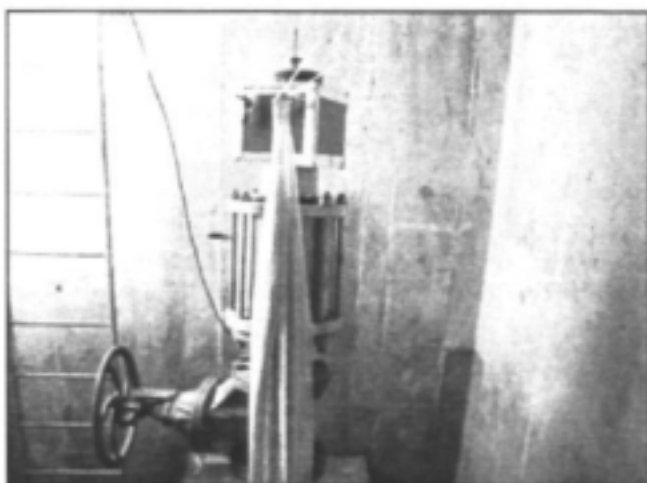


Figure 3.2.9: Air capturing facility on top of air valve

Depending on the location and the tests conducted, different equipment was used as listed in Table 3.2.1.

3.2.3 Procedures for the field tests

To evaluate the different aspects, the following actions were undertaken.

3.2.3.1 Hydraulic transportability of air

Air was introduced at a point in the pipeline with the objective to establish whether it could be transported down the descending sections of the line to the next air valve position. **Figure 3.2.10** reflects the section between chainage 8432 m and 9730 m that was selected for this test. Air was introduced at an air valve at chainage 8531.50 m (point A) while the air valve at chainage 8936.53 m (point C) was isolated and the release from the next air valve at chainage 9728.57 m (point E) was monitored to determine whether the air was transported downstream. Throttling of the pumps at the pumpstation varied the flow rate.

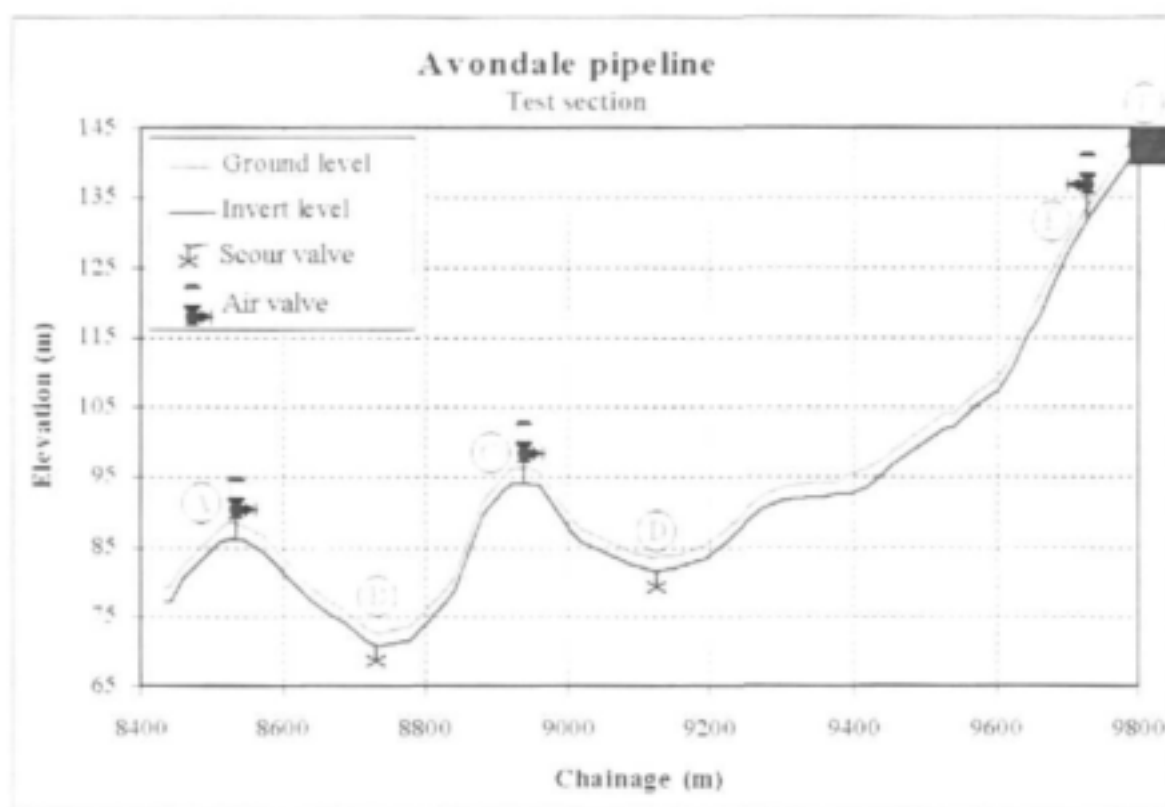


Figure 3.2.10: Section used for the test

Table 3.2.2 reflects the pipe profile for the test section shown in **Figure 3.2.10** where the slopes are negative. The negative slopes can be crucial control points in the system, preventing air being transported further down the line.

Table 3.2.2: Negative slopes of the test pipeline section

Begin chainage (m)	End chainage (m)	Begin invert level (m)	End invert level (m)	Slope (m/m)	Slope (%)	Angle (°)
8531.50	8547.46	86.26	86.01	-0.01566	1.566	0.897
8547.46	8574.59	86.01	84.14	-0.06893	6.89	3.943
8574.59	8618.39	84.14	79.39	-0.10845	10.85	6.189
8618.39	8642.98	79.39	77.01	-0.09679	9.68	5.528
8642.98	8666.70	77.01	75.13	-0.07926	7.93	4.532
8666.70	8690.22	75.13	73.90	-0.05230	5.23	2.994
8690.22	8717.62	73.90	71.48	-0.08832	8.83	5.047
8717.62	8733.14	71.48	70.83	-0.04188	4.19	2.398
8936.53	8945.68	94.09	94.06	-0.00328	0.33	0.188
8945.68	8960.05	94.06	93.65	-0.02853	2.85	1.634
8960.05	9007.10	93.65	86.94	-0.14261	14.26	8.116
9007.10	9018.93	86.94	85.54	-0.11834	11.83	6.749
9018.93	9090.00	85.54	82.49	-0.04292	4.29	2.457
9090.00	9120.22	82.49	81.60	-0.02945	2.95	1.687

The buoyancy characteristic of air will enable it to be transported along a positive slope. When the slope however becomes negative the force of the water attempting to transport it down the pipeline might be less than the buoyancy force working in the opposite direction. The steeper the slope, the greater the buoyancy force and therefore a higher flow velocity is required to transport the air bubble.

None of the air introduced at the induction point (point A) was transported down the line towards the observation point (point E) at chainage 9728.57 m. This result was significant because it reflected that with a flow velocity of 0.75 m/s in the 700 mm pipeline, it was not possible to transport the air down a slope of 14.3% (the steepest section).

This is a significant result, which was then compared with the required velocities suggested previously for the hydraulic transport of the air, as well as with new experimentally determined formulae, described in detail in Section 4.2 of this report. **Figure 3.2.11** reflects that at a flow velocity of 0.75 m/s and at a slope of 14.3 % it was not possible to transport the air hydraulically. Even on less steep slopes the air would not be capable of being transported, see Section 4.2.

The system was further analysed by looking at the entire Hazelmere – Ballito (Avondale) pipeline and determining all the sections where air will not be transported hydraulically. The total length of the pipeline is approximately 9.8 km and the indication is that about 3.2 km of the line will not be able to transport the air hydraulically for a flow velocity of 0.75 m/s.

Although the system has sufficient entrapment capability along the line to capture air, any free air that gets past these discontinuities underneath the air valves will accumulate along the negative slope sections where it cannot be transported any further.

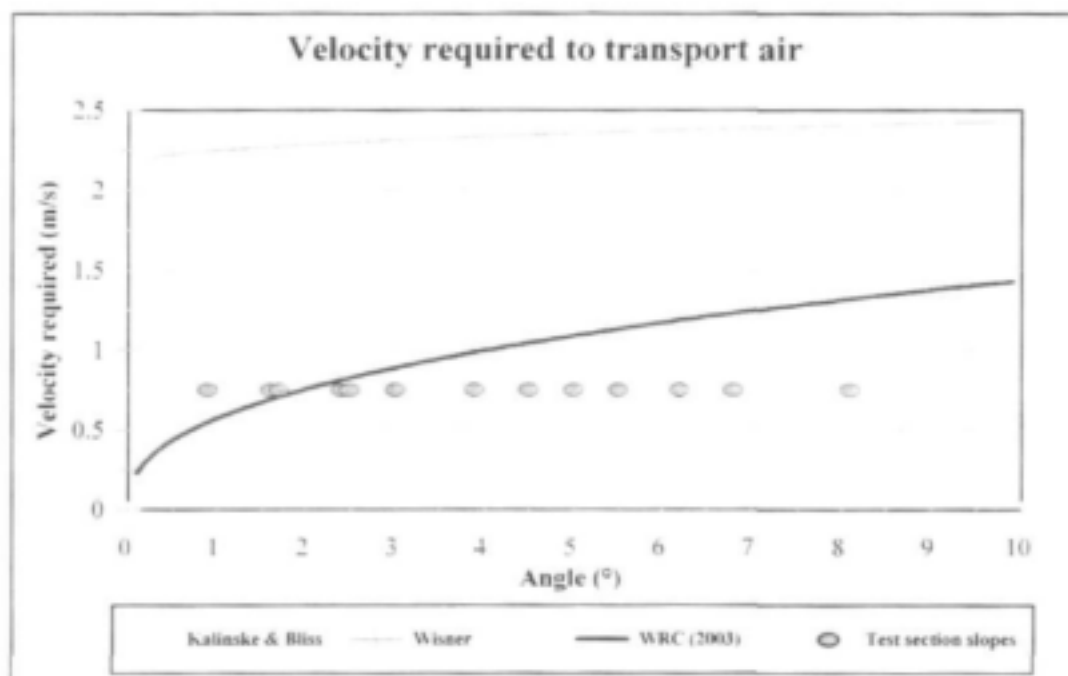


Figure 3.2.11: Test section slopes compared to previous research and proposed formulae for transporting air hydraulically (see Section 4.2)

3.2.3.2 Interception of air at the air valves

For this field investigation, air was introduced upstream of an air valve at a specific rate and the release through the air valve was monitored. To be able to measure the release of air a "chamber" is required that will capture the air and release it in a controlled manner through a specific opening size. By measuring the pressure time relationship and assuming that the temperature remains constant the volume of the air released through the orifice can be calculated (**Figure 3.2.7**). Maintaining safe operation of the line requires that the large orifice function should be available to prevent vacuum conditions. This required a change in the air valve to maintain the large and anti-shock function in the valve and then bleeding off the air under operating conditions to an add-on valve that will be able to release the air through the small orifice.

It was indicated that the small orifice would have to be large enough not to function as the control and hence the air would be released into a small chamber from where it would be released into the atmosphere.

The small orifice of the chamber would always be open and would release the air. The valve used was then changed to allow the air to be captured and measured (**Figure 3.2.9** and **Figure 3.2.12**).

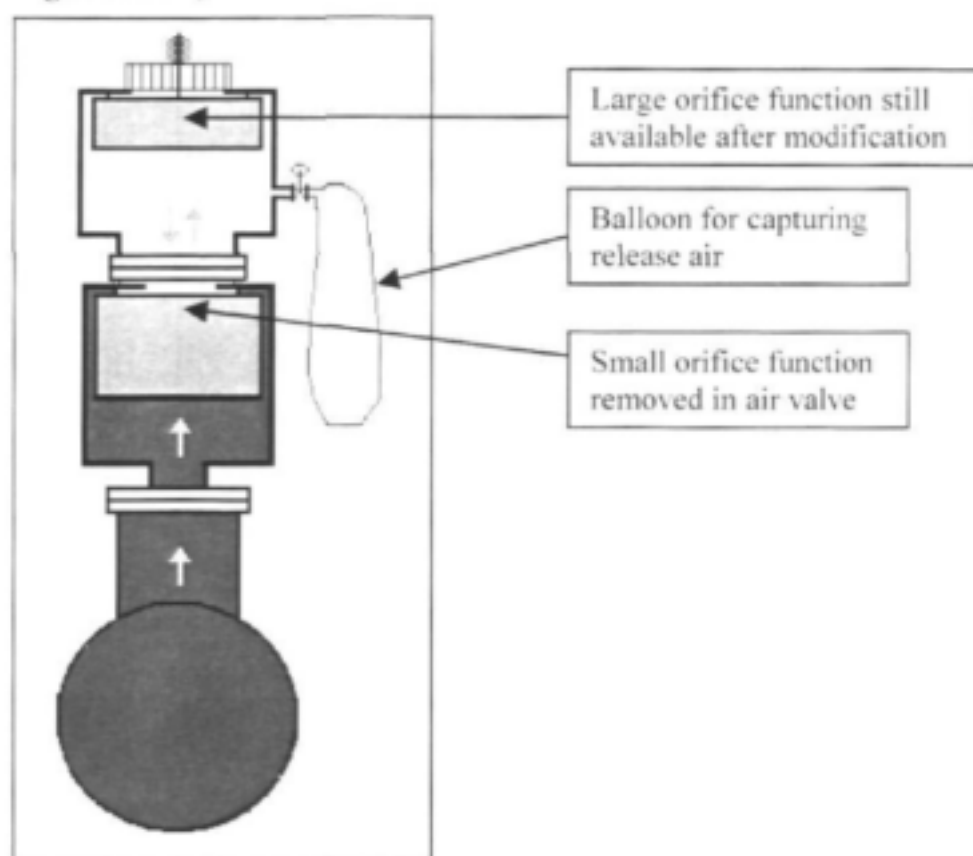


Figure 3.2.12: Modification to air valve layout

This test proved to be impractical, due to the extreme variances in the flow rate of the air through the measuring device. The variation in the flow rate created high pressures in the capturing device and made it unsafe to try and capture the air.

A quantitative assessment of the air valve installation chambers, where equal T-pieces had been used and sufficient accumulation volume underneath the air valve had been provided, led to the conclusion that the de-aeration capacity of the pipeline is effective. This is a conservative design in terms of the discontinuity and if the accumulator is big enough, all the air transported to the air discontinuity will be captured and released.

3.2.3.3 Identification of the presence of air in the pipeline using inverse transients

A recent application of transient flow is the so-called "inverse transient movement". This procedure creates a pressure fluctuation and trails the movement of the pressure wave through the system. At locations where air is present, the wave's magnitude will change. This is used to identify the location of air and the extent of the air. For the protection of the pipeline a negative surge wave was introduced by the creation of flow through the scour valve. The section of the pipeline used for this test is depicted in **Figure 3.2.10** (Section A to F).

Air was introduced at chainage 8528.50 m (**Figure 3.2.10**, point A) and a fast acting valve (**Figure 3.2.8**) was installed on the scour valve at chainage 9120.22 m (point D). Pressure recordings were taken upstream and downstream from the point where the surge wave was created, at points A and E respectively.

The results indicated that the wave celerity at point A reduced between point D and A. The reduction in wave celerity was, however, not significant to attribute it solely to the introduced air at point A. This indicated or confirmed the presence of air along the downward slopes between Point A and D. Calculation of the wave speed celerity revealed that the air bubble of 300 nm³ (operating pressure was about 67 m) at chainage 8528.50m (point A) as well as the free air trapped on the downward slopes reduced the celerity by about 24 %. The downstream reaction was registered 0.640 seconds after the creation of the pressure variance at point D. The wave celerity for this section where no air can be present (positive slopes) of 950 m/s compares favourably with the theoretical calculation of 984 m/s. Point A measured the pressure variance after 0.820 seconds indicating a wave celerity of approximately 721 m/s.

Some of the data sets were influenced by the time shifts of the computer clocks and were discarded. Unfortunately a mechanical problem with the fast acting valve prevented any further testing. **Table 3.2.3** contains some details of the tests.

Table 3.2.3: Results obtained from the tests

Location and description	Chainage (m)	Calculated wave celerity (m/s)
Upstream pressure measuring point	8528.50	721
Point where the negative pressure was created by opening and closing of the valve	9120.22	Na
Downstream location where the pressure was measured.	9728.57	950

When measuring the wave celerity in a pipeline, at least two positions are required where the pressure variation is recorded. The combining of the sets of measured data onto a combined time axis to enable the determining of the wave celerity is intricate. The recordings should start at exactly the same time to enable accurate calculation of the wave celerity. For this a special triggering device was designed and built to synchronise the data capturing when data acquisition is done on separate systems.

3.2.4 Conclusion

The system's capacity to transport air hydraulically is insufficient for certain flow conditions and on specific section of the line. The interception capability is however sufficient to release the air when it reaches the air valve. If air is released from solution on a downward sloping section that has a too steep slope the air will be trapped. The system's design capacity is 0.8 m³/s but is however sometimes operates at 0.3 m³/s. This exaggerates the problem with regard to hydraulic transportability of the system.

If a pipeline has not sufficient transportability capability it might be found that the flow rate would reduce during the day as air starts to accumulate causing additional head loss. This will especially happen at points where there is a slope change from a negative slope to an even steeper negative slope. Once the flow has stopped at the end of the supply period the air will now have the opportunity of moving backwards towards the high points in the line where it is then removed since most peaks are designed with a small orifice air valve.

Pipelines are mostly designed for supplying in the future demand. This might result in operating a pipeline at a lower flow rate (more cost effective) during the beginning of the pipeline design life and a subsequent lower velocity. The low velocity might now result in localised air bubbles forming since in the flow velocity is less than the required velocity to transport the air hydraulically.

From the inverse transient tests it could be deducted that there is free air in the system. During the tests a large air bubble suddenly reached the end measuring point (point E). It was believed that this air was trapped just upstream of the scour valve (point D). The sudden change in flow created at the scour valve increased the flow past the air bubble significantly enough to dislodge it and travel all the way to the last measuring point. This strengthens the theory that at the change in slope the air could not be transported further and started to accumulate. The identification of the presence of air in the pipeline by using inverse transients, however, was not as successful as was hoped although a reduction in wave celerity was found. A mechanical problem with the fast acting valve prevented the completion of the planned tests. This procedure will however still be performed in future on the Rand Water pipelines.

3.3 Field investigations: Zuikerbosch - Mapleton Pipeline (Rand Water)

3.3.1 Introduction

The experience obtained from the tests conducted on the Hazelmere – Ballito (Avondale) pipeline highlighted the following problems:

- *The operating pressure in the lines were outside the capacity of compressors that could be obtained for air induction (within the budget constraints)*
- *The measurement of air release is difficult if the functionality of air valves, as vacuum breakers, should be maintained to protect the pipe during the tests.*
- *The layout configuration of the air valve accumulator resulted in sweeping some of the air along the pipeline, leaving an undefined volume of air in the accumulator.*
- *Capturing the air that is released also proved to be difficult (and dangerous).*

This forced the project team to review the field tests, especially for the larger diameter (2100 mm diameter) B8 pipeline.

There were a number of reasons why the B8 pipeline was identified as a possible pipeline to be included in the fieldwork.

- Rand Water indicated that they were experiencing a loss in capacity on this line. The system was operating at a much lower flow rate of 500 Ml/day instead of the design flow rate of 600 Ml/day i.e. 5.78 m³/s versus 6.94 m³/s.
- Rand Water found that there was a discoloration of the cement mortar lining in the crown of the pipe, indicating that air must have been present for substantial periods of time.
- The system was constructed in 1976 in accordance with the design principles of those days. New design principles as well as new air valve technology have resulted in improved designs. If some of these new principles could be applied i.e. if some changes/alterations to the pipeline could be made, the improvement could be determined. A comparison between the old and new air valve design procedures can be made, based on the change in hydraulic capacity.
- The new B16 pipeline running parallel to the B8 pipeline also allows for a direct comparison between the "new design" and an "old design" in terms of the influence of air valves, discontinuities and accumulators on the hydraulic capacity.

The B8 line runs in a northerly direction from the Zuikerbosch purification plant at the Vaal River to the Mapleton reservoir south east of Johannesburg. Rand Water has experienced a loss in the capacity of this pipeline and during an inspection of the pipeline locations were found where the crown of the pipe was discoloured, suggesting the presence of air in the line.

Information about the profiles of the pipelines is available for both the B8 and the B16 pipelines, to initially identify possible testing sections. Based on experience and the difficulty to conduct some of the tests, it was decided to conduct the following tests on the B8 pipeline:

- Quantitatively evaluate the efficiency of the existing air valves
- Evaluate the presence of air in pipelines by using the inverse transient procedure.

3.3.2 Description of the system

The B8 pipeline runs from the Zuikerbosch purification plant to the Mapleton reservoir some 47 200 m away. This steel pipe was constructed during 1976. Various internal protection coatings were used for the cathodic protection of the pipeline, as well as different wall thicknesses.

Relevant data regarding the B8 - Zuikerbosch to Mapleton pipeline was obtained from construction drawings and other internal reports and documents from Rand Water. This information was used in the planning of the physical site inspections.

Lack of detailed information with regard to the air valves, necessitated field surveys of all the valve chambers. Table 3.3.1 was used to record the information. During the field investigations, the following minor problems were experienced:

- A short section of the pipeline in Vosloorus could not be inspected (safety reasons).
- Since the construction drawings and the installed chambers differed, it was difficult to locate the position on the pipeline. The valve chambers were all marked with AV1 - air valve 1 or SV4 - scour valve 4, to ensure that once the pipeline was surveyed the exact locations of the constructed valve chambers could be established.
- A few valve chambers could not be opened (covers welded or stuck).
- Some valve chambers were under water (high ground water table) thus restricting access.
- The working of none of the air valves was tested.

- Most air valve layout configurations were identical, consisting of a small orifice valve on a 100 mm diameter accumulator (stub), approximately 150 mm long and a 200 mm combination air valve (consisting of a small orifice and a large ball type orifice with a 65 mm opening size) on a 200 mm diameter accumulator approximately 150 mm long.

Table 3.3.1 reflects the format in which the data was gathered.

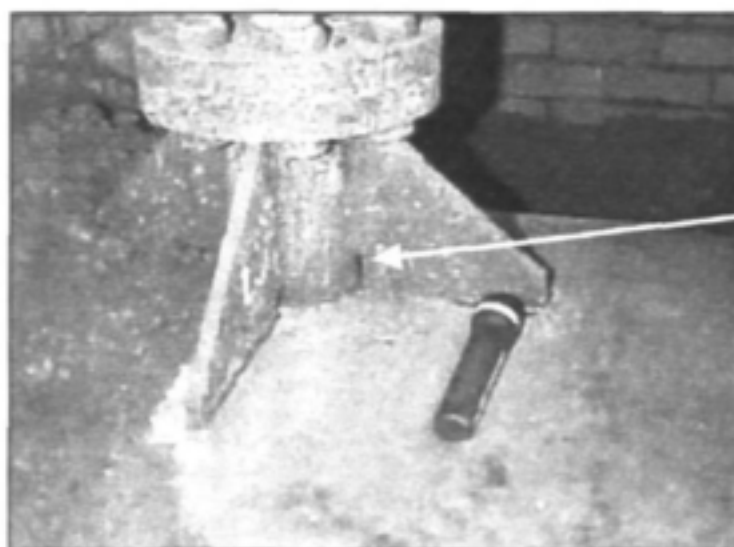
Table 3.3.1: Pro-forma sheet for the recording of the data of the valve chambers

ZUIKERBOSCH TO MAPLETON PIPELINE				
FIELD DATA SHEET				
Chainage (m)				
Elevation (m)				
Valve chamber ID				
Description				
Condition	Seat	Red condition	Accessible	Under water
Valve make				
Size:	Valve			
	Accumulator (L)			
	Accumulator (S)			
Pictures:	Take			
	Numbers			
Chainage (m)				
Elevation (m)				
Valve chamber ID				
Description				
Condition	Seat	Red condition	Accessible	Under water
Valve make				
Size:	Valve			
	Accumulator (L)			
	Accumulator (S)			
Pictures:	Take			
	Numbers			

The data obtained provided the basis for the following actions:

- Determining of critical sections (slopes and poor de-aeration areas). The survey of the pipeline was required because not all chambers were built as initially planned (chainage and elevation on the valve chamber slab with reference to our markings on the chambers).
- To prepare the valves and equipment for the field tests.

The re-survey of the pipeline enabled the incorporation of all the field data and as-built information. **Figures 3.3.1 to 3.3.5** reflect some of the details of the air valves.



100 mm diameter discontinuity
on crown of pipe used for small
orifice function air valve

Figure 3.3.1: Typical 100 mm off take (stub) underneath the air valve



200 mm diameter discontinuity
on crown of pipe used for large
orifice or combination air valve

Figure 3.3.2: Typical 200 mm off take (stub) underneath the air valve



Figure 3.3.3: Typical small orifice air valve



Figure 3.3.4: Typical combination air valve

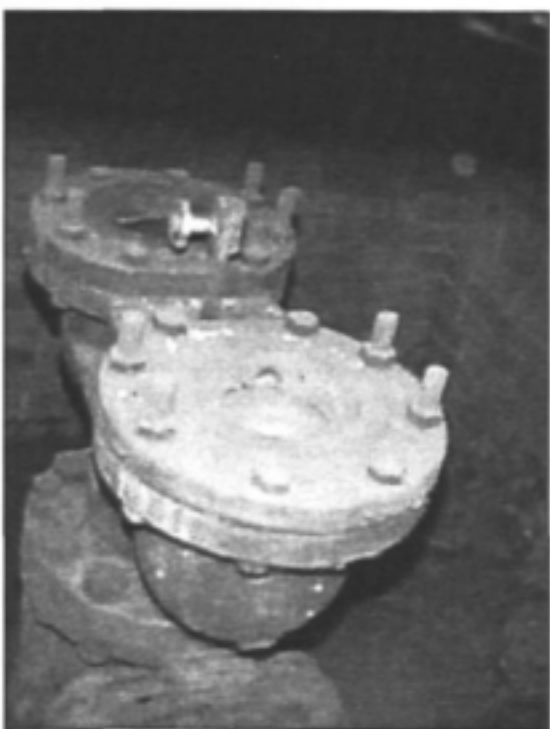


Figure 3.3.5: Typical combination air valve without the protection plate above the large orifice

Details obtained from the longitudinal profile drawings were used to determine locations where measurements could be made (Figure 3.3.6).

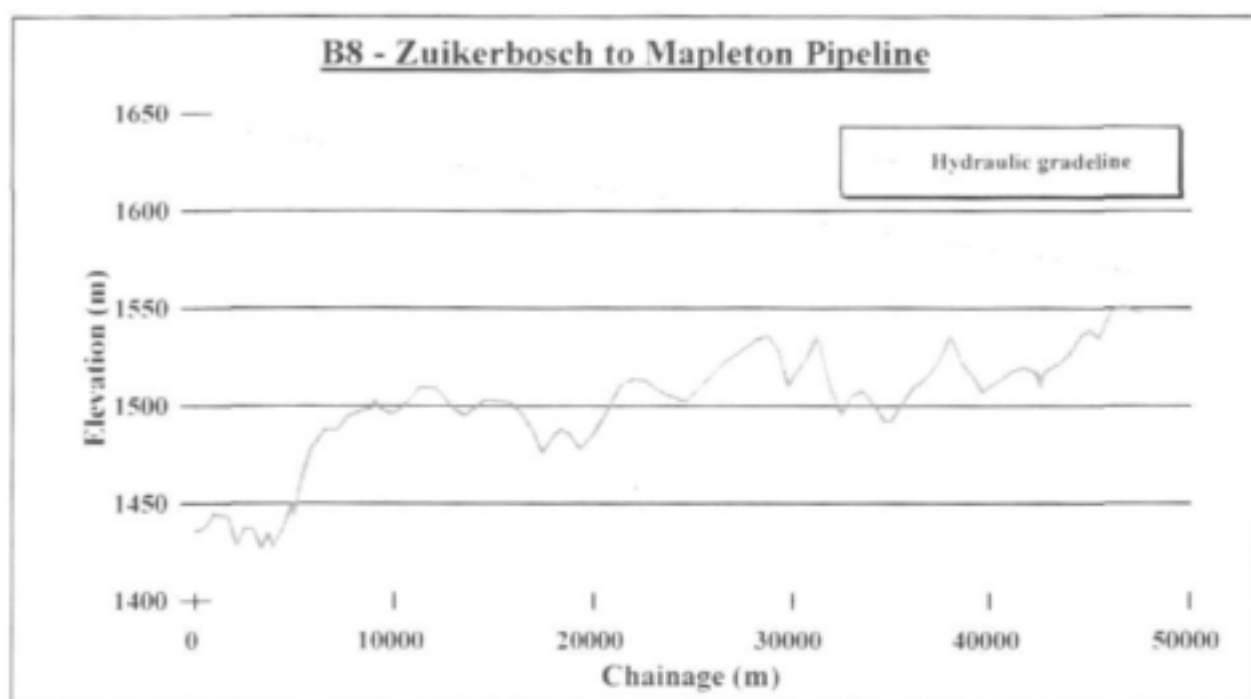


Figure 3.3.6: Profile and hydraulic gradeline

Using the relevant information, the pipeline was analysed to provide a first indication of the changes that will have to be implemented with regard to the air valves when the B8 pipeline is refurbished. The pipeline profile data was used and the system was analysed for drain and pipeline bursts/failures, utilizing the Vent-O-Mat Utility Program. This was done to determine the critical sections on the pipeline that would have the most significant influence if these were to be changed (new de-aeration system), see **Appendix 3.3.1**.

3.3.3 Procedures for field tests

3.3.3.1 Identification of the presence of air in the pipeline, using inverse transients

At the Steering Committee Meeting held in November 2001, Mr Trebicki of Rand Water indicated that some tests were to be conducted on the line and that the use of inverse transients, which had been proposed for the B8 line, could possibly be of value for those tests on the B16 line. The project team was invited to participate in the tests.

In February 2002 the testing was discussed with Messrs Trebicki, Grobler, Botha and Wilson from Rand Water. The sections on the B16 pipeline that were selected for these tests are:

- First section – referred to as section 1 (suspected leak) and
- Last section – referred to as section 4 (possible air in the line)

It was decided at the meeting that Rand Water would manufacture a 300 – 350 mm fast acting (butterfly or ball) valve that could be installed on the scour outlet. The intention was to use this valve to generate a negative wave that would travel through the system. At a number of locations the pressure disturbance would be recorded. These measured pressures would then be compared with calculated data to determine the differences. Based on this, the intention was to identify the factors that could have contributed to the difference between the calculated and measured results.

Inverse transient pressure tests could be conducted on the B16 pipeline (Section 4) to obtain field verification for the experimental work conducted. Water could be released from the pipeline to create an air bubble of known size in the pipeline, prior to the inverse pressure testing.

The intention was to determine the influence of air on the wave celerity in the line as well as to try and determine the exact location of the leaks by analysing the wave pattern obtained from the tests. At the beginning of the test section, the 1500 mm butterfly isolating valve and all the bypasses would be closed while an end cap isolated the system near Mapleton Pump station.

Whenever the “process” of capturing and releasing the air is ineffective, free air will be present in the pipeline. This experiment was used to determine if free air was available and if so, the extent of the free air body in the system.

The scour valve was opened quickly through the addition of a fast acting knife valve installed on the flanged section of the scour outlet to create a pressure drop in the line. The wave movement was traced by capturing data at three locations along the test section. One location was upstream and another downstream from the scour valve.

Figure 3.3.7 below contains a schematic section of the pipeline. The various test procedures are described by referring to the schematic layout.

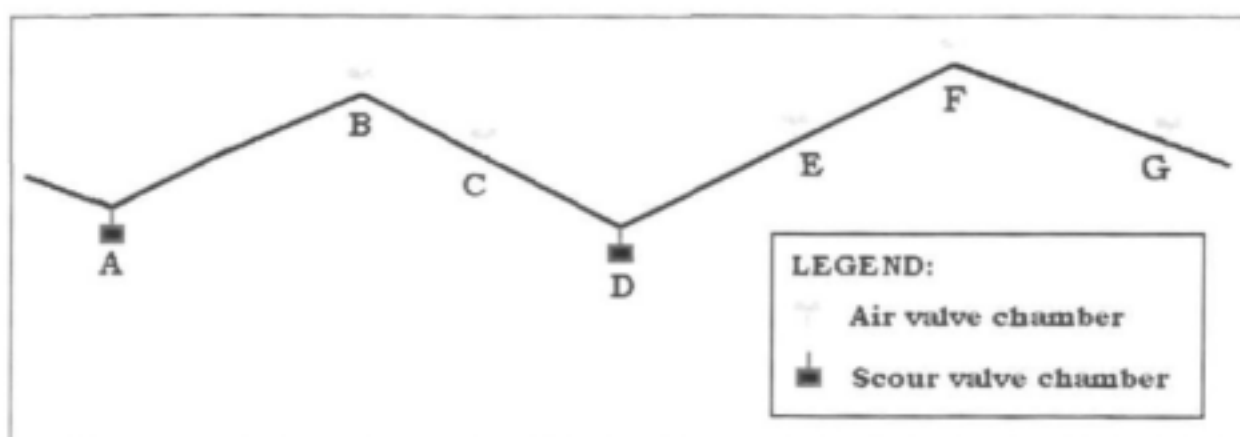


Figure 3.3.7: Schematic layout of the test section (B8 or B16)

Test procedure (Location of air in the pipeline)

- Select critical sections for the test (ineffective accumulation capacity and steep downward sections)
- Allow the system to run, until a stable flow rate has been established
- Capture the pressure fluctuation at valves B, C and G. The pressure fluctuation is due to the negative pressure wave created at A or D.
- Compare the results with those obtained with the theoretical calculation of the system
- To zoom in on the location of the air, various tests should be conducted by moving the pressure gauging closer to the expected location of the air bubble.

In the test on the B16 pipeline all the air valves were isolated except the air valve at chainage 28 360 m, where air was introduced to the test section. The intake and delivery side of the pump truck was used to release water from the pipeline. By allowing a predetermined volume of water (that can be measured accurately in the pump truck) from the line an equal volume of air will be available in the pipeline. The release of the volume of water will be from the B16 to the B8 pipeline. Swapping the inlet and delivery sides of the pump truck will provide the ability to pressurize the system prior to the generation of the pressure wave through the opening of the fast acting scour valve.

The maximum pump capacity of the pump truck is however 4 l/s. This would require a long time (Table 3.3.2) to do the tests and hence the drainage will have to be through the scour valve. These test were intended to be undertaken during March 2002, but due to the unavailability of the pump truck the tests had to be postponed to October 2003, which is unfortunately outside the time scope of this project.

Table 3.3.2: Details of the pipe section to be tested and the determination of the time it will take to pressurize the pipeline through the pump truck

B16 Inverse transient test					
The highest point is at chainage 28 630 m					
Upstream slope =	0.2498	%			
Downstream slope =	0.7959	%			
Water level to crown, h =		m			
Test Pressure, Pt =	4	Bar			
Pipe segment length =	2016	m			
Volume of pipe section =	9120.17	m ³			
H (m)	Area apex	Volume air at test pressure, Pt (m ³)	Volume of air at atmospheric pressure, Pa (m ³)	Time to Pressurize the line (h)	Volume air/Volume of pipe (%)
0.05	0.023	0.302	1.509	0.105	0.02
0.10	0.064	1.696	8.481	0.589	0.09
0.15	0.118	4.644	23.220	1.612	0.25
0.20	0.180	9.471	47.353	3.288	0.52
0.25	0.250	16.435	82.173	5.706	0.90
0.30	0.326	25.750	128.750	8.941	1.41

The findings of these tests will be incorporated in a technical article.

Table 3.3.3 reflects the equipment used for the pressure measurements.

Table 3.3.3: Positions for the installation of the equipment for the tests (B16)

Test number Section 4 Chainage (m)	Measuring device	Pressure devices	
		50 Bar RW transducer	UP transducer (Bar)
Pump truck (29 424)	Mass flow, Pressure	Yes	None
Measurement C (29 250)	Pressure	Yes	25 Bar
Measurement B (28 360)	Pressure	Yes	16 BarA
Measurement C (21 217)	Pressure	Yes	16 BarA
Isolating butterfly valve	None	None	None
Fast-acting valve	Mass Flow, RW	None	None

The importance of data sets with a common time scale is crucial. Past experience indicated that the time drift of computers/laptops makes it impossible to try and obtain data with a common time base.

A special trigger was developed to start all the computers at the same time. **Table 3.3.4** reflects the equipment used for the tests on the B16 pipeline. The final version of these triggers will also be used in the final tests.

Table 3.3.4: Equipment for the inverse transient test on the B16 pipeline

Equipment	Number required	Available from RW	Available from UP
Longitudinal detail of B16		RW	
Pressure transducers (Output 4-20 mA)	3* 40 Bar 3*20-25 Bar 3*10-20 Bar		2 * 60 Bar 1*40 Bar 1* 25 Bar 2* 10 Bar
Power box for transducers	3		3
Laptops	3	M. Webb	2
Power box for laptops	3		3
PCMCIA cards and software	3		3
Cabling for transducers			Available
Shade and rain protection for data stations	3		1
Pump truck to pressurize pipeline		Provided by RW	
Communication between gauging points. Transport on site, with drivers that know the area	3	3 - Provided by RW	
Provision of guards		Provided by RW	
Fast acting valve to be installed to release flow	1	RW	

The useful positions where the fast acting valve can be installed onto the scour valve outlets are shown in **Table 3.3.5**.

Table 3.3.5: Positions for the installation of the fast acting valve (B8 pipeline)

Scour position	Pressure gauging measuring points		
	M1	M2	M3
Section 1			
2 436	2 003	2 609	3 190
3 655	3 190	3 958	5 150
5 331	5 150	5 525	6 798
Section 4			
45 730	46 296	46 786	47 690
47 437	46 296	46 786	47 690
<i>Other points can be identified</i>			

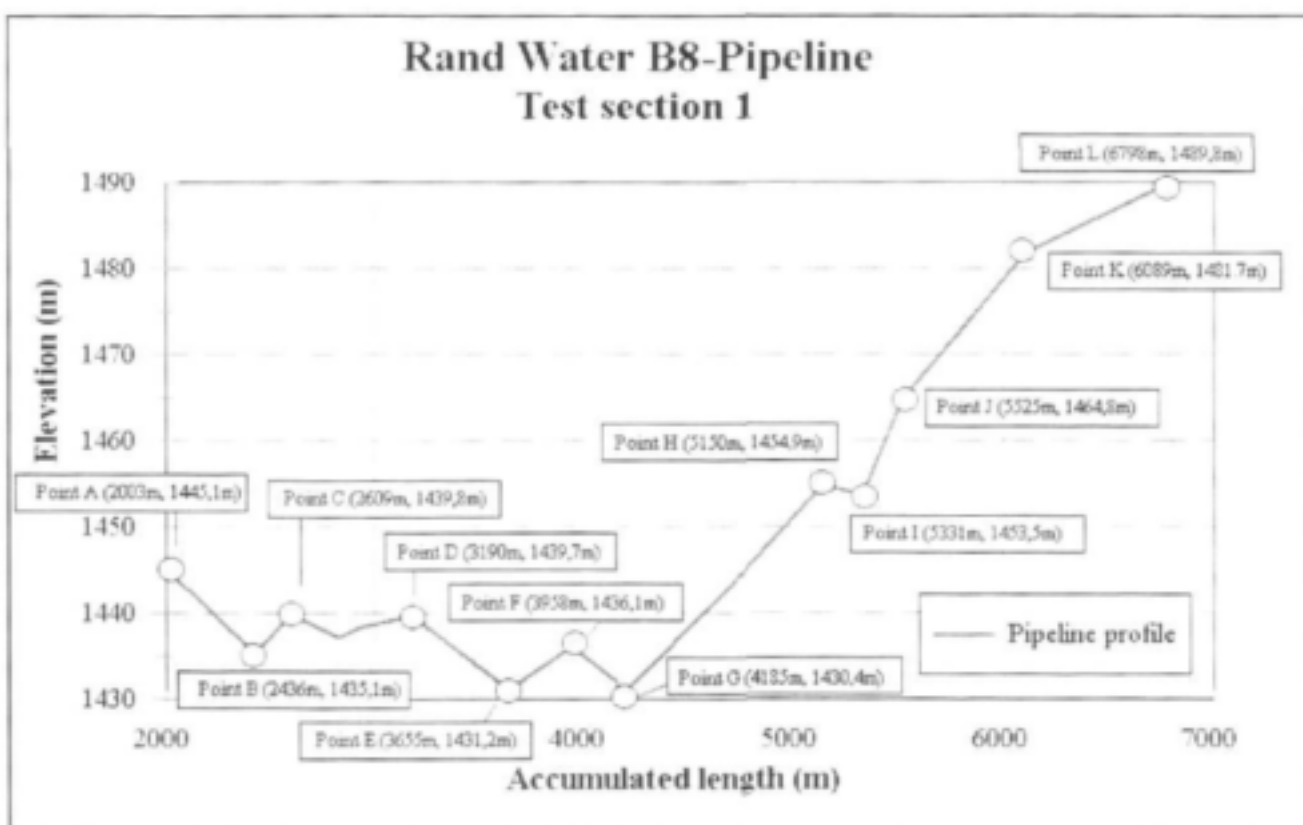


Figure 3.3.8: Test section 1 (B8 pipeline)

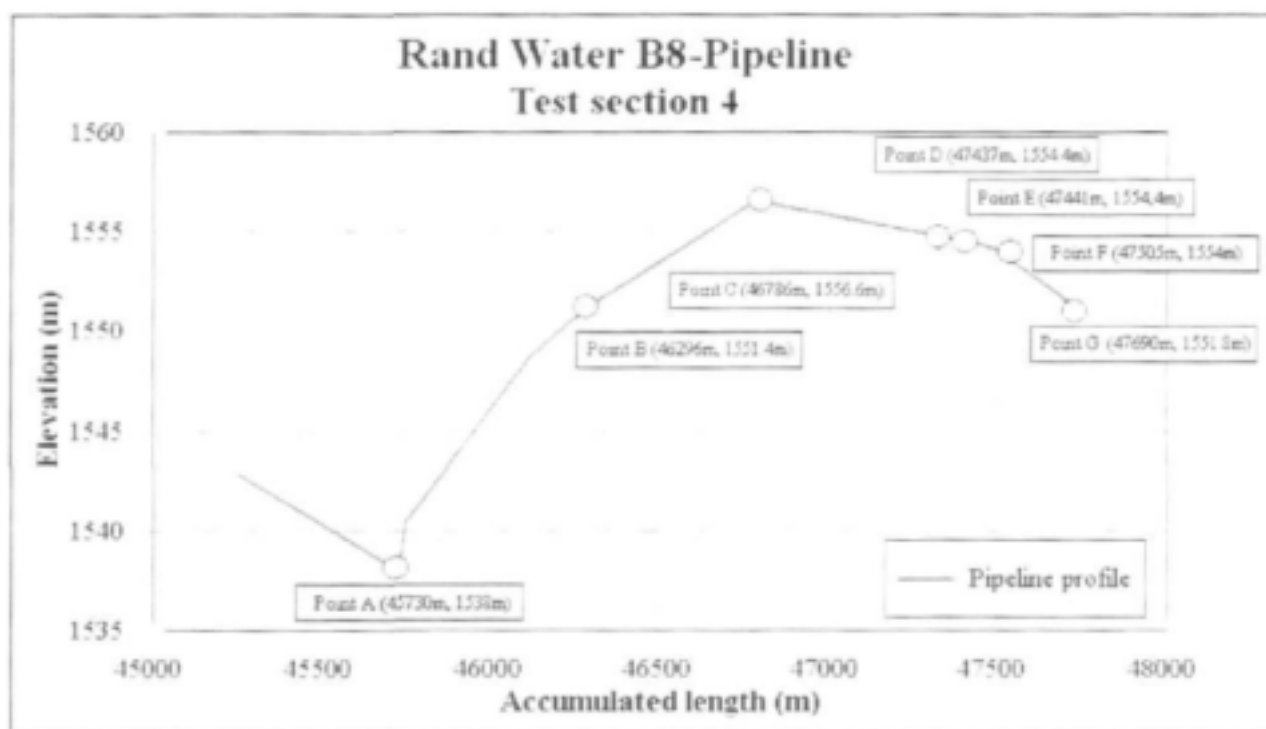


Figure 3.3.9: Test section 4 (B8 pipeline)

3.3.3.2 Hydraulic transportability of air

The design flow rate of the B8 pipeline is approximately 6.94 m³/s. At the maximum flow rate the velocity is 2.05 m/s, which is sufficient to transport the air down along almost the entire pipeline based on the formulae developed from the experimental work that was conducted. As mentioned before this design flow rate is not obtained. **Table 3.3.6** is a summary of maximum negative slopes where air will still be hydraulically transported versus velocities.

Table 3.3.6: Maximum negative slopes versus velocities

Flow rate (ML/day)	Velocity (m/s)	Maximum angle (degrees)
100	0.33	0.065
200	0.66	0.380
300	1.00	1.050
400	1.34	2.150
500	1.67	3.750
600	2.05	5.850

The B8 pipeline was analysed to determine which sections would not be able to transport the air hydraulically. The results of this analysis are reflected in **Table 3.3.7**.

Table 3.3.7: Sections not capable of transporting air hydraulically

Flow rate (ML/day)	Velocity (m/s)	Number of sections where air will not be transported	Length of pipe where air will not be transported (m)
100	0.33	62	15 839
200	0.66	45	11 562
300	1.00	20	3 908
400	1.34	5	306
500	1.67	2	18
600	2.05	2	18

This analysis does not take into account whether or not the air valves are functioning or not. This creates even more additional stationary air pockets along the pipeline.

3.3.3.3 Interception of air at the air valves

The discontinuity on the crown of the pipe to intercept the air bubbles is very small compared to the pipe diameter ($d/D = 0.05$ for small orifice air valves and $d/D = 0.09$ for combination air valves). The air bubbles do not always travel in the center of the crown and therefore it can be assumed that a lot of air is not intercepted. This air will then be trapped further down the line at a point where it cannot be transported hydraulically any further.

3.3.4 Conclusion

In March 2002, pressure tests were conducted on the B16 pipeline. The intention was to also conduct inverse transient tests on the line with the objective to determine the influence of air on the wave celerity and leaks on the wave movement in the line. Section 4, similar to **Figure 3.3.9**, of the said pipeline (chainage 45 730 m to 47 690 m) was used for the tests. At the beginning of the tests, section the 1 500 mm butterfly isolating valve and all the bypasses were closed while an end cap at chainage 47 689 m isolated the system near Mapleton Pump station.

Some difficulties were experienced with regard to the pressure tapping on the drain valve of the air valve points, the flow measurement of the release at the drain valve and the triggering box that was developed to trigger the different data acquisition systems.

All indications are that air has an effect on the capacity of the B8 pipeline. The importance of the pipeline has unfortunately meant that no tests could be conducted on it before the B16 pipeline was complete. This has unfortunately resulted in delays, which forced the project team past the time scope for this project. The tests will however still be conducted and a technical document of this will be released.

3.4 Investigation into the influence of air on raw water rising main of the Buffalo River pumping station (East London)

3.4.1 Introduction

The Buffalo River Raw Water Rising Main transports water from a weir downstream of the Bridledrift Dam to the balancing dam at the purification works. After a new pumping line was installed, several problems were experienced. The system was operating below the designed flow rate and bubbly flow that was gravitated from the surge tanks resulted in blowbacks into the vent pipes and the surge tower, creating spillages.

According to Messrs Fongoqa Skade Toyi & Associates cc (FST), "it appears that the pump at the pump station is pumping air into the new 600 mm ductile iron pipeline. A lot of this air is being vented at the first air valve (chainage 240m). No air is being vented from air vents 2 & 3 (chainage 647.8 and 938.0 m), but water entering the surge chamber (chainage 1 035 m) is fizzing (effervescent) due to the amount of air in the pipeline".

Since the air valve layout and size selection were done in accordance with the best knowledge available, it was decided that this problem would provide valuable information to the research project on air in pipelines.

The purpose of the inclusion of this investigation was to obtain information about the following aspects:

- Solubility and release of air from solution under varying pressures.
- The identification when blow-back occurs.

3.4.2 Description of the system

During a visit to the site in February 2002, water was pumped through the two 600 mm pumping lines towards the surge chambers (**Figure 3.4.1**) operating pumps 1 and 4.

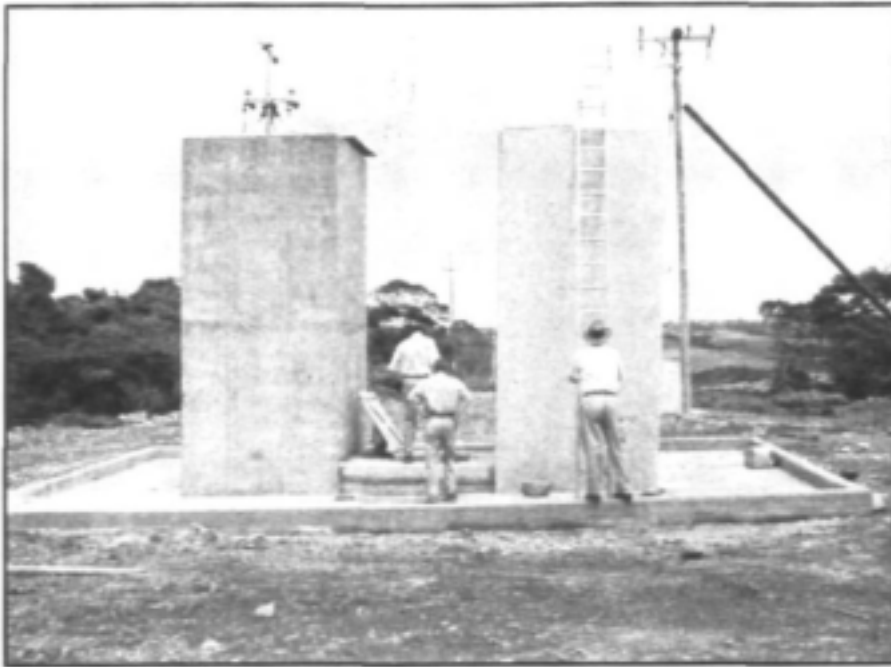


Figure 3.4.1: Surge towers

Not only was the water in the surge chamber fizzing (**Figure 3.4.2**), but large volumes of air also blew back through the 900 mm pipeline, linking the surge chamber with the collection chamber.



Figure 3.4.2: Fizzing water in the surge tower

Vent pipes were provided on the gravity section. The vent pipes are 80 mm in diameter and were coupled to a stub Tee of 600 mm. Water and air spurts out of these 3.5 m long vent pipes. It was evident that air intruded into the system from the suction side of the pumps. The free air then dissolved due to the increased pressure at the pumps (± 80 m). This dissolved air was then released from solution when the temperature increased or the residual pressure dropped.

3.4.3 Data acquisition

The field observations will be discussed in the following order: reflecting the flow of water from the weir in the Buffalo River through the pump station, up to the surge tower from where it gravitated to the collection chamber upstream of the culvert. Temperature, dissolved oxygen concentration and conductivity were recorded at a number of locations.

3.4.3.1 Weir

From the weir (**Figure 3.4.3**) about 1.9 km upstream from the pump station, water was fed into two 900 mm diameter AC pipelines which fed the suction manifold of the pump station. No evidence of vortex formation at either of the intakes was visible.

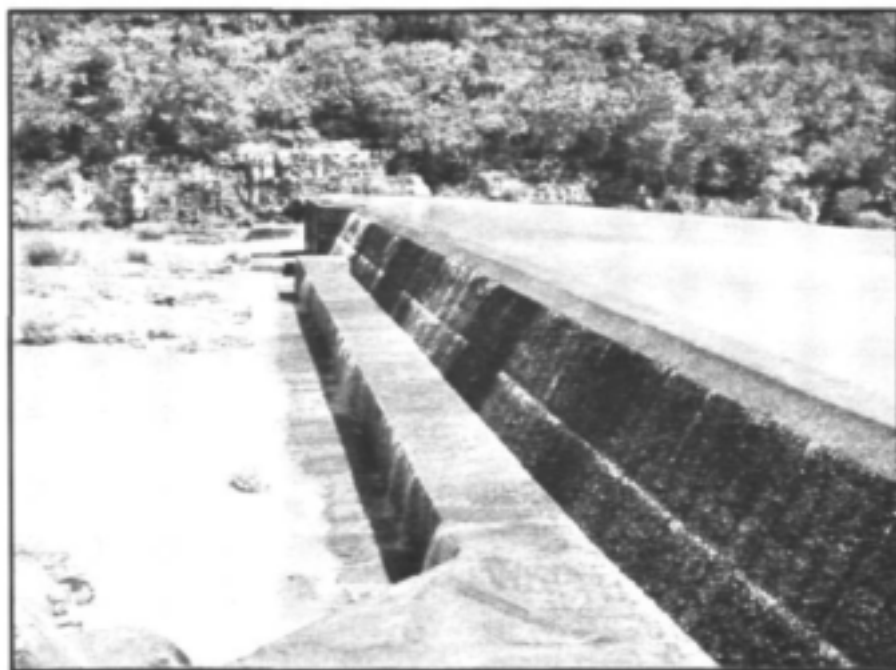


Figure 3.4.3: Weir from where the water entered the gravity line to the pump station

3.4.3.2 Gravity pipeline to the pumps

The only access point along the two 900 AC gravity pipelines was the position where an air valve was installed on both lines, approximately 630 m upstream from the pump station. At this point, the pressure was still positive, although it was easy to press down the large ball of the air valve. The pressure at this point was estimated at about 1.5 to 2 meters.

The gravity line was linked to a surge tank (RWST) that protected the gravity line against induced pressures when the pumps tripped. Unfortunately, it was not possible to record the water level in the surge tank (RWST).

3.4.3.3 Pump station

Four pumps were installed. Pumps 1 and 2 conveyed water through the two 600 mm pipelines to the old and new surge towers (**Figure 3.4.1**). The gauge reflecting the pressure in the suction line (**Figure 3.4.4**) indicated that when pumps 1 and 2 were running, the pressure was sub-atmospheric. This pressure reading corresponded to calculated values that also reflected negative pressures on the suction side of the pump.



Figure 3.4.4: Negative pressure on the suction side of the pumps

A number of pump trips and start-ups were initiated, with the objective of determining the extent of pressure fluctuations at the pumps. Comparing these with the calculated pressures would reflect the influence of air on the surge pressures.

3.4.3.4 Pump line

At the three air valve positions along the line, water samples were taken and the dissolved oxygen concentration was measured.

3.4.3.5 Surge tower and gravity pipeline

At the surge towers (**Figure 3.4.1**) the water was fizzing (**Figure 3.4.2**) and intermittently large air bubbles were entering the surge tower with a rumble from the downstream (gravity pipeline) side. At the vent pipes, evidence was found of the occasional spilling of water from the pipe (**Figure 3.4.5**). The spilling was initiated by large air pockets that move upstream (blow-back) and was partially intercepted and expelled through the vent pipes.



Figure 3.4.5: Standpipe with indication of spillage

Work to increase the size of the vent pipe was underway on the older of the two gravity pipelines. The increase of the vent size would exacerbate the situation, due to the fact that when blow-back does occur, more water would expel from the vent pipes.

3.4.3.6 Collection chamber

The two 900 mm gravity lines gravitating from the surge tower and the old pump line discharged into the collection chamber. Water flows from the collection chamber into the Umzoniana Tunnel. In the collection chamber, evidence was visible of air released from the gravity flow.

3.4.4 Results

3.4.4.1 Oxygen concentrations

Table 3.4.1 reflects the results of measurements taken with an YSI Model 85 handheld probe calibrated for an evaluation of 168 m above msl.

Table 3.4.1: Results of sample analysis

Location of sample	Temperature °C	Oxygen concentration		Conductivity µs/m
		%	mg/l	
Weir (Reading 1)	21.0	59.2	5.95	420
Weir (Reading 2)	20.8	58.7	4.83	417
Air valve on gravity line (680 m upstream of the pump station)	22.0	63.4	5.42	432
Downstream of pump	21.0	115	10.80	472
Air valve 1	20.6	82.7	#	#
Air valve 2	22.0	79.4	9.70	#
Air valve 3	22.3	72.8	6.71	#
Surge tank	20.3	124	11.5	482
Pretoria tap water	24.0	25.7	2.14	302
Boiled Pretoria Water	20.4	14.7	1.31	155

Note: # Results not available

The measured results indicated that:

- The water at the weir had a high concentration of oxygen and that it increased substantially to a maximum value just downstream of the pumps. This indicated that somewhere along the gravity section, air intruded into the system.
- With the increase of pressure at the pumps, most of the free air was dissolved and transported towards the surge chamber.
- At air valve 1 (chainage 240 m), occasional release of air occurred. This free air escaped from the air valve due to the following reasons:
 - all the free air was not dissolved at the pumpstation or
 - air was set free from the over-saturated solution due to the drop in pressure between the pumpstation and air valve location 1 (elevation difference of about 14 m).
- Virtually no air was released at air valve location 2 (chainage 647.8 m), since very little air is released from the liquid although it is saturated.
- A similar situation is experienced at air valve location 3 (chainage 938 m) as for the air valve at location 2. The residual pressure at location 3 is about 18.0 m.
- A large portion of the air from the over-saturated liquid was released when the pressure dropped to atmospheric pressure in the surge tower. The process took time and since the retention time in the surge tank was short (about 9 seconds), the rest of the air was released in the gravity line.
- The bubbles grew in the gravity line until they became unstable and blew back through the line towards the surge tower.

3.4.4.2 Ability to de-aerate effectively

The investigation of this problem indicated that the release of air from a solution can take up to 9.6 minutes under atmospheric pressure and an ambient temperature of 21 degrees.

3.4.4.3 Capacity to transport the air

The investigation of this problem indicated that a flow velocity of 0.764 m/s is insufficient to transport large air bubbles hydraulically along a slope of 2.8%, although small air bubbles are transported at this flow velocity.

3.4.4.4 Influence of air on the propagation of the pressure waves

A surge analysis was undertaken and a pump trip was modelled. These results indicated that a pressure drop of about 110 m could be experienced if pump 1 tripped. During the field trip, pump 1 was switched on and tripped. The measured pressure drop of about 31 m was far less (about 25%) than expected for a flow velocity of 1.7 m/s. According to Jouwkowsky (Chadwick & Morfett, 1994) it is known that:

$$\Delta h = \frac{cV}{g} \quad \dots\dots\dots (3.1)$$

Where:

- Δh = Change in pressure (m)
- c = Wave celerity (m/s)
- V = Average velocity (m/s)
- g = Gravitational acceleration (m/s²)

This suggests that a pressure drop of 120 m can be expected. However, **Figure 3.4.6** reflected a pressure drop of only approximately 31 m. It is known that the celerity of an air-water mixture is much less than that of water, **Figure 3.4.7** shows this relationship. It is therefore clear from this that abundant air is present in the pipeline.

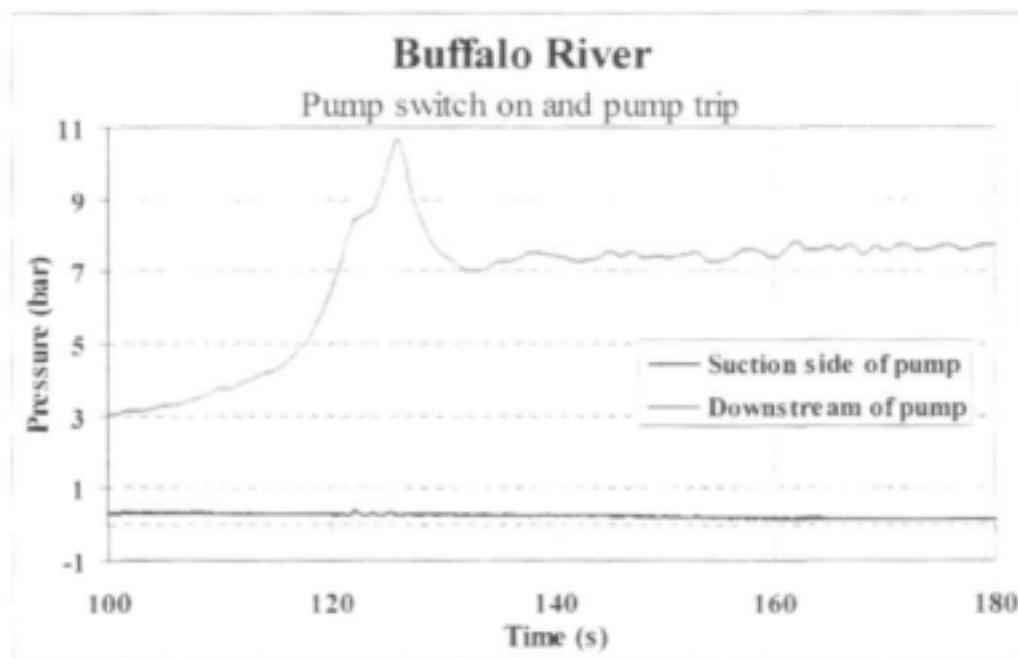


Figure 3.4.6: Measured pressure drop in the system

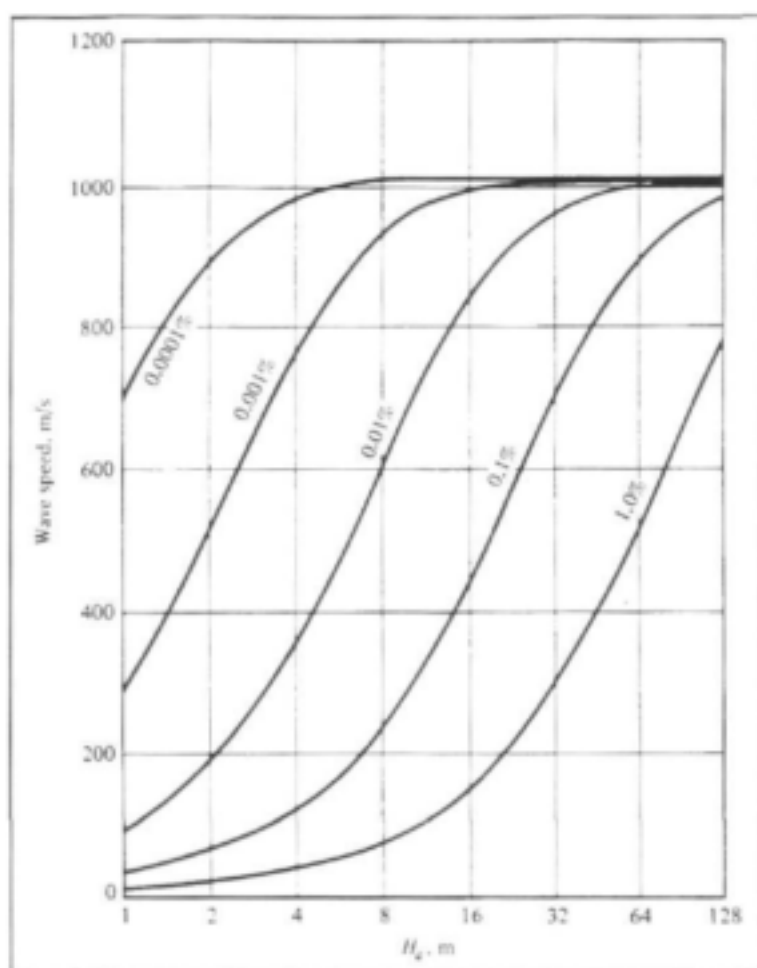


Figure 3.4.7: Reduction in wave celerity due to free air

The investigation of this problem indicates that the presence of air reduced the surge (negative) pressure by as much as 75 m.

3.4.5 Conclusion

From the field investigation it was apparent that excessive quantities of air were present in solution. This air was released from solution as soon as the pressure in the system was reduced, i.e. when reaching the surge tower. The free air, still in the form of small bubbles, was swept down the gravity line, started accumulating to form large air pockets, which could not be transported hydraulically down the line. The air pockets moved upstream, causing blow-back into the surge towers. The source of the excessive air volumes in this system was found on the suction side of the pumps where air was drawn in through air valves on the feed line to the pumps.

4. EXPERIMENTAL WORK

4.1 Introduction

The aims of the research included the conducting of experimental work to establish to influence of air in pipelines. The shortcomings and conflicts with regard to the understanding of the influence of air in pipelines and the lack of clear directives on where air release valves are required on a pipeline, necessitated the experimental evaluation of the following aspects:

- Hydraulic transport of air in pipelines.
- Quantify the discontinuity required in a pipeline to effectively capture the free air that is hydraulically transported.
- Use inverse transients to detect leaks and air bubbles in pipelines.
- Determine the energy loss associated with localised air bubble.

In Sections 4.2 to 4.5 the experimental work that was undertaken to obtain knowledge on these aspects is discussed in detail.

4.2 Hydraulic transport of air in pipelines

4.2.1 Introduction

Entrapped air in pipelines adversely affects the economy of operation as the effectiveness of the system to convey water is reduced. Various methods can be employed to prevent air from entering a pipe system and to remove air from the system by mechanical and hydraulic means. Kalinske & Bliss (1943) as well as Wisner, Mohsen & Kouwen (1975) suggest empirical formulae describing the hydraulic removal of air. The formulae were derived from the results obtained in tests performed on inclined pipes transporting varied sizes of air bubbles in the flow. The proposed formulae yield different values for the critical velocity at which the hydraulic transport of air takes place. For a 500 mm diameter pipe, inclined at 10° to the horizontal, the empirical formulae yield the following critical flow velocities.

According to Kalinske & Bliss (1943):

$$\frac{Q_c^2}{gD^5} = 0.707 \tan \phi \dots\dots\dots (4.2.1)$$

Where:

- Q_c = Minimum discharge for the transportation of air in the flow (m^3/s)
- g = Gravitational acceleration (m/s^2)
- D = Internal pipe diameter (m)
- ϕ = Inclination of pipeline to the horizontal i.e. slope (degrees)

Thus:

$$Q_c = \sqrt{(9.81)(0.5)^5 (0.707) \tan 10^\circ} = 0.196 \text{ m}^3/s$$

$$V_c = \frac{Q_c}{A} = \frac{(0.196)(4)}{\pi(0.5)^2} = 1.0 \text{ m/s}$$

The minimum velocity at which air will be transported in the flow, according to Kalinske & Bliss (1943) is $V_c=1.0 \text{ m/s}$

According to Wisner, Mohsen & Kouwen (1975):

$$V_c = \sqrt{gD}(0.25\sqrt{\sin\phi} + 0.825) \dots\dots\dots (4.2.2)$$

Where:

- V_c = Minimum flow velocity for the transportation of air in the flow (m/s)
- g = Gravitational acceleration (m/s^2)
- D = Internal pipe diameter (m)
- ϕ = Inclination of pipeline to the horizontal (degrees)

Thus:

$$V_c = \sqrt{(9.81)(0.5)}(0.25\sqrt{\sin 10^\circ} + 0.825) = 2.1 \text{ m/s}$$

The minimum velocity at which air will be transported in the flow, according to Wisner, Mohsen & Kouwen (1975) is $V_c = 2.1 \text{ m/s}$. This value is double that obtained in the Kalinske & Bliss (1943) expression.

4.2.2 Purpose of the experiment

The objective of the experimental work was to establish the conditions for hydraulic transport of air bubbles of varying sizes. The relationship governing flow velocity, pipe geometry and pipe inclination is established for the effective hydraulic removal of air from the system. Aspects of mechanical removal of air and the problems attributed to the presence of air in pipelines have been discussed in this report.

The findings of this experimental work were compared to previous research findings of Kalinske and Bliss (1943) and Wisner, Mohsen & Kouwen (1975).

4.2.3 Experimental set-up

In order to study the movement of air in a large diameter pipeline, an experimental set-up was constructed in which the movement of air in the pipeline could be observed and the volume of air in the pipeline controlled. The experimental set-up was constructed at the hydraulic laboratories of the Department of Water Affairs in Pretoria West.

4.2.3.1 First experimental set-up

A pipe with an internal diameter of 110 mm was used in the experimental set-up, as it is a commonly used pipe size. A transparent pipe was used so that the bubbles in the pipeline could be observed. The induced air bubble volume was controlled by an Air Induction Box (AIB), which was specifically assembled for the experimental trials. A constant head tank supplied the flow. The supply line was equipped with accurate flow meters.

A schematic layout of the experimental set-up is shown in plan in **Figure 4.2.1**.

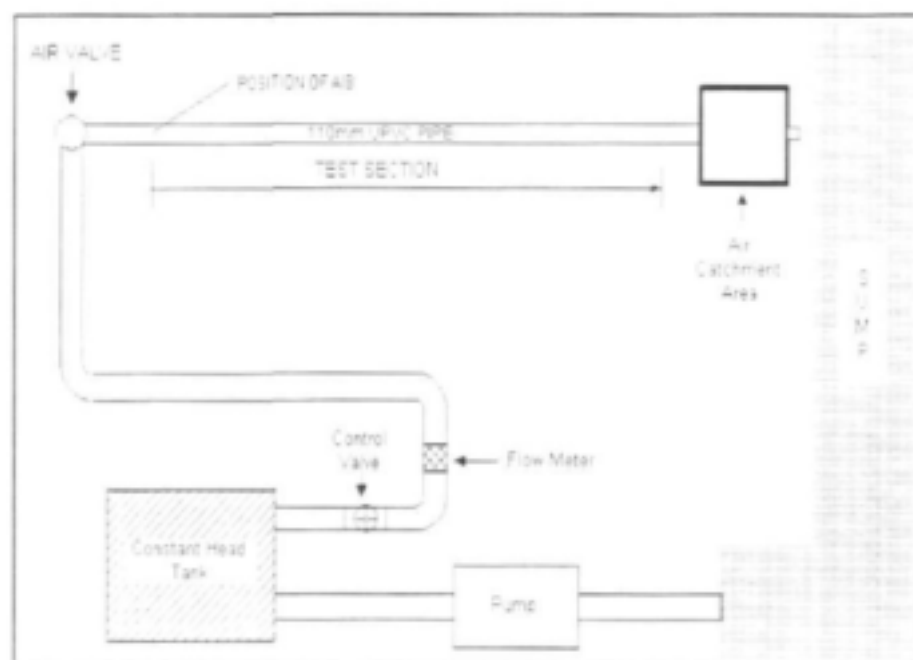


Figure 4.2.1: Schematic layout of experimental set-up shown in plan view

The couplings between pipe segments were constructed by using ordinary uPVC couplings. The plain plastic ends were glued with "PVC Weld" and sealed with silicon. **Figure 4.2.2** shows the two types of plastic couplings used in the set-up. The coupling on the left consisted of a rubber seal (left side) and a glued side (right side), while the coupling on the right had two glued sides. The reason for using couplings with a rubber seal was the ease of reassembly of the pipe system on change of pipe inclination.

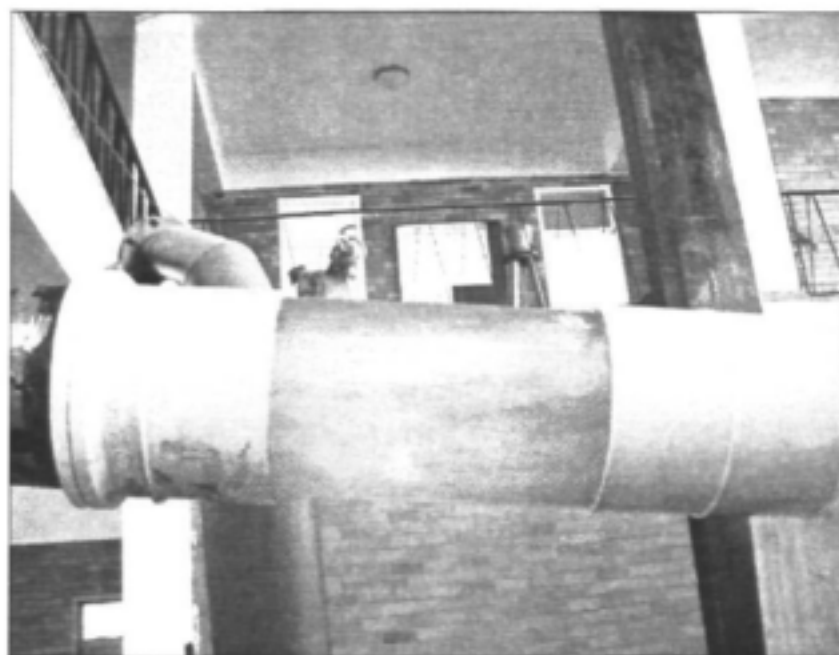


Figure 4.2.2: The two types of couplings used in the set-up

The clear transparent 110 mm Ø uPVC pipe was connected to a section of 90 mm Ø ribbed hose on both ends.

The inflow end had to be flexible to accommodate the changes in height with varied inclination of the uPVC test section. The ribbed hose was stiff and difficult to work with. Lay-flat hose was used as an alternative to the ribbed hose. A coupling with a rubber O-ring was forced over the 110 mm Ø side of a 110 mm – 100 mm steel reducer. A length of lay-flat hose with 100 mm Ø was connected to the 100 mm end of the reducer and to the 100 mm pipe supplying the water. Pipe clamps were used to secure the lay-flat hose to both ends (**Figure 4.2.3**).

To simulate bubbles in pipelines at a variety of gradients, the set-up had to be adjustable to different gradients with relative ease. The height of the set-up on the inflow side was limited to just over 3 m, which is the distance to the ceiling of the roof. Hence the pipe could only be lifted to an angle of 15°, measured to the horizontal. In practice, the gradients of pipelines seldom exceed 0.268 m/m (15°). The investigation was conducted for angles from 0° to 15° in increments of 2.5°.



Figure 4.2.3: The length of lay-flat hose

The horizontal spacing of the vertical pipe supports was fixed and the required heights of the horizontal elements of the supports for each experimental gradient was calculated. A schematic diagram of the pipe angles and the vertical supports is shown in **Figure 4.2.4**.

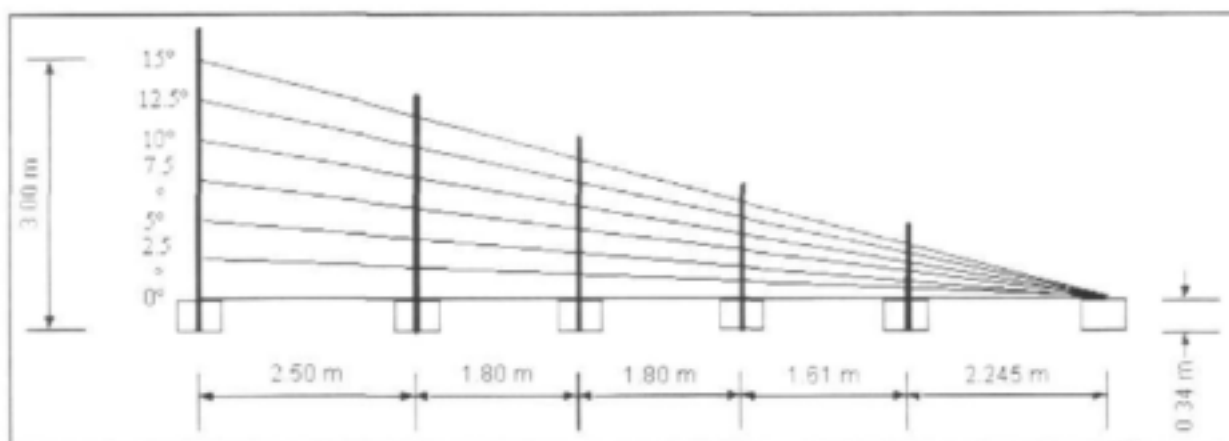


Figure 4.2.4: Schematic layout of pipe supports (elevation)

The vertical elements of the supports were fixed to the concrete pillars using 8 mm $\varnothing$ holding-down-bolts. Foot-plates were also welded to the bottom of the vertical elements and these in turn bolted to the floor, to provide extra stability.

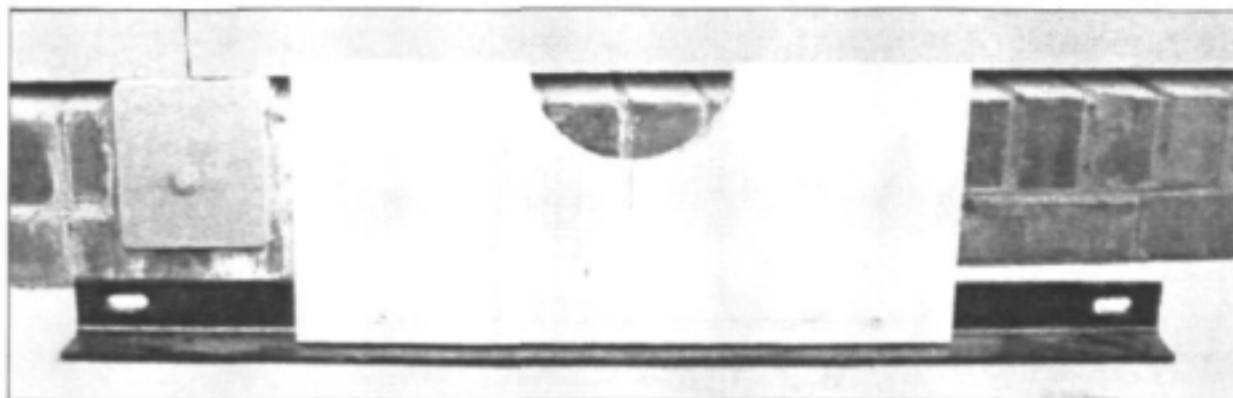


Figure 4.2.5: The horizontal element of the pipe support

The pipe was held in place by a wooden support (**Figure 4.2.5**). The support was fixed to the angle using 5 mm Ø x 30 mm long bolts. The distance from the underside of the pipe to the bottom of the wooden support measured 100 mm.

It is necessary to have control over the volume of air induced into the pipeline. A device controls the airflow. "Productive Solutions", a Pretoria based company, provided an electronically controlled air regulator to regulate the induction of air into the system. The Air Induction Box (AIB) is shown in **Figure 4.2.6**.

A solenoid valve controls the flow of air through the regulator, with rapid aperture control to allow for small volumes of air to pass. The system is also fitted with a pressure regulator, which can be adjusted manually, to set the pressure as, required.

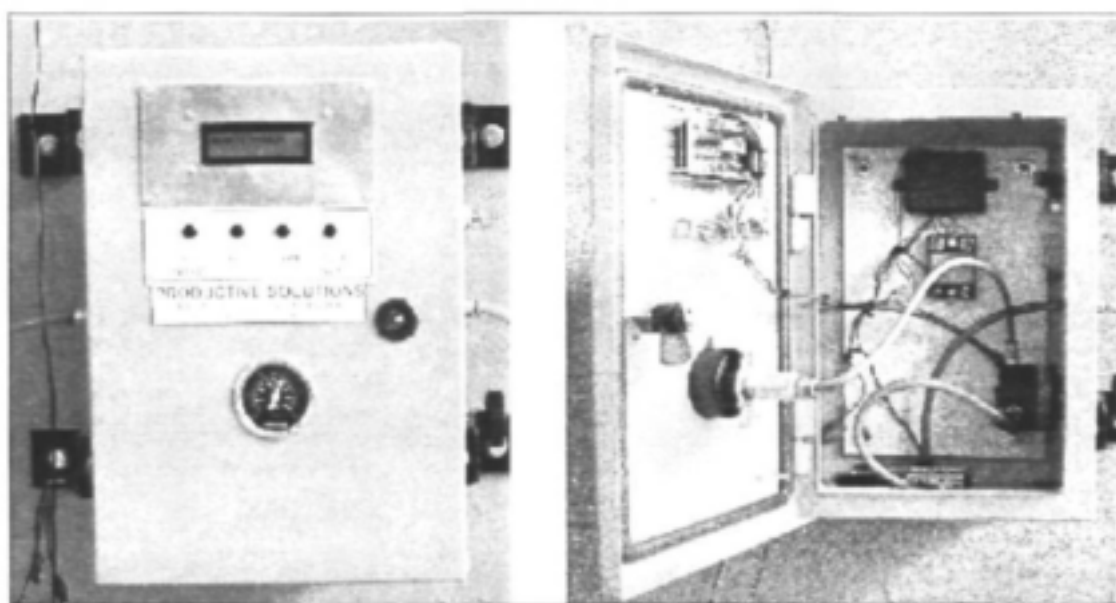


Figure: 4.2.6: The Air Induction Box (AIB). Assembly shown

Compressed air at a pressure of 8.75 bar was supplied from a compressor as shown in **Figure 4.2.7**. The compressor was connected to the regulator of the AIB. To protect the instrumentation from excess pressures and from vapour attack, the compressed air passed through a 0.03 μm air filter installed upstream of the solenoid valve.



Figure 4.2.7: The compressor used for supplying air

The following four parameters can be set on the AIB, to control the volume of air and airflow rate into the pipeline.

- "Bubble" size
- "Gap" size
- "Number" of bubbles and
- The supplied "Pressure" to the solenoid valve.

The "*Bubble*" on the display of the AIB indicates the bubble size in milliseconds (ms). This is the duration for which the solenoid valve will open release air. In this way, the bubble size is controlled. By altering the duration, the volume of air passing through the valve is controlled.

The "*Gap*" on the display of the AIB reflects the time in milliseconds (ms) between consecutive bubbles. The time is taken as the duration that the valve is closed, thus between valve closure and valve re-opening.

The “*Number*” on the display of the AIB indicates the amount of consecutive bubbles to be induced into the pipe system.

The “*Pressure*” on the display of the AIB indicates the pressure at which the air is induced into the system. The pressure can be varied manually by adjusting the pressure-control valve of the air regulator.

A brass fitting was used to connect the air hose passing between the AIB and the uPVC pipe. The brass coupling is shown fitted to the uPVC pipeline in **Figure 4.2.8**.

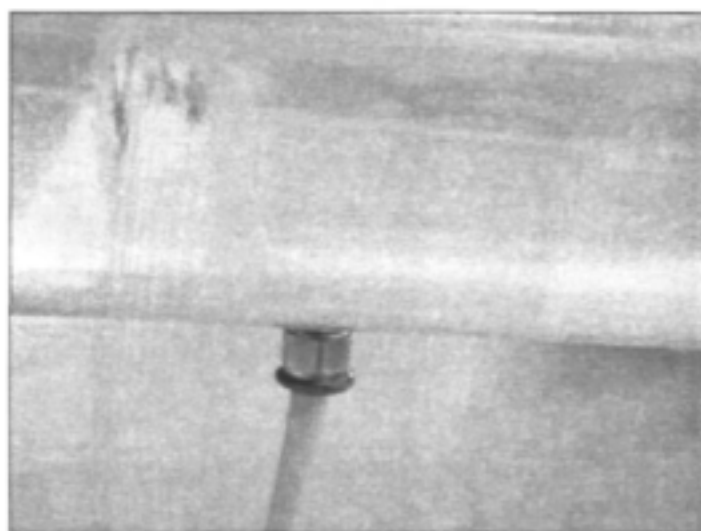


Figure 4.2.8: The brass coupling for air induction

The coupling allows for the removal of the air pipe by pulling at the blue lip of the fitting, which in turn releases the hold of the little brass teeth. This fitting worked well. It was airtight and was easy to attach / remove the air pipe. A non-return valve in the air pipe upstream of the brass fitting prevented the water, from the test pipe, from flowing towards the AIB.

In order to calibrate the volume of induced air, the size of the required bubble had to be determined. Three bubbles of varying sizes were investigated to obtain information on the movement and behaviour of entrapped air in pipelines. The size of the bubbles was determined visually and by changing the air regulator settings until a small, medium and large bubble were generated.

To calibrate the air induction, the volume of each bubble had to be determined. A concrete air trap was built around the outlet of the pipeline to capture the air exiting the system.

A Perspex measuring container shown in **Figure 4.2.9** was constructed and used to capture the air. A valve was fitted to the top of the container so that the container could be submerged and all the air could be removed through the valve. Once all the air had been removed, the valve was closed and air was pumped into the system with the AIB set up as desired. As air exits the system, water is driven from the measuring container.

The volume of the container was determined and the container marked in regular increments so that the volume could be read off. Once all the air had been captured, the container was lifted or lowered until the water level inside and outside the container were the same. This ensured that the pressure inside the container was equal to atmospheric pressure. The volume of captured air was then determined, visually using the markings on the sides of the container.

The measuring error was minimised by inducing as many bubbles as possible into the container. Once all the bubbles had been captured in the Perspex air-measuring container, the total volume of air captured was divided by the number of bubbles injected into the system. This provided the average volume of each injected bubble. The more bubbles injected, the smaller the measuring error would be.

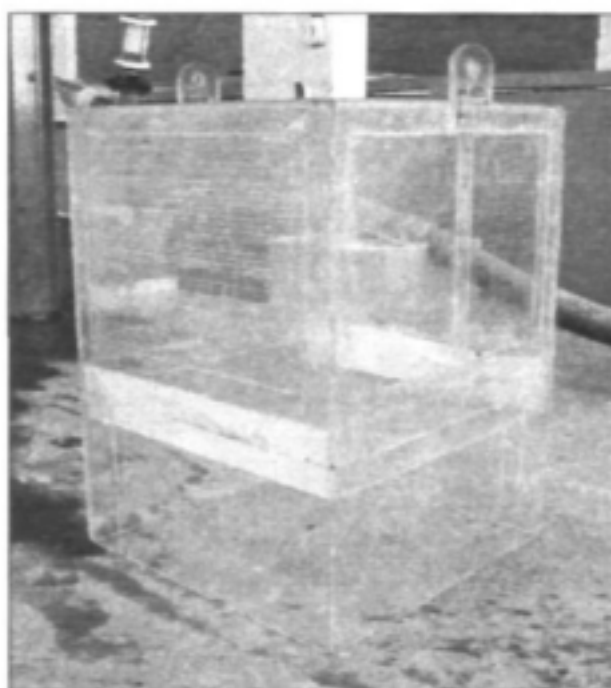


Figure 4.2.9: The Perspex air measuring container

The determined volumes and air regulator settings for each size of bubble are given in **Table 4.2.1**.

Table 4.2.1: Regulator settings and bubble volumes

Bubble Size	Air Regulator Settings		Bubble Volume Millilitres (ml) (Atmospheric Pressure)
	Bubble Milliseconds (ms)	Pressure (bar)	
Small	25	3	25
Medium	75	6	75
Large	80	6	490

The velocity of the flowing water in the pipeline effects the movement and behaviour of the induced air in the pipeline. In practice, flow velocities are usually in the order of 1.0 – 2.0 m/s, hence velocities of up to 2.0 m/s were used. The movement of the air was tested at incremental increases of 0.5 m/s in the water velocity from 0.5 m/s to 2.0 m/s.

The flow required to maintain the chosen water velocities had to be determined. This was determined from the following simple relationship:

$$Q = VA \dots\dots\dots (4.2.3)$$

Where:

- Q = volumetric flow (m³/s)
- V = velocity of flow in pipeline (m/s)
- A = cross sectional area of pipeline (m²)

The flow meters could record flow rate in litres per second. The required flows were determined in litres per second, shown in **Table 4.2.2**.

Table 4.2.2: Required flows to obtain correct water velocities

Water velocity in pipeline (m/s)	Required flow (l/s)
0.5	4.752
1.0	9.503
1.5	14.255
2.0	19.007

By opening or closing the control valve in the system, shown in **Figure 4.2.10**, the required flow could be set.

The flow in the pipeline was monitored by the magnetic flow meters in the pump house (**Figure 4.2.10**). The readings of the flow meters could be displayed on the large digital display boards in the laboratory for easy viewing while testing.

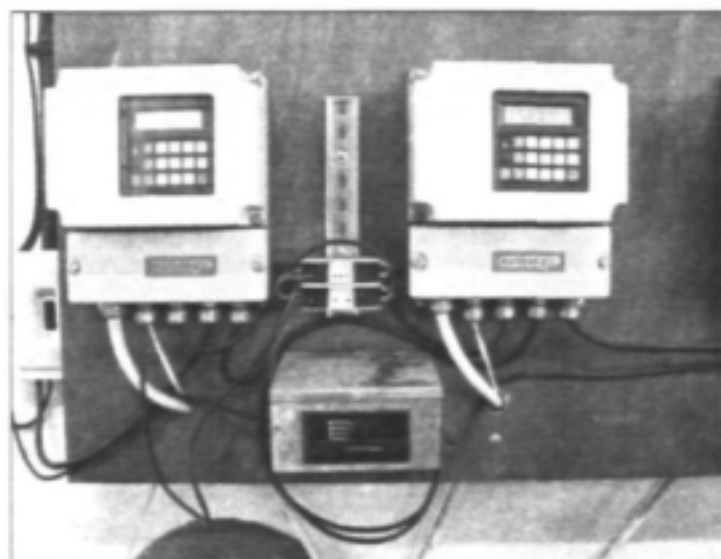


Figure 4.2.10: The displays of the digital flow meters

Ensuring that the length of lay-flat hose used was as short as possible solved the problem of turbulence created through the lay-flat hose.

Static slugs formed at some of the couplings of the pipeline. This made testing difficult, as the slug would increase in size, until the shear force on the slug was large enough to transport it down the pipeline. The results of these tests, where bubbles and slugs were identified, were removed from the final data set.

In some of the experiments, the bubble tended to move upstream from the induction point. The air transfer hose, which linked the AIB to the uPVC pipeline, was a relatively long, small-diameter pipe. The problem caused by this pipe was that for medium to large bubbles, the time taken to induce the full volume of air into the uPVC pipe was significant. This resulted in the transfer of a portion of the air bubble before the total bubble was released into the system generating a shock wave. This problem can be overcome by using the system shown in **Figure 4.2.11**, where a section of the pipeline can be isolated and filled with the required volume of air. The water is then diverted to pass through this section and the air carried away as one bubble.

This method of inducing air was used at the Institute de Mecanique des Fluides de Toulouse (IMFT) where a study on the structure of horizontal slug flow was conducted (J.R. Fagundes et al 1999).

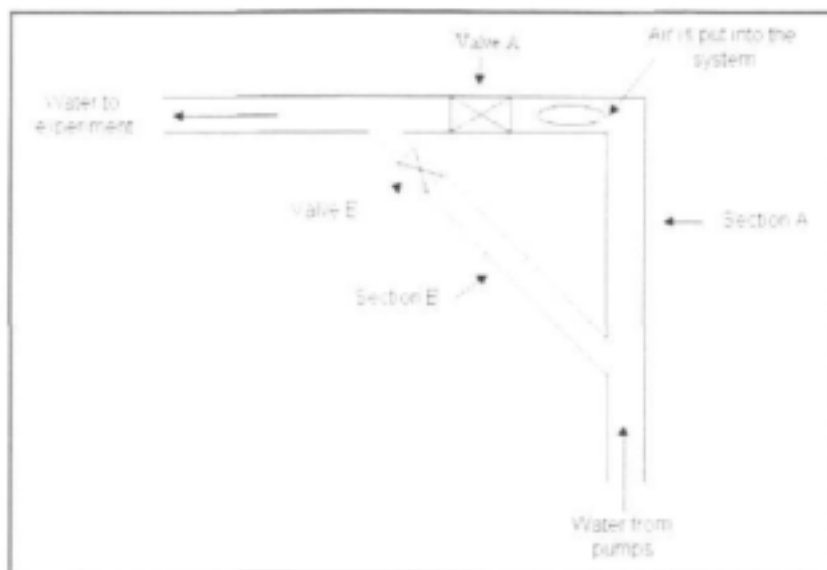


Figure 4.2.11: Alternative method of air induction

The supply pipe to the experiment was extended for those tests where the gradient was more than 10° (see **Figure 4.2.12**).



Figure 4.2.12: The extended supply pipe and improved angle of lay-flat hose

4.2.3.2 Second experimental set-up

To determine if there would be any scale effects, a similar set-up was made with a 160 mm diameter pipe. The following difficulties were encountered during the first experimental set-up and were taken into account for the second experimental set-up.

- Perspex flanges in order to join lengths of PVC pipes, using Perspex glue and bolting the two separate pipe sections together after placing a rubber gasket between them, proved unsuccessful, as the flanged connections leaked profusely.
- Glued ends tend to leak slightly and have to be sealed with silicone.
- A ribbed hose that initially was to be used for the inlet was very stiff and difficult to work with.
- A steel connector was used for induction of air into the pipeline instead of plastic connectors that leaked in previous set-up.
- Turbulence was caused in the pipeline by using a lay-flat hose at the inlet. The lay-flat hose would kink where it was too long. The turbulence caused made it difficult to perform accurate experiments. Due to the turbulence, air that was introduced would not travel at the crown of the pipe.
- Large static slugs were formed at connections and at the air induction connector when it was initially placed at the top of the pipe. The pipe was rotated until the connector was at the bottom of the pipe.
- The air transfer pipe, which linked the air regulator to the PVC pipeline, was a relatively long-, small-diameter pipe. The problem caused by this pipe was that for a medium to large bubble, the time taken to induce the full volume of air into the PVC pipe was too long. Thus, the bubble was drawn out and dispersed.

Transparent PVC pipes (160 mm in diameter) were used, in order to enable visual tracking of the air movement in the pipe. A pipe with an external diameter of 160 mm was chosen for the set-up, as it is a commonly used pipe size in practice. The pipe has an internal diameter of 153 mm and hence a wall thickness of 3.5 mm.

The sections of 160 mm diameter transparent PVC pipes were joined with 160 mm PVC connectors, each with a double rubber seal. This type of connection proved to be perfectly watertight. The pipeline supplying water was also fitted with an accurate ultrasonic flow meter, which was critical when the velocity of the water in the experimental pipeline had to be determined and controlled.

An isolating valve was inserted, enabling the control of the water inlet at all times. A flexible ribbed hose of 3 meters in length was used to join the water inlet with the transparent PVC pipe. Limited turbulence was experienced and kinks in the supply pipe (as previously experienced with the lay-flat hose) were eliminated. The 3-meter length for the flexible ribbed hose gave sufficient flexibility for changes in the gradient. At gradients larger than 10° , the additional length of supply pipe required to accommodate for the height of the test section was made possible by the inclusion of a 2 m length steel extension.

A 160 to 90 mm uPVC reducer (Figure 4.2.13) connects the 100 mm flexible ribbed hose with the 160 mm transparent PVC water pipe. The reducer was fastened to the flexible- and transparent PVC pipes by means of clamps on either side and was made water-tight with silicone sealant.

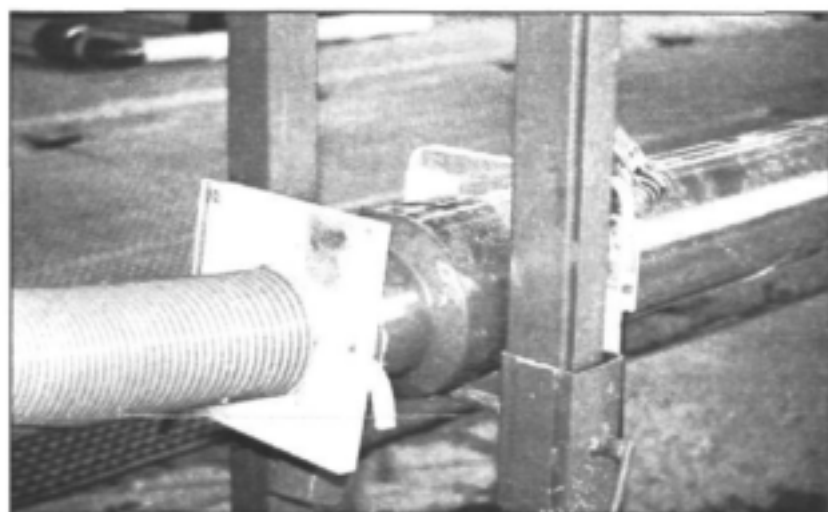


Figure 4.2.13: Connection of flexible hose with transparent water pipe at inlet

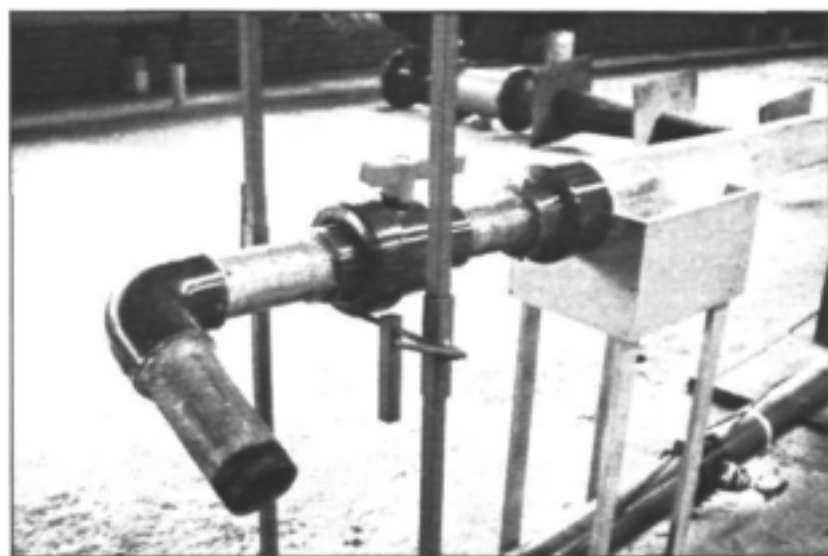


Figure 4.2.14: Construction of the outlet of the pipeline

Figure 4.2.14 shows the construction at the outlet of the pipeline. A 90 mm solvent ball valve was installed at the outlet of the pipeline. The isolation valve was joined to the 160 mm transparent PVC pipe by means of a short section of 80 mm steel pipe and sealed with silicone sealant. The water in the pipe was released by means of a 2.5 m lay-flat hose into a sump.

The experimental tests were executed at several gradients. Adjustable supports were a necessity in order to enable the adjustment of the support heights with ease. Five steel stands were used for supporting the 160 mm transparent PVC pipe. Each of these stands has an adjustable steel support, which clamped onto the stand. These steel supports can be adjusted vertically according to the heights required at each of the supported positions, enabling the setting of a specific gradient. In order for the 160 mm transparent PVC pipe to remain stationary at the steeper gradients, the pipe was firmly strapped to the adjustable steel supports. Adjustable straps with a ratchet mechanism were used to anchor the end of the test section (90° bend).

Similar to the first experimental set-up, three different bubble sizes were introduced into the 160 mm diameter pipeline, see **Table 4.2.3**.

Table 4.2.3: Bubble volumes

Bubble Size	Bubble Volume Millilitres (ml) (Atmospheric Pressure)
Small	61
Medium	221
Large	547

4.2.4 Methodology

The experimental data was recorded on a data sheet (example attached in **Appendix 4.2.1**). The following testing procedure was followed during the execution of the tests:

- Ensure that the flow meter is firmly attached to the pipe and that the settings on the flow meter are correct.
- Position the pipe such that it has the desired slope.
- Make sure all the supports are steady and that the pipeline is properly fastened to the supports.
- Fill the line and ensure that no air is trapped at any position.

- Regulate the water flow to the required velocity.
- Set the bubble size (in ml), gap between the bubbles (in milliseconds) and the number of bubbles to be induced into the system by means of the air induction system.
- Run the air-induction system.
- Take the times the induced bubbles take to pass a 5-meter distance marked off on the pipe in order to determine the bubble velocity.
- Record the movement of the bubbles or the accuracy of the test.
- Complete all the tests for that specific pipe slope and then empty and loosen the pipe from its supports in order to adjust the slope to the next required pipe gradient.

4.2.5 Experimental results

4.2.5.1 Velocity of the air bubble

The velocity of the bubbles in the pipeline was determined using equation 4.2.4

$$V = \frac{D}{t} \dots\dots\dots (4.2.4)$$

Where:

- V = Velocity at which the air bubble moves (m/s)
- D = Distance travelled by the bubble in a specific time (m)
- t = Time taken for the bubble to travel a distance (s)

The distance, which the bubble had to travel, was determined prior to the testing. In the case of the horizontal pipeline (angle = 0°), the pipe was marked, with insulation tape, 5 m downstream of the point of air induction. Once the pipe's gradient increased, two marks were made on the uPVC pipeline, 3 m either side of the air induction point.

A hand held stopwatch was used to measure the time taken, from the moment the bubble was injected into the uPVC pipe, to the moment the bubble crossed the insulation tape.

The time readings could vary considerably depending on whether smaller bubbles joined together to form a larger bubble or whether it broke-up into a conglomerate of small bubbles.

Twenty readings were taken for each of the three bubble sizes, at each of the four water flow velocities and at each of the gradients. The average of the time readings was taken to minimise the effect of extreme deviations. The data is shown in **Appendix 4.2.2**.

The distance travelled was then divided by the average time, to obtain the average velocity of the bubble in the pipeline. The normal volume of the bubbles was determined at atmospheric pressure and it was assumed that the pressure was constant.

By applying Boyle's law, equation 4.2.5, the normal volume could be determined.

$$P_1 V_1 = P_2 V_2 \dots\dots\dots (4.2.5)$$

Where:

- P_1 = Pressure of air at point 1
- V_1 = Volume of air at point 1
- P_2 = Pressure of air at point 2
- V_2 = Volume of air at point 2

4.2.5.2 Results of the 110 mm diameter experimental set-up

For the different pipe slopes and bubble sizes the results are graphically represented in **Figures 4.2.15 to 4.2.21**. Each Figure represents a specific pipe slope installation.

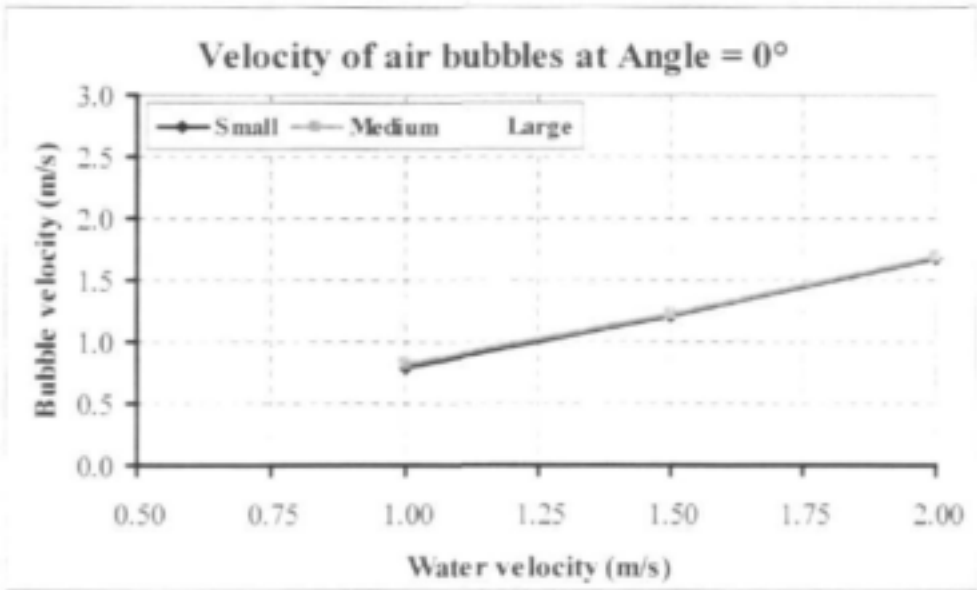


Figure 4.2.15: Movement of bubble in a horizontal pipe (110 mm diameter)

Figure 4.2.15 indicates an interesting phenomenon, which suggests that if slug flow is generated, the hydraulic grade line becomes steeper; hence the air bubble moves faster than the average velocity of the liquid.

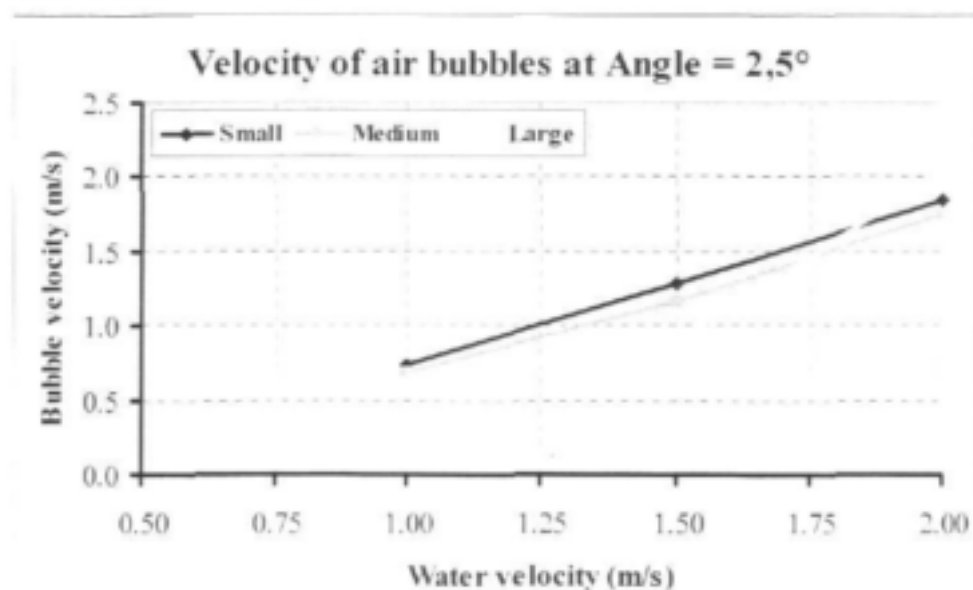


Figure 4.2.16: Movement of bubble in pipe angled at 2,5° (110 mm diameter)

No points were plotted for a water velocity of 0,5 m/s, Figure 4.2.15 and Figure 4.2.16. This is due to the fact that for such a low velocity at relatively low gradients, the bubbles tended to group together to form large stationary slugs. Once these slugs grew too large, smaller bubbles would disengage themselves at the tail of the slug. The large stationary slugs prevented the accurate measurement of the bubble velocities.

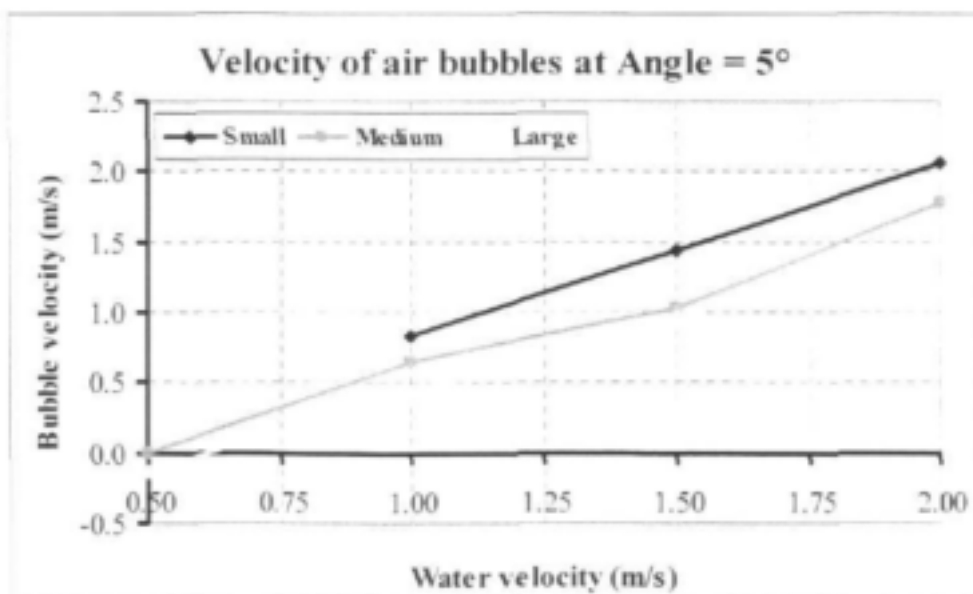


Figure 4.2.17: Movement of bubble in pipe angled at 5° (110 mm diameter)

Figure 4.2.17 indicates that at a velocity of 0.5 m/s the large bubble tended to move upstream and the medium bubble was stagnant. This indicates that at an angle of 5° the velocity must be at least 0.6 m/s to transport the air.

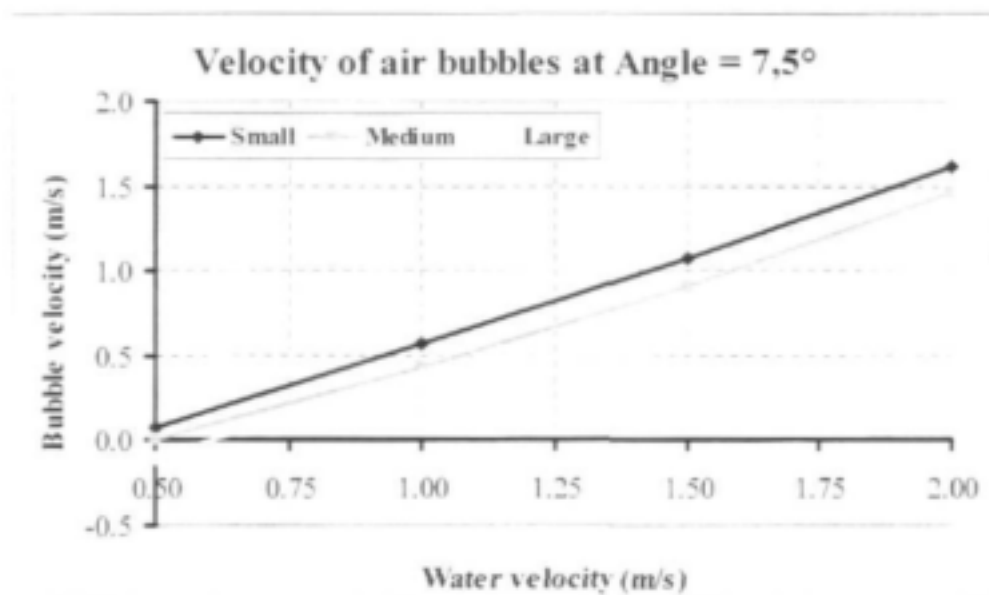


Figure 4.2.18: Movement of bubble in pipe angled at $7,5^\circ$ (110 mm diameter)

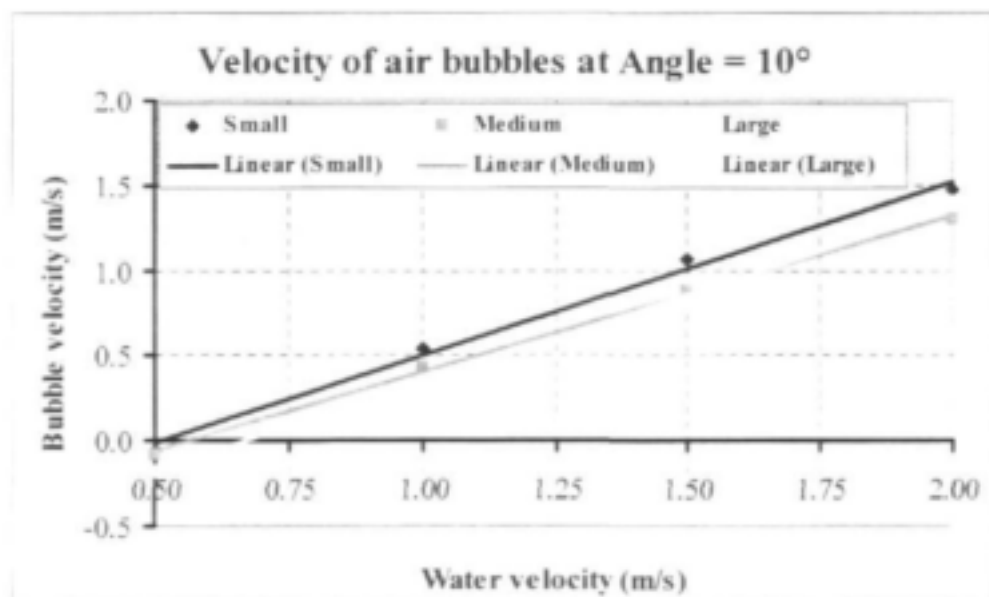


Figure 4.2.19: Movement of bubble in pipe angled at 10° (110 mm diameter)

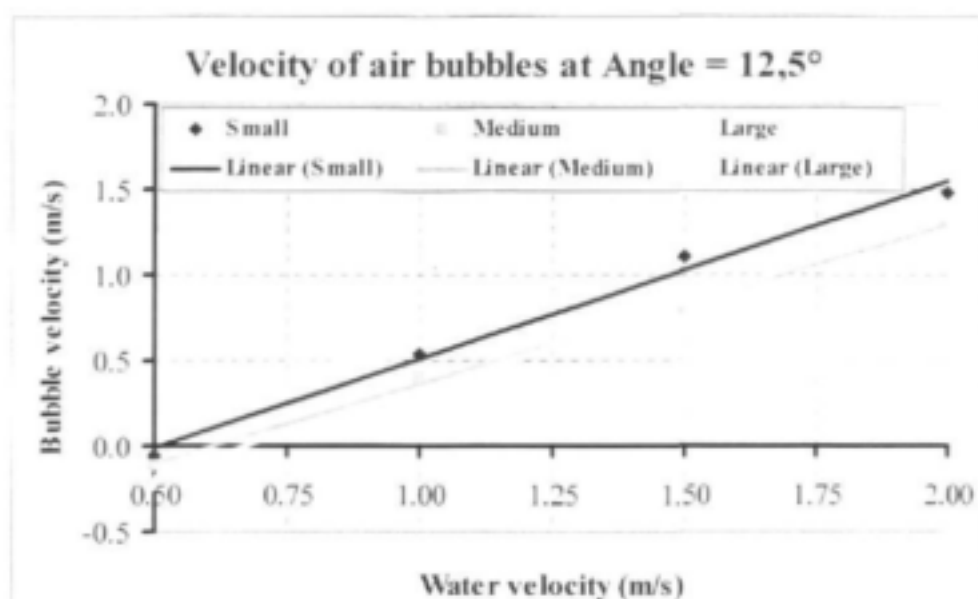


Figure 4.2.20: Movement of bubble in pipe angled at 12,5° (110 mm diameter)

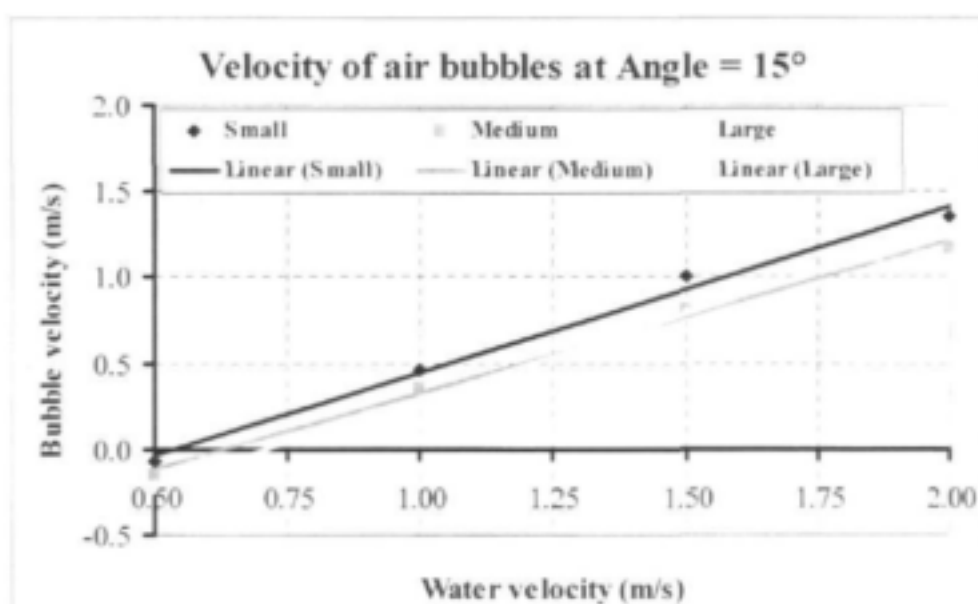


Figure 4.2.21: Movement of bubble in pipe angled at 15° (110 mm diameter)

The Figures 4.2.15 to 4.2.21 reflect a definite relationship between the volume of the induced bubble and the velocity of the bubble in the 110 mm diameter pipeline. For low gradients, a large air bubble moved at a velocity higher than the average velocity of the water. This is due to the phenomenon that the resistance to flow of the water is higher than that of air, resulting in slug flow.

However, as the pipe gradient increased, a large bubble generally travelled slower than the other bubble sizes, due to the effect of buoyancy, working in the opposite direction. Buoyancy, which is directly proportional to the volume of the bubble, played a large role in the movement of the bubbles at a low water velocity of 0.5 m/s.

Figures 4.2.15 to 4.2.21 also show that for a water velocity equal to or larger than 1.0 m/s, and an angle less than 3° , the shear force on the large bubble far exceeds the force of buoyancy.

The velocity of the small and medium bubbles tends to be the same for a slope equal to 0° and a water velocity equal to or greater than 1.0 m/s. Once the slope increases though, the differences in velocity become more apparent due to the increasing influence of buoyancy.

In the following **Figures 4.1.22 to 4.1.25** the relationship is shown between the bubble velocity and pipe slope for a constant water velocity.

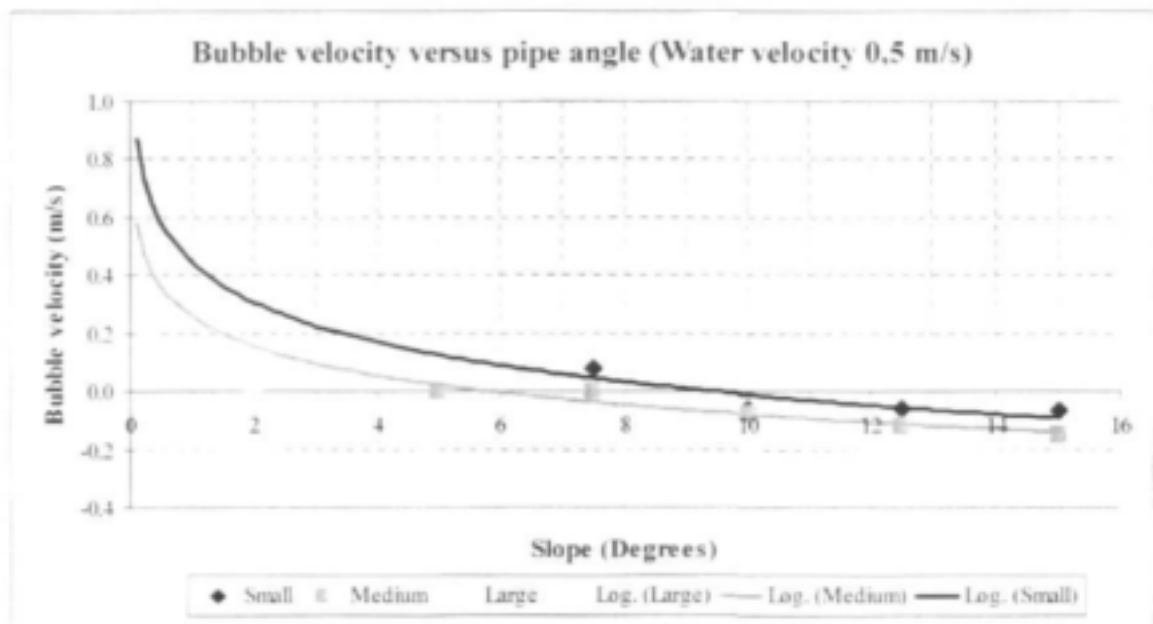


Figure 4.2.22: Relationship of bubble velocity to pipe slope for a flow velocity of 0.5 m/s (110 mm diameter)

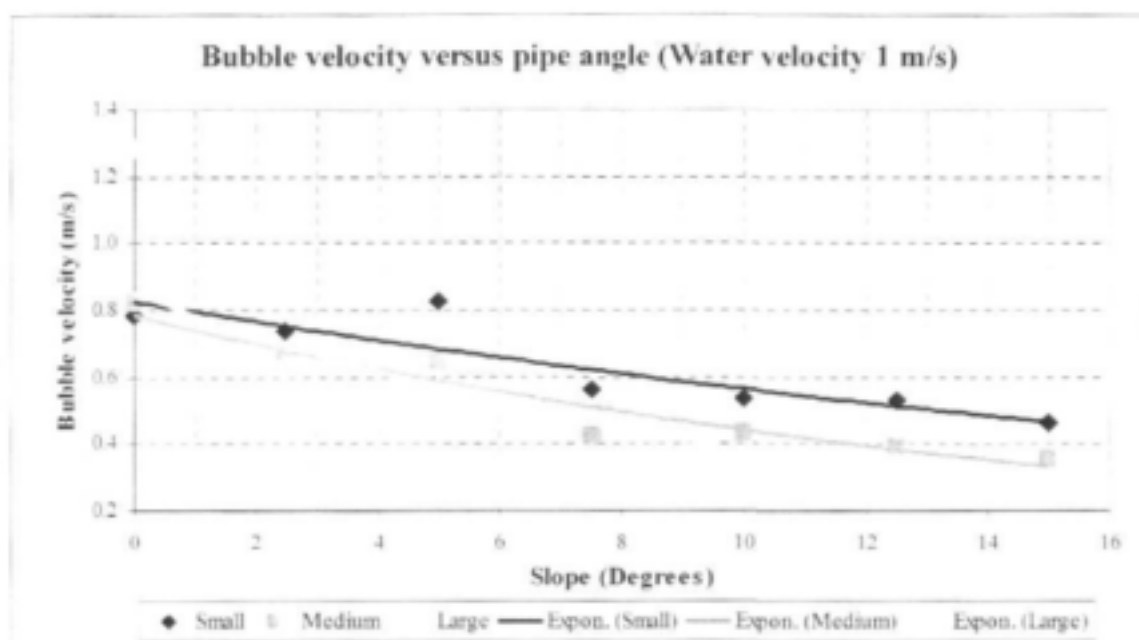


Figure 4.2.23: Relationship of bubble velocity to pipe slope for a flow velocity of 1,0 m/s (110 mm diameter)

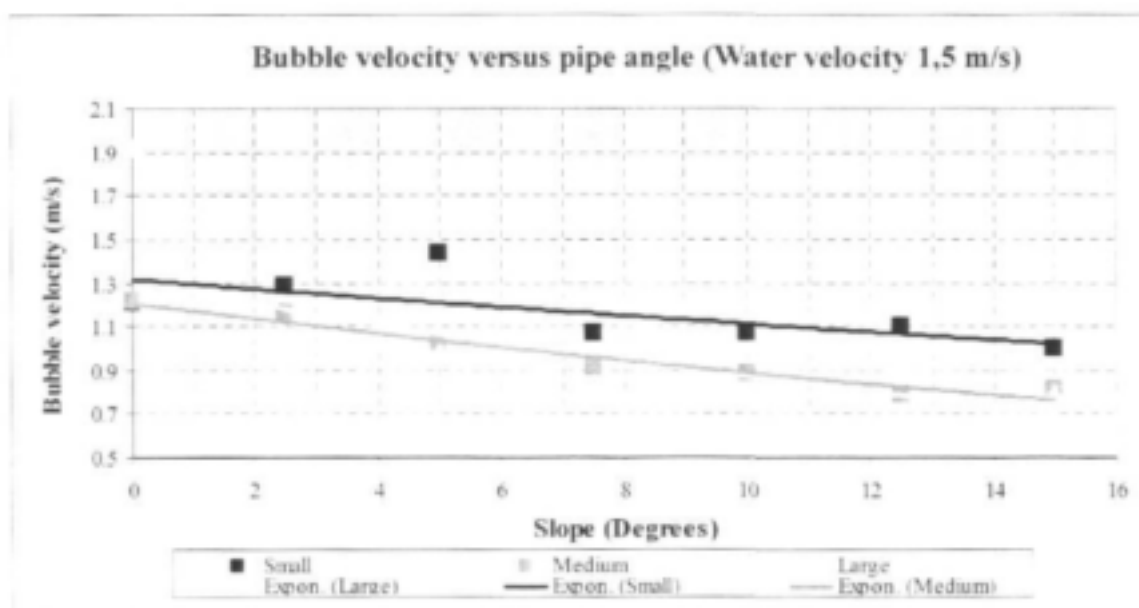


Figure 4.2.24: Relationship of bubble velocity to pipe slope for a flow velocity of 1,5 m/s (110 mm diameter)

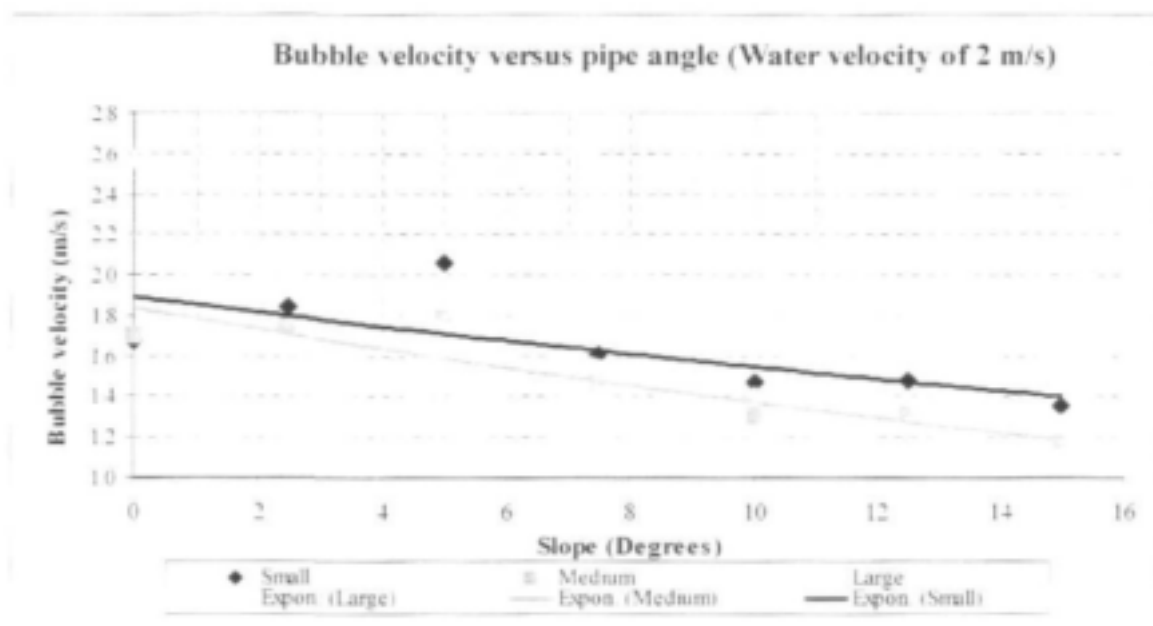


Figure 4.2.25: Relationship of bubble velocity to pipe slope for a flow velocity of 2.0 m/s (110 mm diameter)

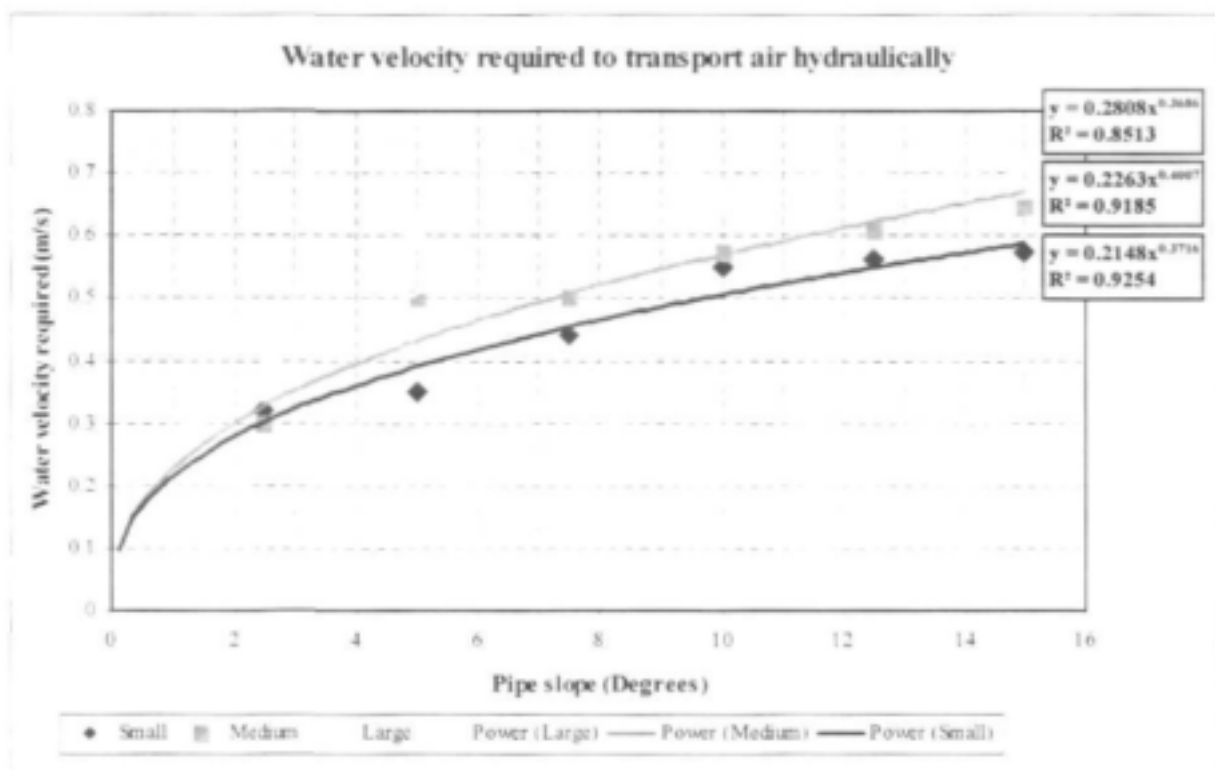


Figure 4.2.26: Relationship of clearing velocity in 110 mm Ø pipe to varied bubble size

Subsequently these results led to the following relationship:

$$V_{\min} = a \theta^b \dots \dots \dots (4.2.6)$$

Where:

V_{min} = Minimum velocity required to transport air (m/s)

a = Constant value (experimentally determined)

θ = Slope of the pipe (degrees)

b = Constant value (experimentally determined)

The equations of the trend lines fitted through the data points (for a 110 mm Ø pipe) in Figure 4.2.29 for different bubble sizes as reflected in Table 4.2.4.

Table 4.2.4: The relationships for a 110 mm Ø pipe

Bubble size	a	b	R ²
Small	0.2148	0.3716	0.9254
Medium	0.2263	0.4007	0.9185
Large	0.2808	0.3686	0.8513

These equations were derived from the tests conducted on the 110 mm Ø pipe with angles varying from a horizontal to a 15° slope. For pipe angles larger than 15°, experimental results might reflect velocities that are less than the extrapolated relationships. The high correlation coefficient reflects a good fit for the data sets. During a similar experimental set-up, with an angle of 22,5°, the velocity of flow required to transport a medium size bubble was 0.8 m/s. This coordinate plotted on the graph lies very close to the fitted curve for medium size bubbles.

The equation of Kalinske and Bliss (1943) contains the following factor:

$$\sqrt{gD} \dots\dots\dots (4.2.7)$$

Where:

g = Gravitational acceleration (m/s²)

D = Internal pipe diameter (m)

When the equation of Wisner, Mohsen & Kouwen (1975) Equation 4.2.2 is written in terms of velocity, the same factor appears. Therefore, to make the equations of the fitted curves applicable to all pipe diameters, the factor must be included in the equation in some way.

Using the derived relationships the influence of the pipe diameter was incorporated and the relationships were changed as follows:

$$V_{min} = a\sqrt{gD} \theta^b \dots\dots\dots(4.2.8)$$

Where:

V_{min} = Minimum velocity required to transport air (m/s)

g = Gravitational acceleration (m/s²)

D = Internal pipe diameter (m)

a = Constant value (experimentally determined)

θ = Absolute value of slope of the pipe measured from the horizontal (degrees)

b = Constant value (experimentally determined)

Table 4.2.5: General relationship for determination of the required flow velocity at which air is transported in a pipeline (based on 110 mm diameter experimental data)

Bubble size	a	b	R ²
Small	0.2068	0.3716	0.9254
Medium	0.2178	0.4007	0.9185
Large	0.2703	0.3686	0.8513

4.2.5.3 Results of the 160 mm diameter experimental set-up

The data of the 160 mm diameter pipe was also analysed to determine what the scale effect would be on the proposed formulae. For the different pipe slopes and bubble sizes the results for the 160 mm pipe are graphically represented in **Figures 4.2.27 to 4.2.33**. Each Figure represents a specific pipe slope installation.

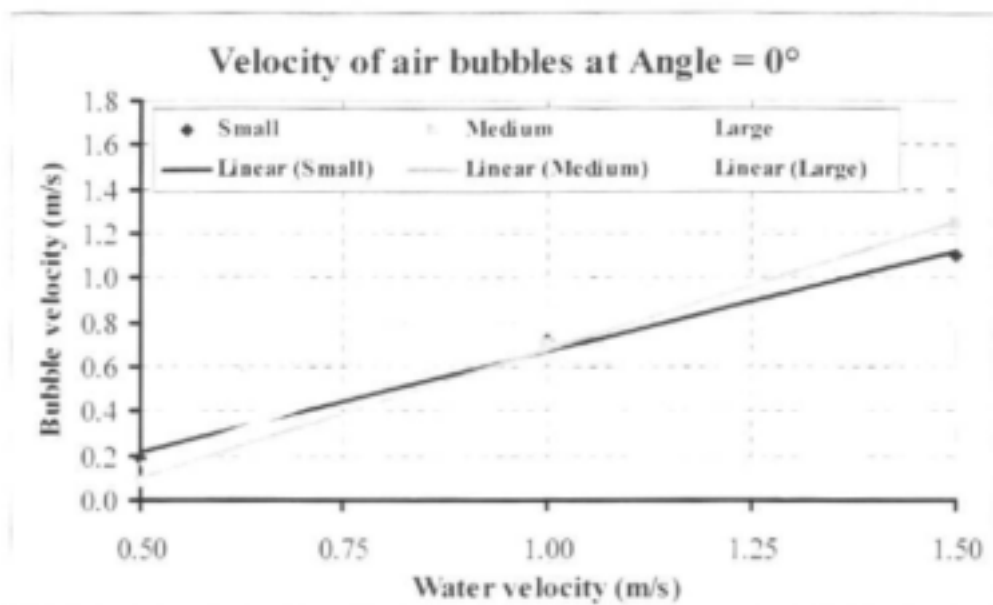


Figure 4.2.27: Movement of bubble in a horizontal pipe (160 mm diameter)

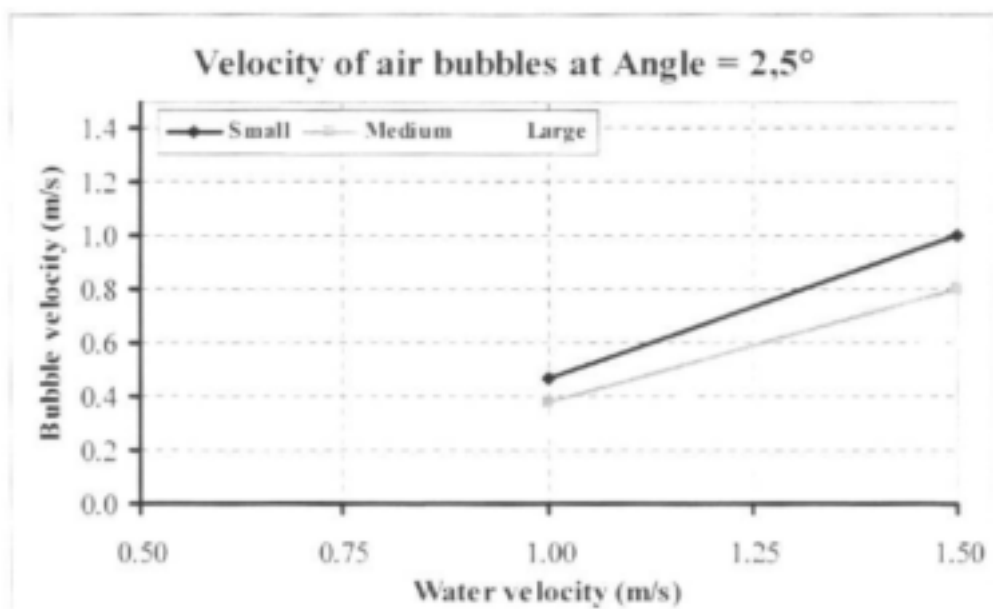


Figure 4.2.28: Movement of bubble in pipe angled at $2,5^\circ$ (160 mm diameter)

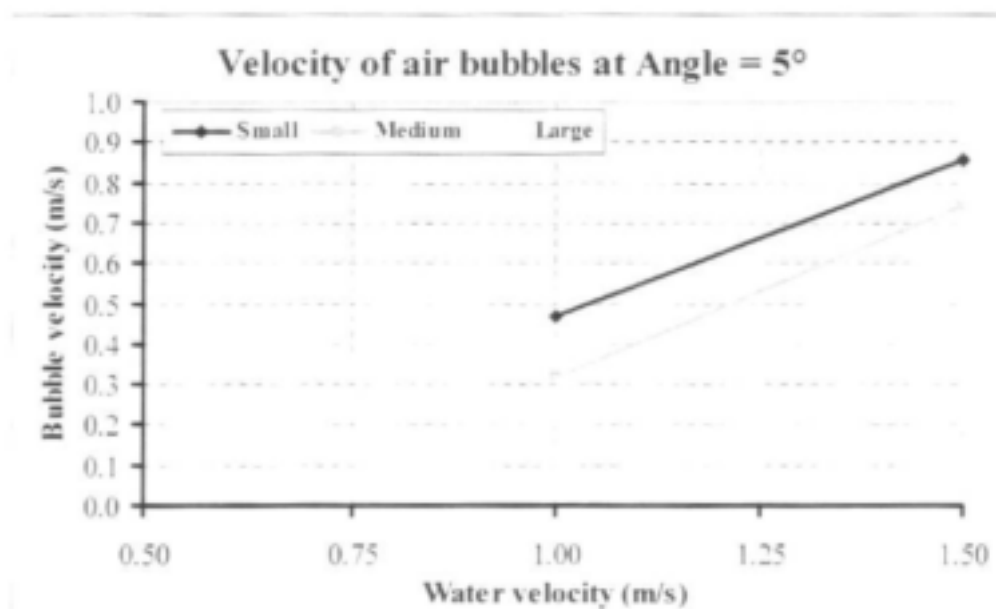


Figure 4.2.29: Movement of bubble in pipe angled at 5° (160 mm diameter)

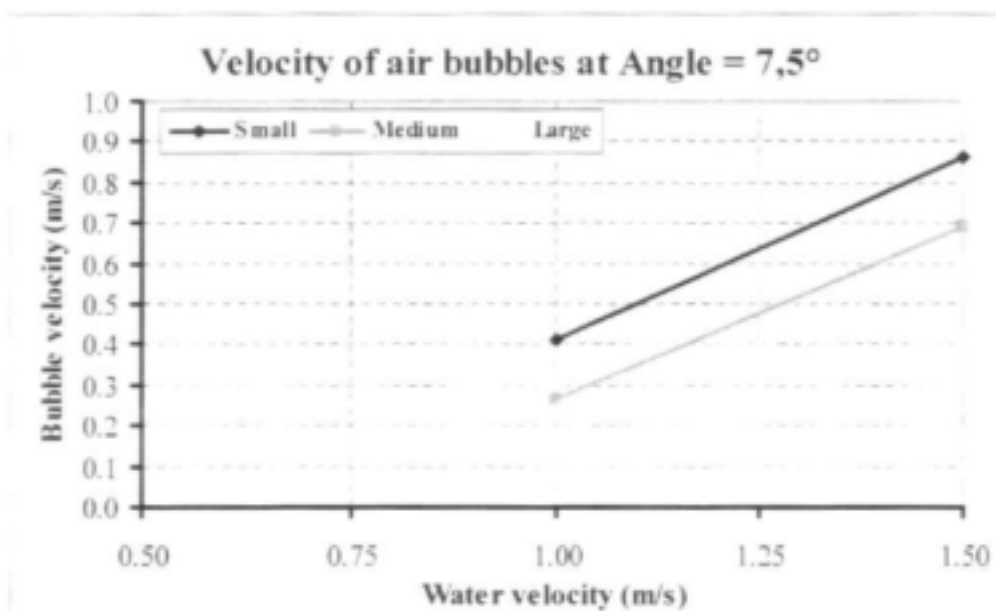


Figure 4.2.30: Movement of bubble in pipe angled at $7,5^{\circ}$ (160 mm diameter)

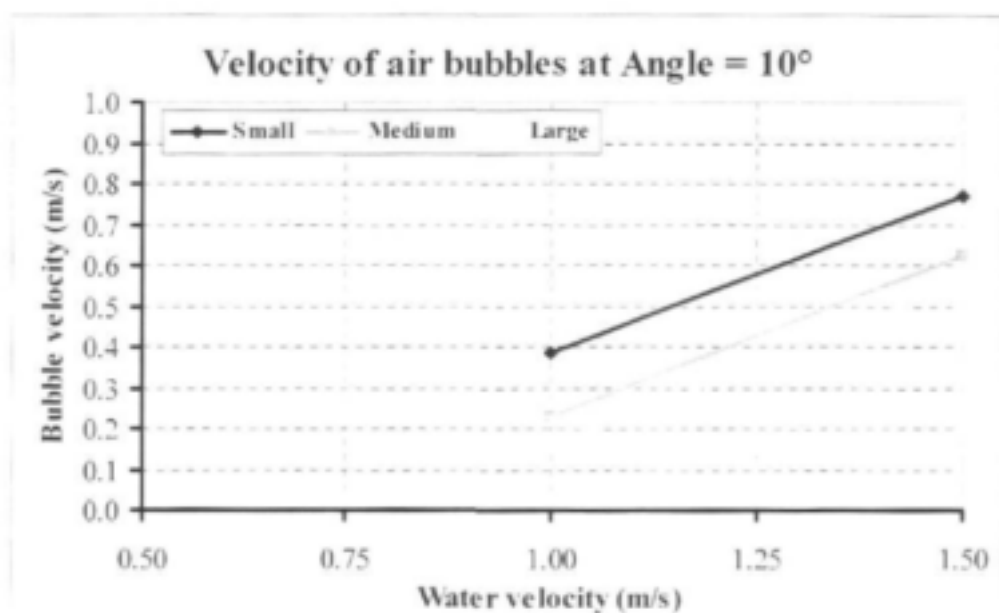


Figure 4.2.31: Movement of bubble in pipe angled at 10° (160 mm diameter)

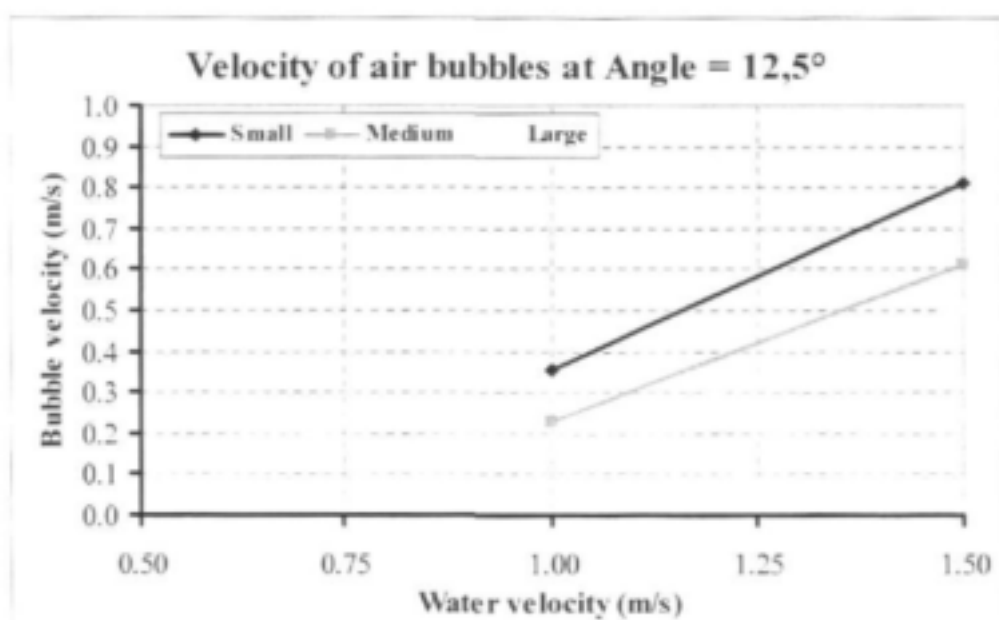


Figure 4.2.32: Movement of bubble in pipe angled at 12,5° (160 mm diameter)

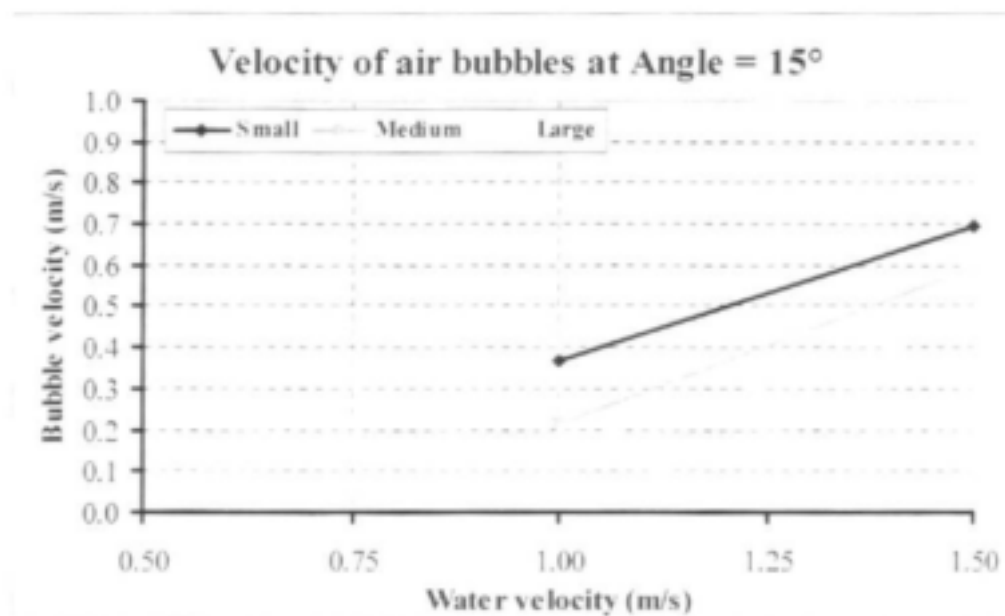


Figure 4.2.33: Movement of bubble in pipe angled at 15° (160 mm diameter)

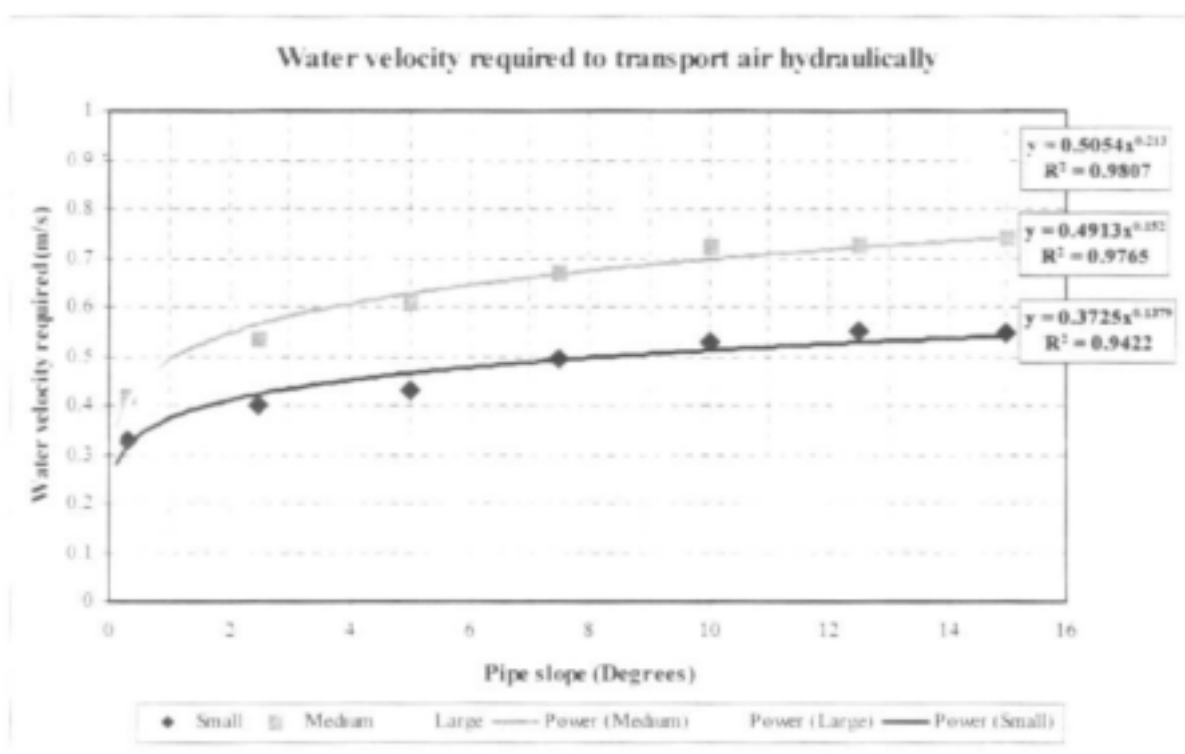


Figure 4.2.34: Relationship of clearing velocity in 160 mm diameter pipe with varied bubble size

The equations of the trend lines fitted through the data points (for a 160 mm Ø pipe) in Figure 4.2.34 for different bubble sizes are reflected in Table 4.2.6. Similar to the 110 mm diameter pipe the square root of the gravitational acceleration times the diameter was incorporated and the relationships were changed as follows:

Table 4.2.6: General relationship for determination of the required flow velocity at which air is transported in a pipeline (based on 160 mm diameter experimental data)

Bubble size	a	b	R ²
Small	0.4125	0.213	0.9807
Medium	0.4010	0.152	0.9765
Large	0.3041	0.138	0.9422

The values for “a” and “b” as provided in **Table 4.2.6** should be used in Equation 4.2.8.

4.2.5.4 Comparison of results to existing theory

The two most commonly used equations for the determination of the required flow velocity to hydraulically remove air from a pipeline are that of Kalinske & Bliss Equation 4.2.1 Wisner Equation 4.2.2.

Figure 4.2.35 reflects a graphical comparison between these two relationships and the results that were obtained for different bubble sizes in the first experimental set-up with the 110 mm diameter pipeline.

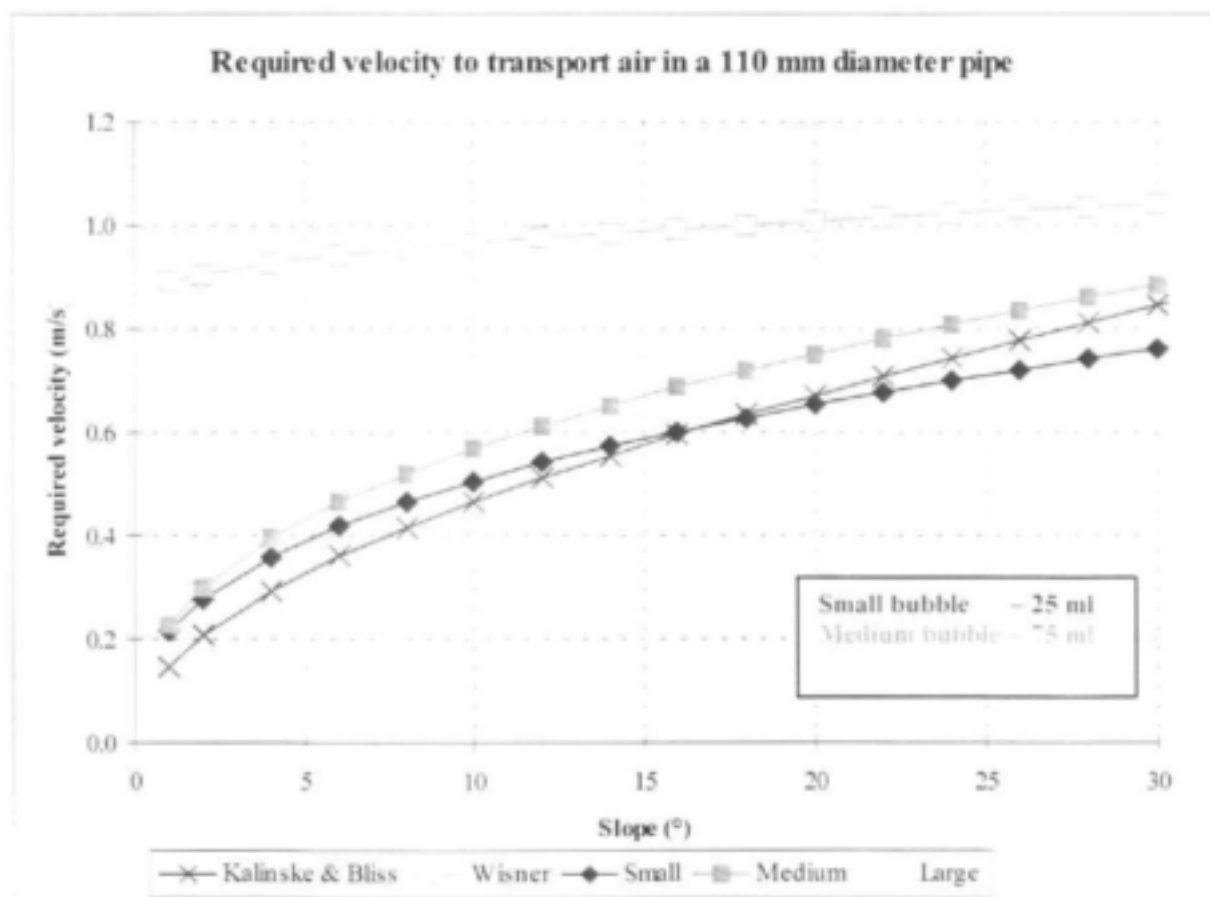


Figure 4.2.35: Comparison of experimental results with known relationships for the clearing velocity of air in the 110 mm diameter pipe

It appears from **Figure 4.2.35** that a much higher clearing velocity is required when the Wisner et al. (1975) relationship is used and the slope of the pipe is less than 20° . Kalinske & Bliss (1943) require a higher clearing velocity for angles more than 43° (where these two relationships coalesce). The discrepancy required a comparison between the two relationships.

Kalinske & Bliss performed their experiments on small bubbles. This theory compares well, as the velocities obtained in the experimental results for a small bubble are relatively close. It seems as though the volume of Kalinske & Bliss's small bubble could have been less than the volume of the small bubbles in this experiment (110 mm diameter pipeline experimental set-up).

Wisner performed experiments on larger bubbles of size up to half the diameter of the pipe as well as using a larger diameter pipe, 250 mm. This also compares favourably to the results obtained in this experiment, as the line for large bubbles lies closer to that of Wisner. In this instance it seems that Wisner's bubbles were of a larger volume than the volume of the large bubble in this experiment. Wisner might also have increased his velocities to be conservative, to ensure that the air will be transported hydraulically.

Figure 4.2.36 reflects a graphical comparison between the two most commonly used relationships and the results that were obtained for different bubble sizes in the second experimental set-up with the 160 mm diameter pipeline.

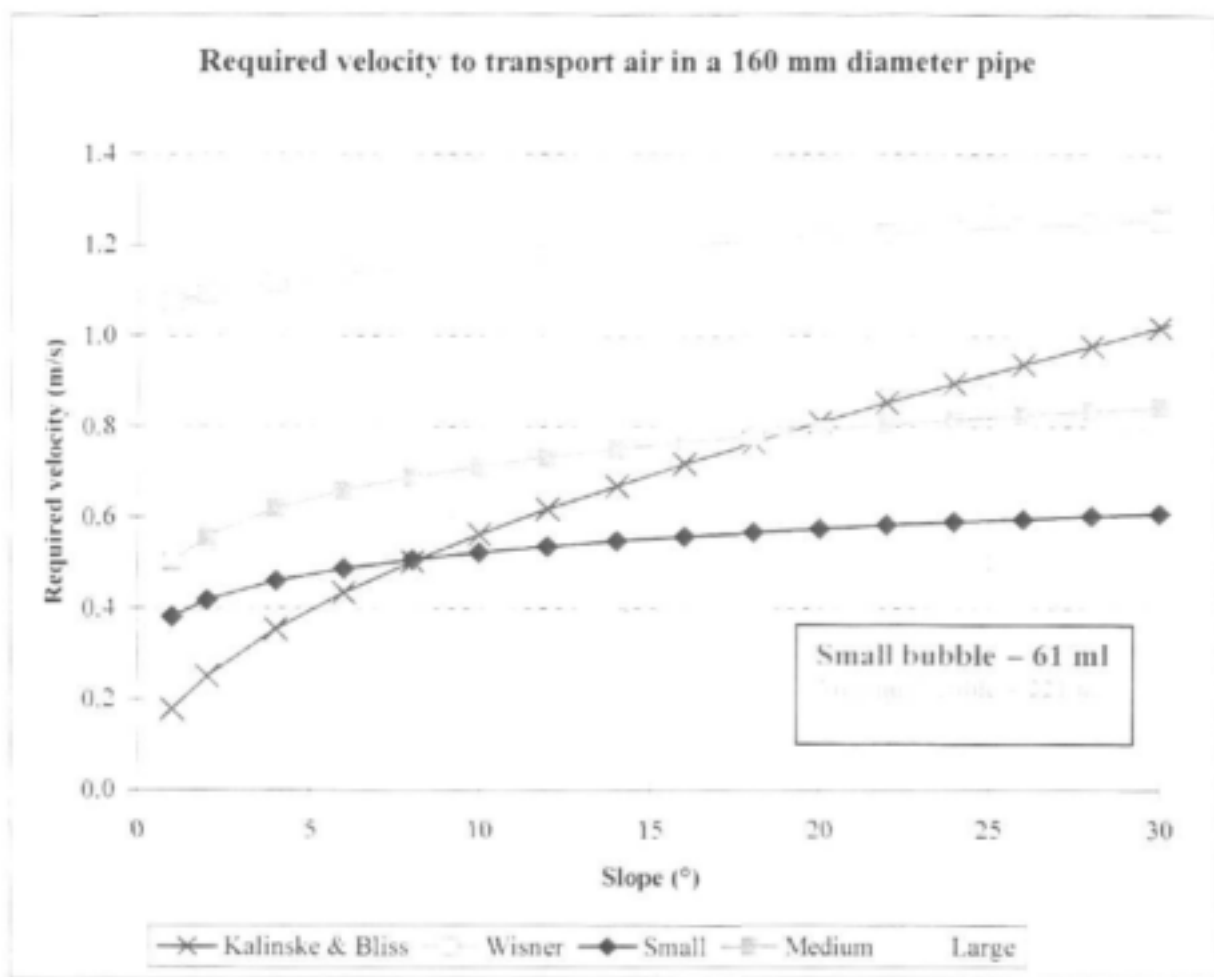


Figure 4.2.36: Comparison of experimental results with known relationships for the clearing velocity of air in the 160 mm diameter pipe

These equations provide satisfactory relationships and **Figure 4.2.37** reflects the results for a 500 mm diameter pipe. **Figure 4.2.37** also indicates the function plot of Kalinske & Bliss (1943) and Wisner et al.(1975).

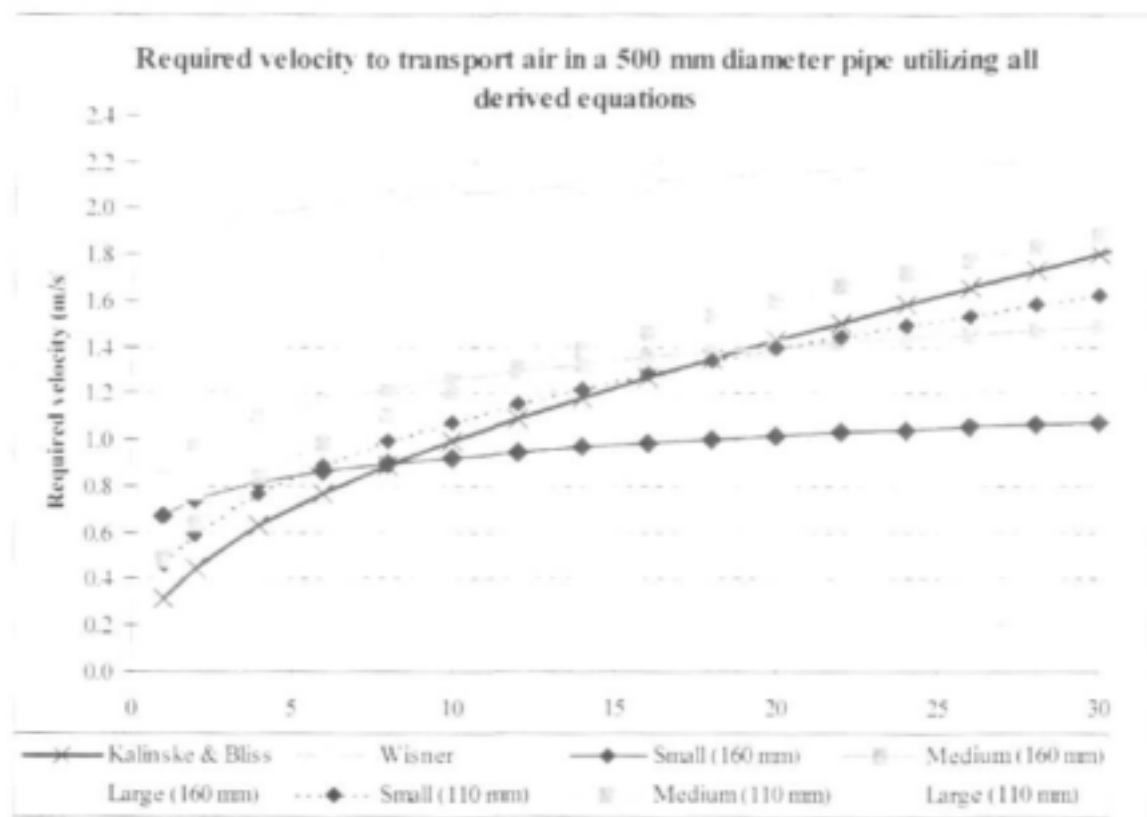


Figure 4.2.37: Comparison of the required velocity for air transport in a 500 mm diameter pipe

Wisner et al. utilized a 250 mm diameter pipe in their experimental set-up. One of the main objectives of their investigation was to determine the scale effect related to the mechanics of the air pocket. It is mentioned that other researchers have investigated air movement utilizing small pipelines 25 mm, 100 mm etc. The reason for this is that laboratory facilities are limited in terms of flow rate, available head and floor space.

Investigators have indicated that for slopes ranging between 15° and 60°, the clearing velocity remains fairly constant (Wisner et al., 1975). Wisner also indicated that from the Buckingham theorem a functional relationship should exist among the governing parameters:

$$\frac{V_c}{\sqrt{gD}} = f\left(\frac{L_B}{D}, R, \theta\right) \dots\dots\dots (4.2.9)$$

Where:

- V_c = Clearing velocity (m/s)
- g = Gravitational acceleration (m/s²)
- D = Internal pipe diameter (m)

- $L_B/D =$ Constant value (experimentally determined)
 $R =$ Reynolds number
 $\theta =$ Pipe slope (degrees)

From Wisner et al.(1975) it could be conducted that for values of the Reynold's number above 100 000, the relationship $\frac{V_c}{\sqrt{gD}}$ becomes independent of the Reynolds number. Based on Veronese's (1937) and Wisner, Mohsen, and Kouwen's (1975) results on limit length and limit velocity it indicated that for a certain velocity, stability would be achieved for a specific bubble size. **It was, however, indicated that these values are unique for a particular diameter and slope.**

Wisner also investigated the time of clearance of particular size air pockets and concluded that larger pipes clear much faster than smaller ones at the same Froudian velocity. **This would indicate that there is no similitude, which can be, used to extrapolate adequately the laboratory results to larger pipes used in practice.**

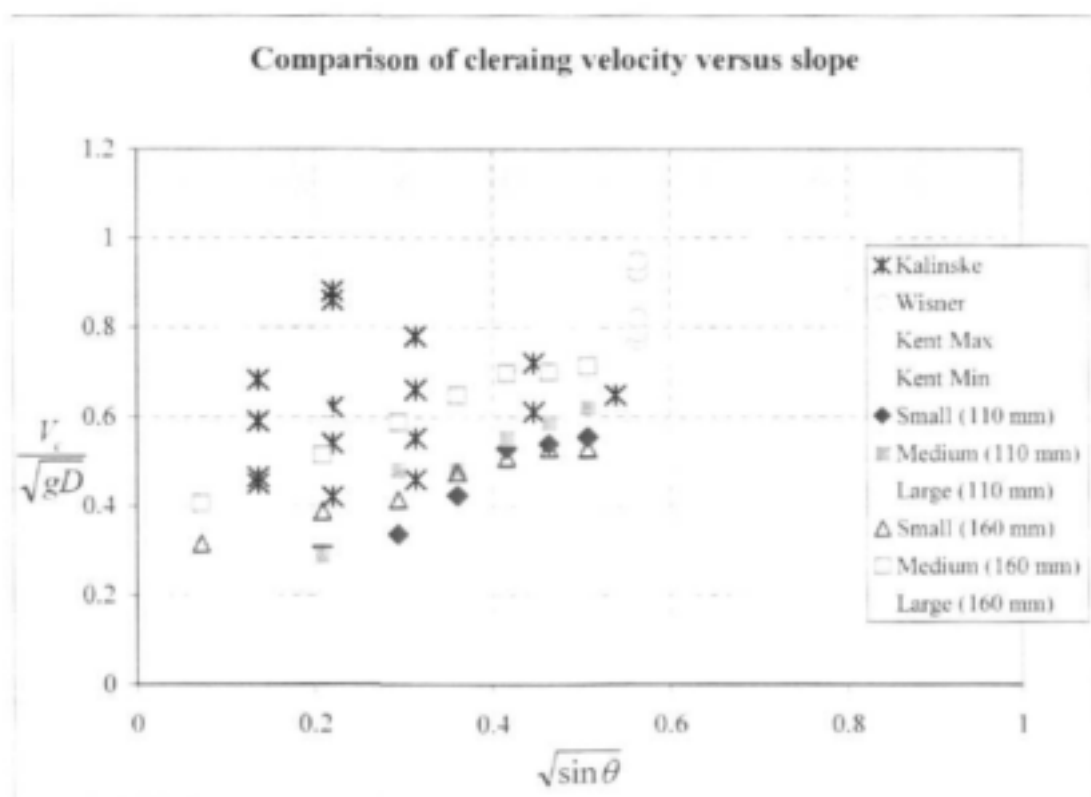


Figure 4.2.38: Comparison of various researchers data of clearing velocity versus pipe slope

Figure 4.2.38 clearly indicates that the clearing velocity for larger pipe diameters cannot simply be extrapolated using the formulae developed from the experimental work. The reason for this is because the shape of the bubble will change for a larger diameter pipe.

4.2.6 Conclusions and recommendations

4.2.6.1 Conclusions

The experiments were performed on a 110 mm and 160 mm diameter transparent PVC pipeline. The experimental results proved that the size of an air bubble has a significant influence on the movement and behaviour of the bubble in pipelines flowing full at different angles.

The results obtained also seem to lie in between the existing theories of Kalinske & Bliss (1943) and Wisner et al. (1975). The results showed a very good correlation compared to other investigator's results. The change in diameter from a 110 mm to the 160 mm diameter also confirmed the work by Wisner who concluded that there is no similitude, which can be utilized to extrapolate the data of experimental work. The main reason for this is the fact that the bubble characteristics will not be similar in larger diameter pipes. For each diameter and slope there will be a so-called limit length (L_B) for air pockets, which will determine the required velocity to clear. This limit length needs to be determined for every possible pipe diameter and pipe slope to accurately determine the clearing velocity.

4.2.6.2 Recommendations

Based on the results that were obtained it is recommended that the following relationship be used to determine whether the flow velocity is sufficient to transport air in a pipeline:

$$V_{min} = 0.2178\sqrt{gD\theta^{0.4007}} \dots\dots\dots (4.2.10)$$

Where:

- V_{min} = Minimum velocity required to transport air (m/s)
- g = Gravitational acceleration (m/s^2)
- D = Internal pipe diameter (m)
- θ = Absolute value of the slope of the pipe (degrees)

4.3 Experimental work to quantify the discontinuity required in a pipeline to effectively capture the free air

4.3.1 Introduction

Engineering skill and effort, based on physical laws, is normally applied to create cost effective functional products. Misconceptions regarding the operation of air valves and their positioning have led to ineffective conveyance systems, however. One of the parameters of air valves that contributed to ineffective de-aeration of pipelines was insufficient discontinuity in the pipe through which the free air has to be released into the air valve. This resulted in situations where free air remained trapped in the pipeline, leading to hydraulic capacity far less than the designed capacity. The presence of free air in a pipeline reduces the capacity but also complicates the filling of the pipeline, creates dynamic pressures when the air is released in an uncontrolled manner and increases the tendency towards corrosion of the inner pipe surfaces.

4.3.2 Objectives of the experimental investigation

The purpose of this experimental investigation was to determine the size of the discontinuity to capture the free air that is transported hydraulically in a pipeline. Tests were performed for various parameters, which influence the volume of air that will be de-aerated through the off-take. The parameters that dominate the process are:

- Velocity;
- Air bubble sizes;
- Pipe slope; and
- Size of the off take.

No literature could be found regarding the required discontinuity to effectively de-aerate a pipeline system.

4.3.3 Experimental Set-up

The experimental set-up was constructed in the laboratories of the Department of Water Affairs and Forestry (Pretoria West). Pumps delivered water to a constant head tank, from where the water was fed into a network that ran through the laboratory. The constant pressure supply was then regulated through a valve to create a specific flow through the experiment.

Two different pipe sizes were used in the experiment, viz. a 110 mm ND and a 160 mm ND pipe.

4.3.3.1 First experimental set-up

Water that flows through the experimental setup drains back into the pump sump, making it a closed system. **Figure 4.3.1** provides a schematic layout of the experiment.

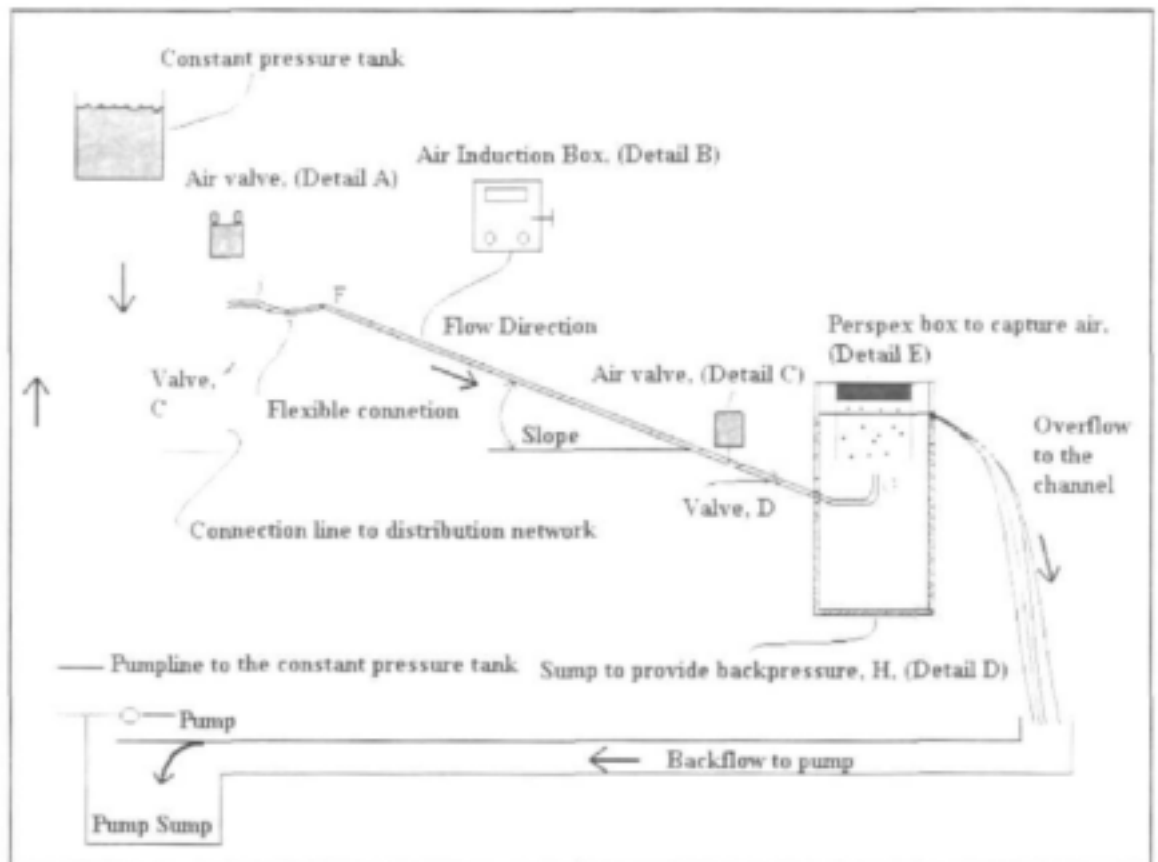


Figure 4.3.1: Schematic layout of the Experimental setup

Photographic details of some of the components of the experimental layout, shown in **Figure 4.3.1**, are referred to below.

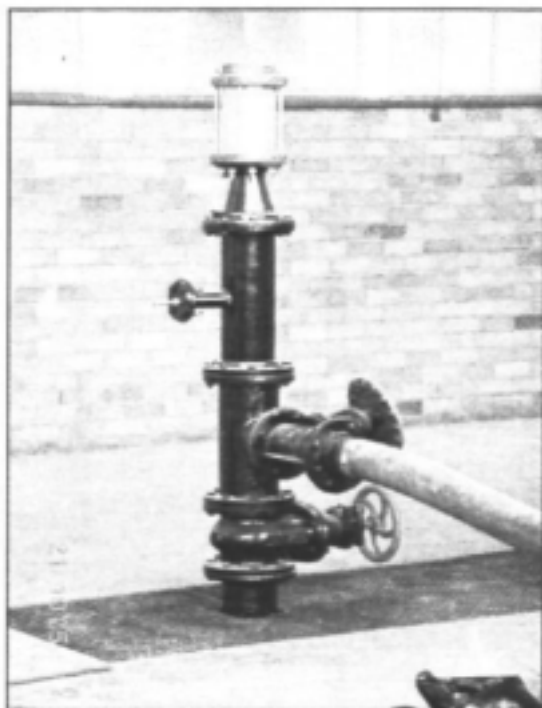


Figure 4.3.2: Detail A of the schematic drawing (Figure 4.3.1) showing the air valve

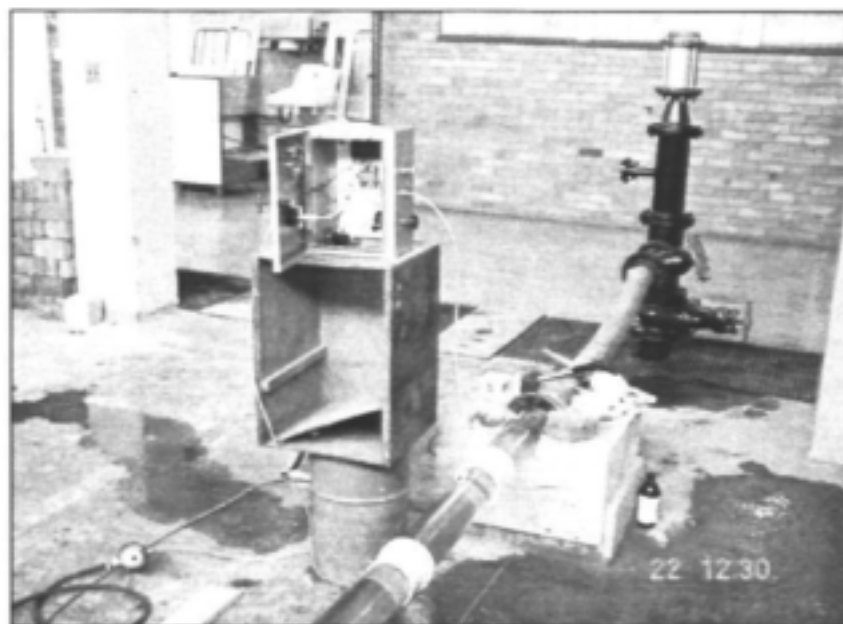


Figure 4.3.3: Detail B of the schematic drawing (Figure 4.3.1) showing the Air Induction Box



Figure 4.3.4: Detail C of the schematic drawing (Figure 4.3.1) showing the transparent air valve installed on the off-take



Figure 4.3.5: Detail D of the schematic drawing (Figure 4.3.1) showing the sump at the end of the pipe to provide back-pressure and accommodate the Perspex box to capture the air that had been transported through the pipeline

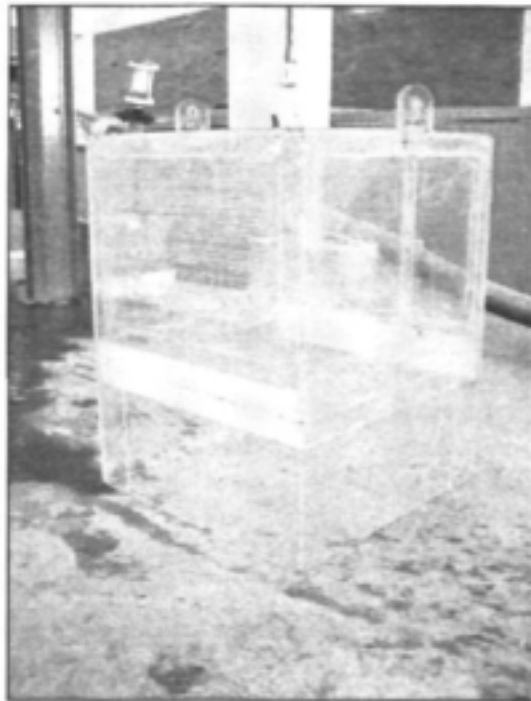


Figure 4.3.6: Detail E of the schematic drawing (Figure 4.3.1) showing the Perspex box that was used to capture the air that had been transported through the pipeline



Figure 4.3.7: The pipeline set-up at a slope of 15°

Free air in the system was captured on the upstream side of the experiment and released through an air valve, Detail A, **Figure 4.3.2**. This air valve was included to ensure that no air was fed into the experiment from the supply side. The air valve was placed on a standpipe in such a way that it captured all the air before it entered the experimental setup.

Air was introduced into the system through the Air Induction Box (AIB), Detail B, **Figure 4.3.3**. The AIB was specifically designed and built for this experiment by Productive Solutions. The AIB had the facility to set the bubble size (BUB) by setting the opening time of the constant pressure solenoid valve in milliseconds, while the time delay between consecutive air bubbles could also be set (GAP) as well as the number of air bubbles that would be released (QTY) for a specific test. The photos in **Figures 4.3.8 to 4.3.10** give a quantitative indication of the different air bubble sizes that were used in the tests.

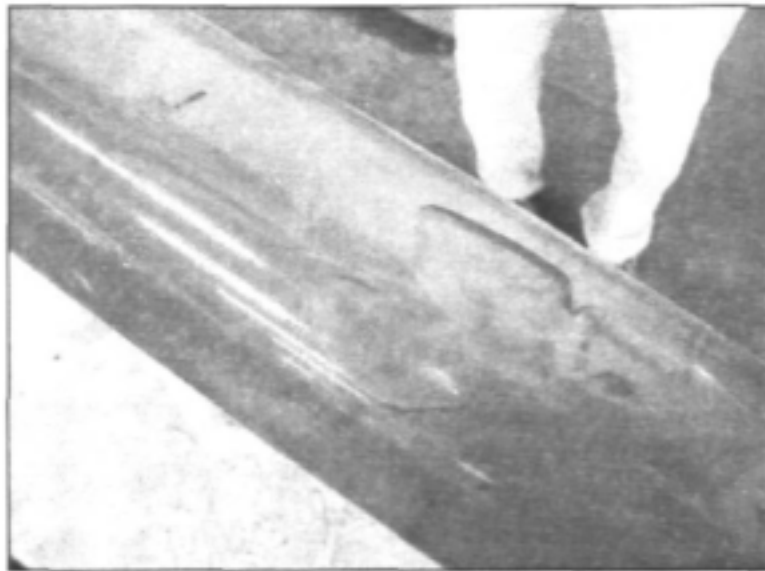


Figure 4.3.8: View of the small bubble (25 nml) used in the tests (110 mm diameter pipeline)

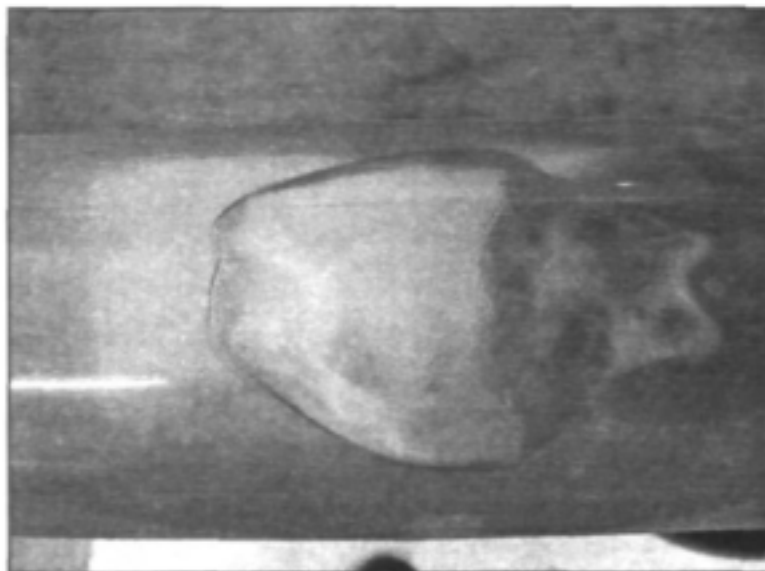


Figure 4.3.9: View of the medium bubble (75 nml) used in the tests (110 mm diameter pipeline)

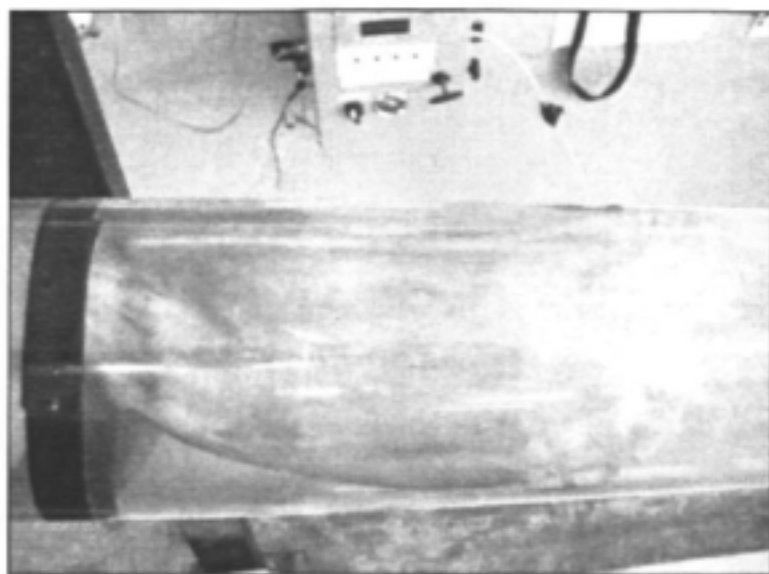


Figure 4.3.10: View of the large bubble (490 nml) used in the tests (110 mm diameter pipeline)

The transparent pipeline made it possible to track the movement of the air bubbles towards the Perspex air valve (manufactured by Vent-O-Mat), Detail C, **Figure 4.3.4** or to the end of the system, Detail D, **Figure 4.3.5**. The movement of air could thus be seen visually throughout the entire experimental set-up. Problems such as the development of slugs at the connections could be identified and excluded from the results.

At the end of the experimental set-up, a facility was created to capture all the air that had been transported past the off-take. This device consisted of a Perspex box with dimensions 0.5m x 0.5 m x 0.4 m, with calibration lines for measurements of the volume inside the box. These lines were drawn every 10 mm on the opposite sides of the Perspex box. On top of the box, a valve was placed to empty the box when it was filled with water.

The discontinuity was drilled into the transparent uPVC pipeline to enable the air to enter the air valve. The discontinuity was enlarged to allow more air to enter the air valve. The discontinuity was represented by the hole drilled in the pipeline for the experiment on the 110 mm ND pipeline (**Figure 4.3.11**).

The purpose of this alteration was to ensure that all the boundary effects that could restrict the release and capturing of air were incorporated in the experiment.



Figure 4.3.11: The discontinuity of 30% of the diameter in the pipeline

Through using the mass balance of the induced air and the air captured at the end of the line, the volume released through the air valve (discontinuity) could be determined.

4.3.3.2 Second experimental set-up

Similar to the first experimental set-up a 160 mm diameter pipe was also tested with various discontinuities to determine the discontinuity required to enable effective de-aeration, see the schematic drawing of the experimental layout in **Figure 4.3.12**.

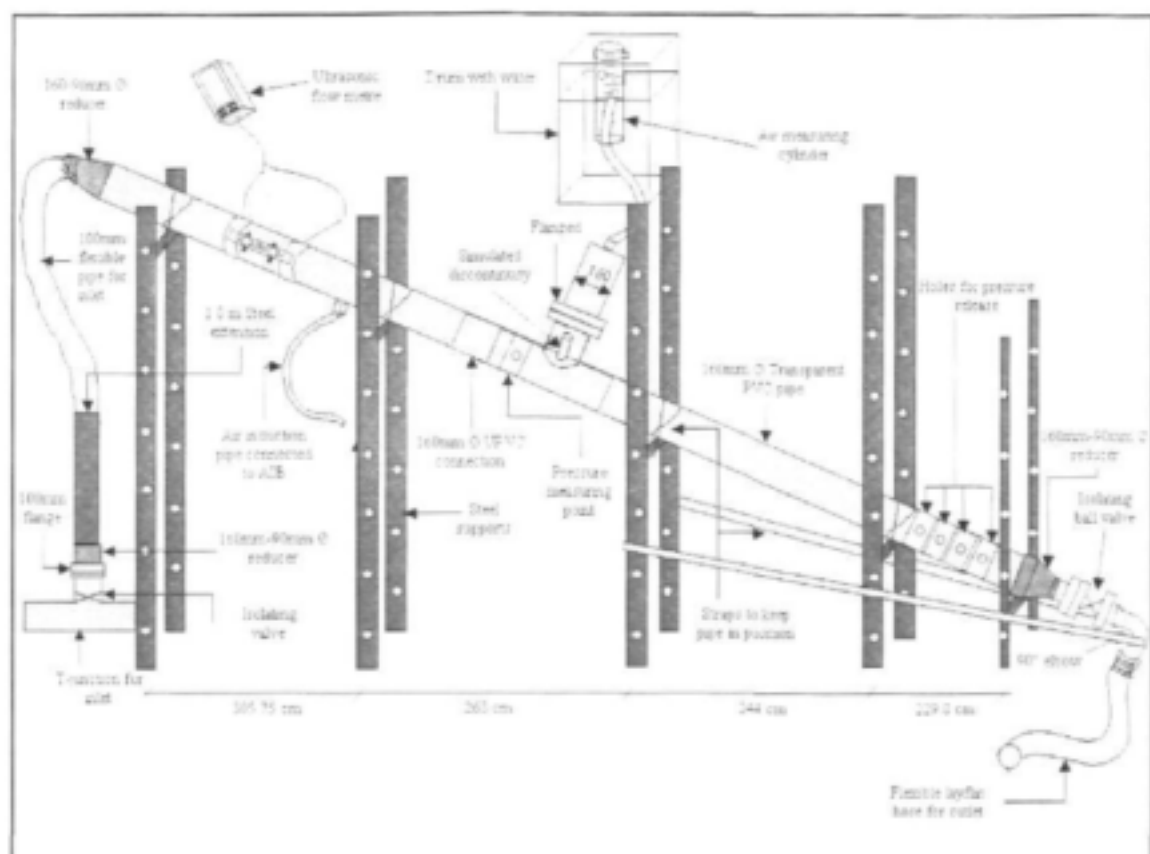


Figure 4.3.12: Schematic drawing of the experimental layout

Transparent PVC pipes (160 mm in diameter) were used in order to enable visual tracking of the air movement in the pipe as well as the capturing of air at the discontinuity. The sections of 160 mm diameter transparent PVC pipes were joined with standard PVC connectors. The pipeline supplying water was fitted with an accurate ultrasonic flow meter. The flow meter had to be placed as such that the undisturbed flow in the pipeline could be measured. This was accomplished by placing the flow meter before the air induction mechanism. The experimental set-up was also used for determining of the clearing velocity as described in Section 4.2.3.2 and therefore the description of the set-up is not repeated here.

Air induction mechanism

To control the volume of air in the pipeline, a computerized air regulator connected to a compressor was used to inject air into the pipeline (also see **Figure 4.2.6**).

Capturing mechanism

A discontinuity for capturing the air bubbles released into the pipeline by means of the air induction system at the beginning of the pipe was constructed. The discontinuity on the pipeline can be seen in **Figure 4.3.13**.

Two 160 mm diameter transparent PVC pipes were bolted to each other at the two flanges. **Figure 4.3.14.** A 10 mm thick layer of Jeka foam, acting as a seal, was placed in between the two flanges before bolting it together. This combined cylinder box was fixed with Perspex glue onto the transparent pipe and sealed with silicone for water tightness. At the bottom-enclosed end of the combined cylinder unit (enclosed by the 160 mm transparent PVC pipe as a part of the pipeline) a hole was made in the 160 mm transparent pipeline to create the discontinuity. The outer edge of the hole in the pipeline was extended into the combined cylinder unit by means of another smaller pipe for a length of about 20 centimetres. This extension serves to simulate the discontinuity, as it would be installed in the practice. The absence of such an extension indicated air bubbles entering the capturing device through the discontinuity much faster i.e. the discontinuity was functioning as an orifice. The opening size of the discontinuity varied between 20 mm and 70 mm. The top cylinder served as a measuring device for the pressurised air captured.



Figure 4.3.13: Discontinuity on pipeline with riser pipe (same diameter as hole)

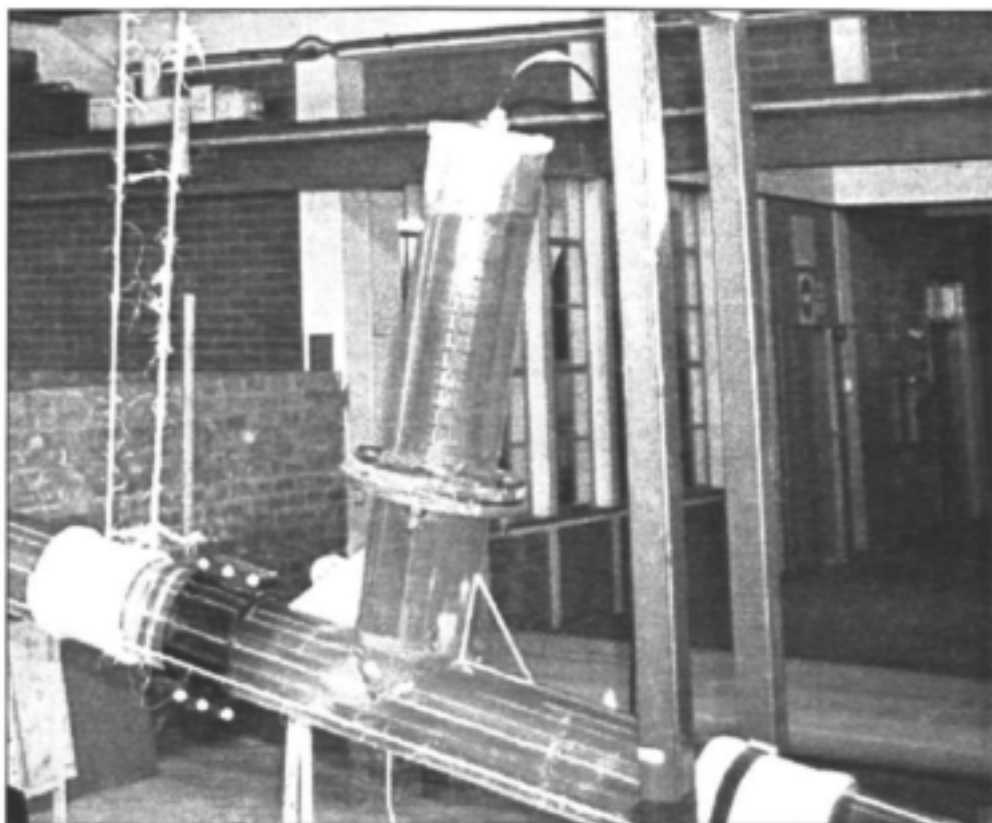


Figure 4.3.14: Discontinuity and capturing device on pipeline

The captured volume of air was checked by releasing the air and measuring it at atmospheric pressure in a separate cylinder, see **Figure 4.3.15**. For this measurement an accurate 2-liter cylinder, subdivided into 0.25-liter units, was placed in a drum filled with water. These measurements indicated the volume of air at atmospheric pressure. A threaded hole made in the top of the enclosed end of the combined cylinder allowed the controlled bleeding off of the captured air with a small diameter pipe. This configuration allows the volume of captured air under pressure to be measured in an upright cylinder at atmospheric pressure.

In order to convert the captured pressurized air to normal litres of air a pressure transducer was installed at the discontinuity, measuring the internal pressure in the pipe. The pressure transducer was mounted onto a saddle, see **Figure 4.3.15**, which was placed and tightened over the hole made in the pipeline.



Figure 4.3.15: The measurement of air captured at the discontinuity

The main changes between the second experimental set-up and the first were the following:

- The pipeline diameter was increased to 160 mm diameter
- In practice the discontinuity/off-take will have a certain diameter, which would be the same as the accumulator underneath the air valve. In the first experimental set-up on the 110 mm diameter pipe only a hole/orifice was made in the pipe. It seemed as if this configuration allowed air to easily enter the air valve chamber. This was also detected in the initial *Computational Fluid Dynamics (CFD)* numerical modeling that was performed. When air enters the air accumulator water has to be displaced back into the pipe. It was for this reason decided to determine what the effect would be if there was a riser pipe with similar diameter as the hole. This would be how it would be installed in practice i.e. the discontinuity size and the accumulator size would be the same.
- The experimental work on the 110 mm diameter pipe had another difficulty. By capturing the air that has gone past the discontinuity and then subtracting that volume from the introduced volume the efficiency could be determined, using a mass balance approach.

When the efficiency of the discontinuity was very high only a small quantity of air past the discontinuity and thus it took very long time for a sufficient volume of air to accumulate in the end measuring box. In the second experimental set-up the air captured through the discontinuity was rather measured to immediately determine the effectiveness of the discontinuity. This resulted in the testing time to be dramatically reduced.

4.3.4 Methodology

The Air Induction Box (AIB) was used to introduce air at a certain rate into the pipeline. The settings on the AIB that could be changed are the size of the air bubble, the number of air bubbles and the time step between the consecutive air bubbles. To establish the effectiveness of the discontinuity it was important to establish the volume of the air introduced into the experiment by the AIB. For each set of parameters and AIB setting, it was necessary to calibrate the AIB. The calibration was done by allowing a long enough time for the air to be introduced into the system through the AIB and by measuring the air volumes captured at the end of the system. Since the number of air bubbles introduced into the system by the AIB was known, the average size of the air bubbles for the specific set-up could be calculated.

A discontinuity of a specific size was created in the pipeline and the effectiveness was determined of the opening to capture the air that had been introduced by the AIB. For each of the calibrated air bubble sizes, different flow rates were investigated. The tests were repeated three times. After each set of tests, the discontinuity was enlarged (percentage of the pipeline's diameter) and the tests were repeated for a different pipe gradient.

For the first experimental set-up (110 mm diameter pipe), at the start of each test, the Perspex box was placed into the sump and filled with water (no air entrapped in the box). The tap in the lid of the box was left open until the pipeline had been cleared of all the free air.

Since the volume of air induced into the system is known from the calibration of the system, it is possible to determine the effectiveness of the discontinuity to capture the air moving down the pipeline. This was calculated by measuring the volume (in normal litres) of air trapped in the box.

In the second experimental set-up (160 mm diameter pipe), the captured air was measured directly.

Three different size bubbles were used for the testing. The three bubble sizes were defined as small, medium, and large. The size bubble, however, differed for the two experimental set-ups. The AIB was programmed to induce a certain size bubble by setting the differential pressure and the time the solenoid valve would remain open to allow a certain volume of air into the system. **Table 4.3.1** and **4.3.2** contains the details of the AIB settings for the different air bubbles that were tested.

Table 4.3.1: Bubble sizes (Photo's of all three bubble sizes used in the first experimental set-up (110 mm diameter pipe) are shown in Figures 4.3.8 to 4.3.10)

Description		Small	Medium	Large
Air Induction Box settings	Bubble size (ms)	25	75	80
	Pressure (Bar)	3	6	6
	Quantity	400	200	90
	Setting	Low	Low	High
	Regulator in use?	With	With	Without
Results	Calculated bubble size (nml)	25	75	490

Table 4.3.2: Bubble sizes (Used in the second experimental set-up (160 mm diameter pipe))

Description		Small	Medium	Large
Air Induction Box settings	Bubble size (ms)	25	75	80
	Pressure (Bar)	3.5	7	7
	Quantity	400	200	90
	Setting	Low	Low	High
	Regulator in use?	With	With	Without
Results	Calculated bubble size (nml)	61	221	547

During the tests for a specific air bubble size, the velocity, pipe gradients and the size of the discontinuities were varied (**Figure 4.3.16**). The first test was conducted for a pipe gradient of 0 degrees. The tests were then repeated for 2,5° to 15° in steps of 2,5°. For each of the slopes, the size of the discontinuity was varied from 5% to 35 % of the 110 mm pipeline's diameter in steps of 5%. The flow velocity investigated for each of the set-ups was 0.5 m/s, 1.0 m/s, 1.5 m/s and 2.0 m/s.

During the second experimental set-up the tests were then repeated for pipe gradients of 0° to 15° in steps of 2.5° . The flow velocities which were investigated were 0.5 m/s, 1.0 m/s and 1.5 m/s. For each of the slopes, the size of the discontinuity was changed from 20 mm to 70 mm i.e. 12.5 % and 44 % of the 160 mm pipeline's diameter.

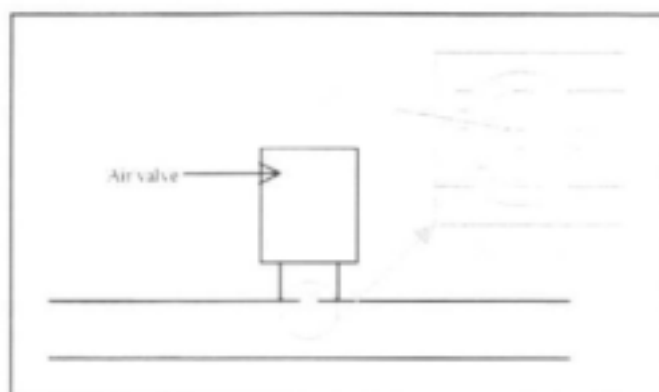


Figure 4.3.16: Schematic layout of the discontinuity

A difficulty experienced during the testing was the development of slugs at the pipe connections. This effect was minimized through forcing the pipes deeper into the connections, creating a smooth inner surface. This problem was more prominent at higher slopes and lower flow velocities.

4.3.5 Experimental results

4.3.5.1 Results of the tests that were conducted on the 110 mm ND pipeline

The results for the 110 mm and 160 mm ND pipelines are discussed separately. Firstly, attention is given to the results of the 110 mm ND pipeline. **Table's 4.3.2 to 4.3.8** (separate pages) give the average results for the three tests that were conducted on the 110 mm ND pipeline.

From these results, it became evident that the size of the discontinuity needs to be increased to capture the air effectively as pipe slope is increased. **Table 4.3.3** illustrates the efficiency of a 35 % discontinuity and a flow velocity of 1.0 m/s.

Table 4.3.3: Sample of the results obtained for the first experimental set-up with a 35% discontinuity and velocity of 1,0 m/s

Gradient	Effectiveness
0°	100,0%
2,5°	99,8%
5°	99,7%
7,5°	99,5%
10°	99,2%
12,5°	99,0%
15°	97,6%

The results indicated that the effectiveness of a specific opening is inversely proportional to the slope and the flow velocity. For high slopes and low velocities, the air bubbles tended to form a slug that moved upwards against the flow direction. The buoyancy forces on the air slug created this.

The results from the tests on the 110 mm ND pipeline suggested that a discontinuity of at least 35 % of the pipe diameter is required to reach an efficiency of more than 96 % for the tests conducted (slopes less than 15 ° and a flow velocity between 0,5 and 1,5 m/s).

The results of all the tests are graphically presented in the **Figures 4.3.17a to 4.3.21c** for the different size discontinuities, 5%, 10%, 25%, 30% and 35% respectively.

Figures 4.3.17a and Figure 4.3.17c to Figure 4.3.21a and Figure 4.3.21c all show that for a constant water velocity that the efficiency decreases as the gradient becomes less. Furthermore it is clear that for larger air bubbles the efficiency reduces. This can be attributed to higher turbulence created by the larger air bubbles and the fact that the bubbles are not always at the crown of the pipeline. It was also noticed that the induction of the large air bubble created a surge in the pipeline that pushed the bubbles away from the discontinuity, resulting in less air flowing towards the air valve.

At most gradients and a discontinuity of 35% of the pipeline's diameter, the discontinuity for all practical purposes is sufficient to de-aerate a pipeline completely. This result is also more effective for a velocity of 0,5 m/s but since no results could be obtained if the pipeline was raised for comparing the results, only 1.0 m/s could be used.

The bubble size also contributes to a large difference, since a small bubble can enter the discontinuity more easily than a larger bubble and therefore the small bubble at 0.5 m/s can very easily enter the air valve through the discontinuity.

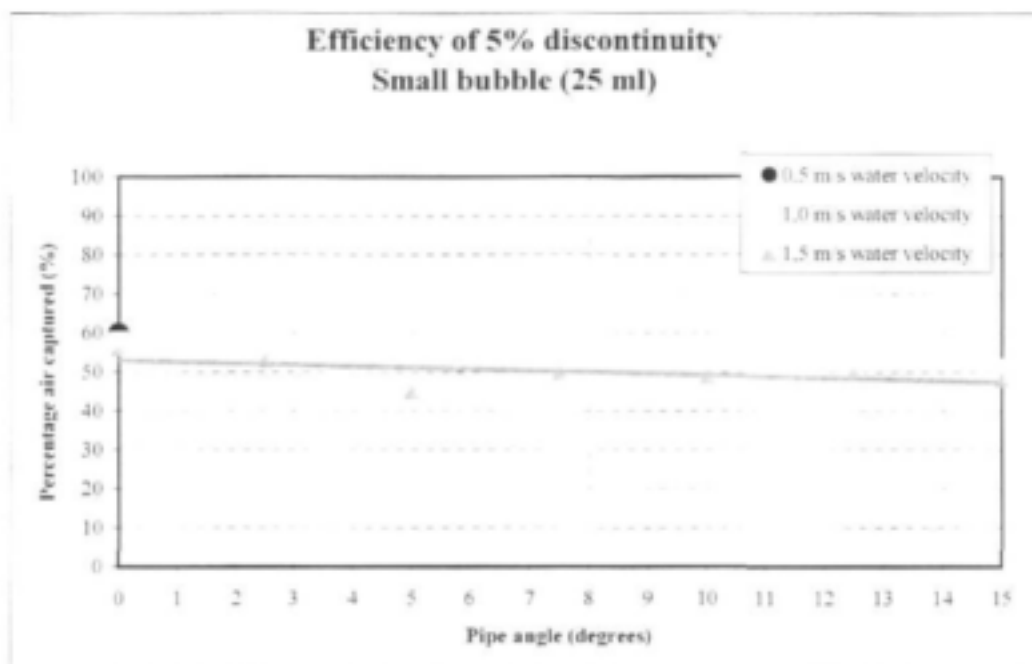


Figure 4.3.17a: 5% Discontinuity with small bubble (110 mm diameter pipe)

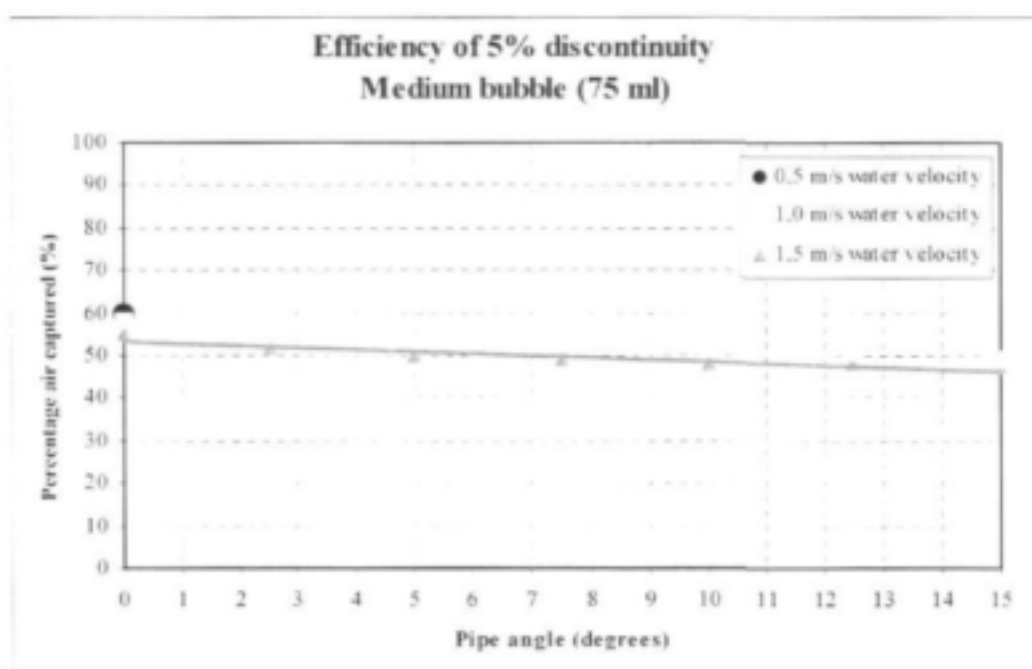


Figure 4.3.17b: 5% Discontinuity with medium bubble (110 mm diameter pipe)

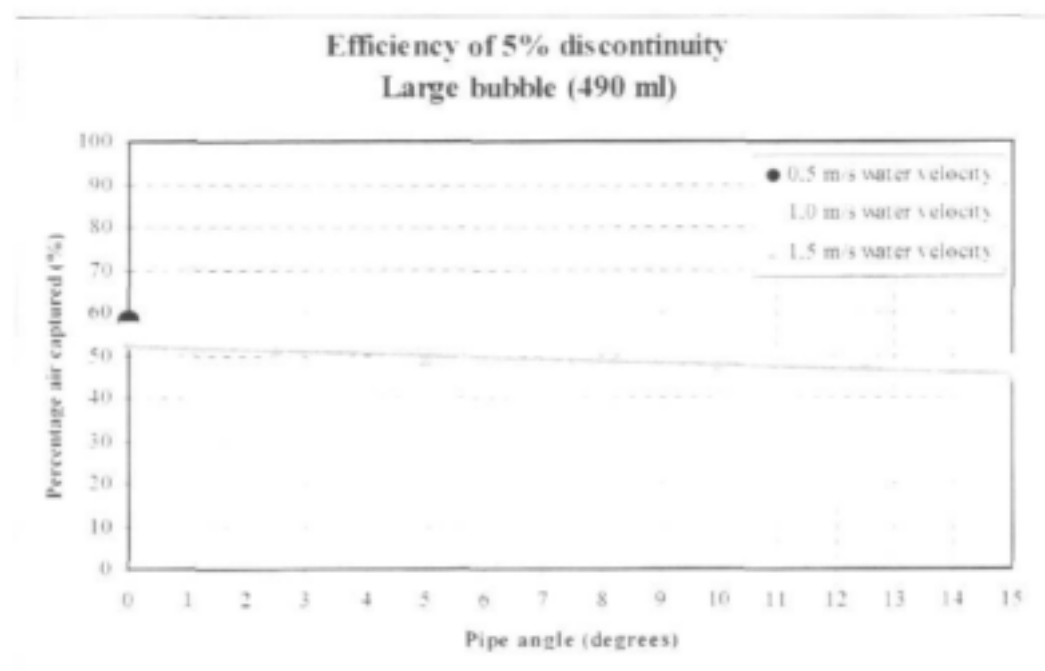


Figure 4.3.17c: 5% Discontinuity with large bubble (110 mm diameter pipe)

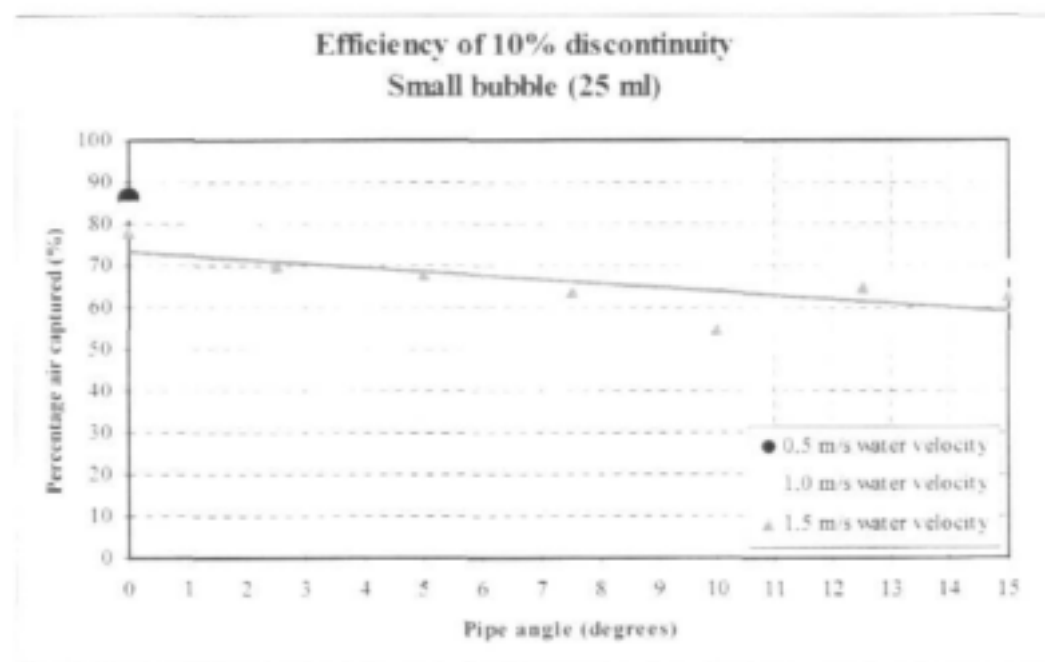


Figure 4.3.18a: 10% Discontinuity with small bubble (110 mm diameter pipe)

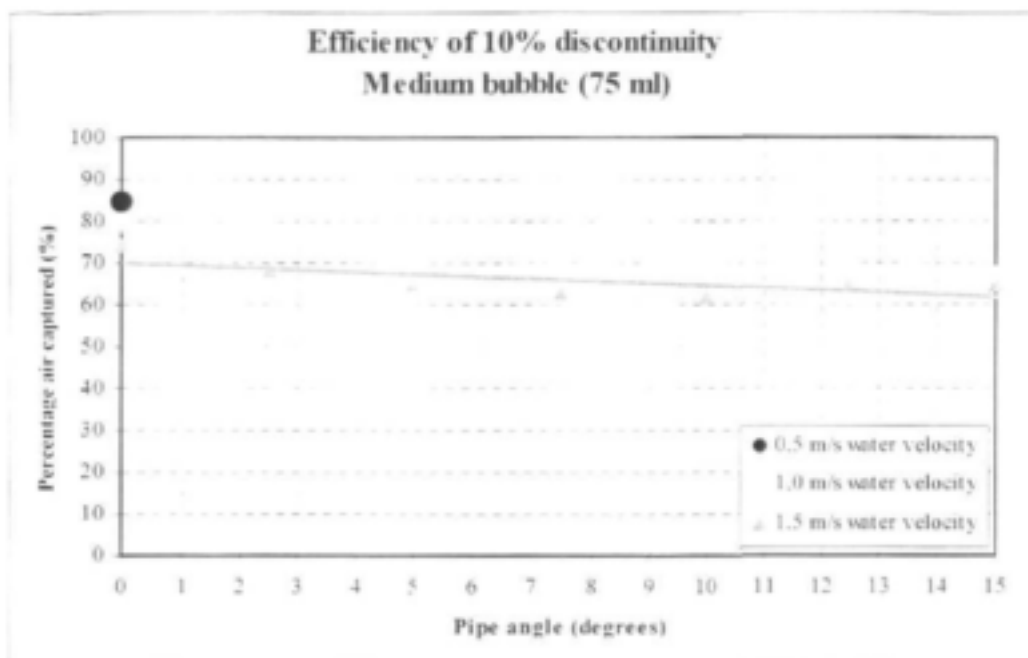


Figure 4.3.18b: 10% Discontinuity with medium bubble (110 mm diameter pipe)

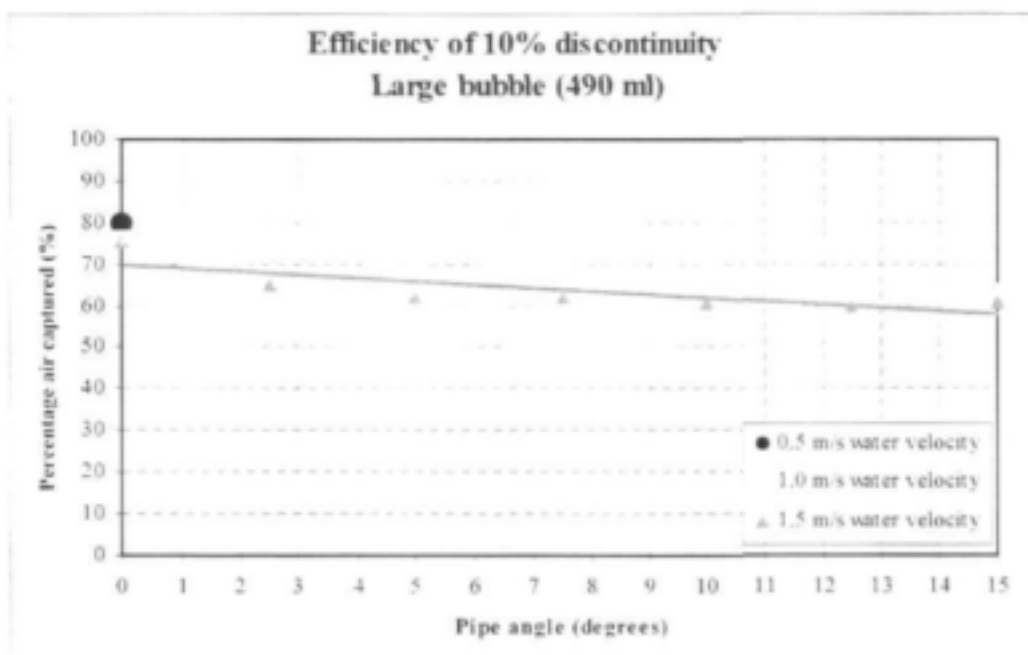


Figure 4.3.18c: 10% Discontinuity with large bubble (110 mm diameter pipe)

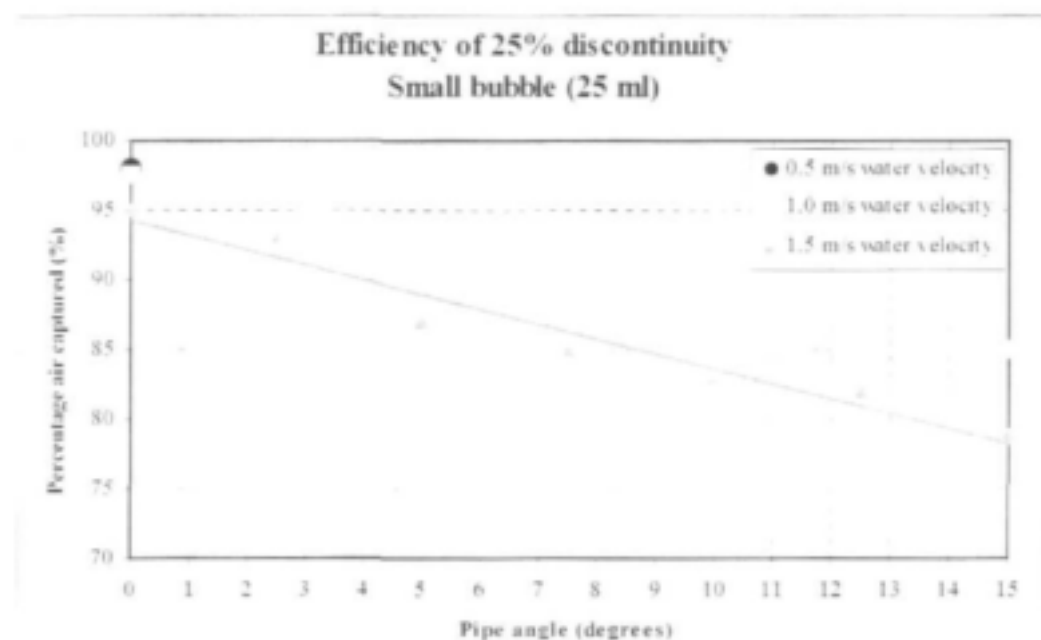


Figure 4.3.19a: 25% Discontinuity with small bubble (110 mm diameter pipe)

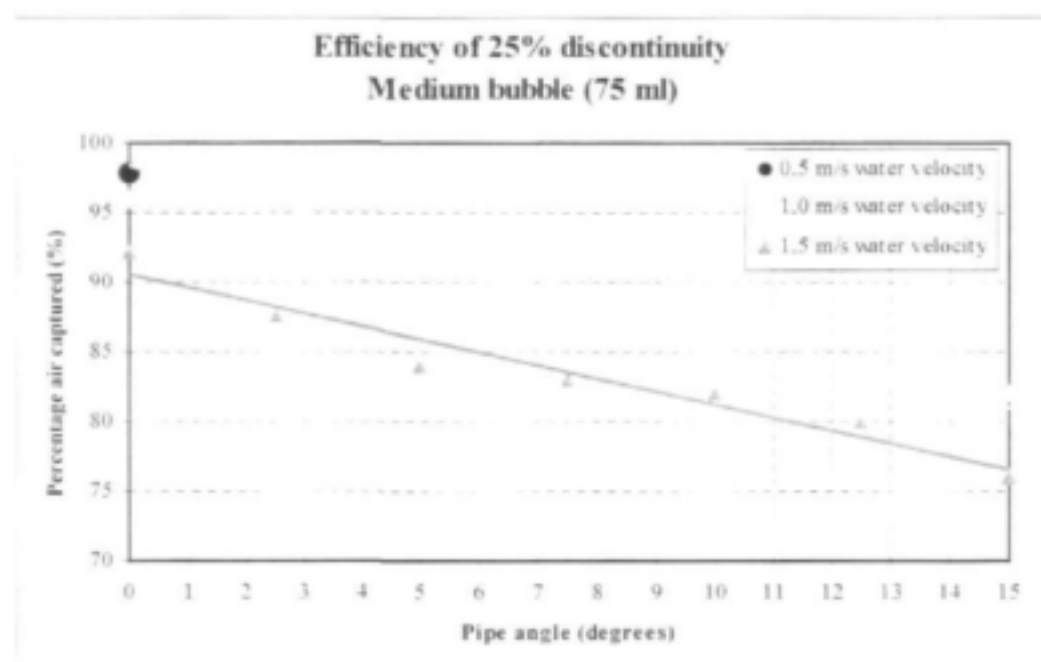


Figure 4.3.19b: 25% Discontinuity with medium bubble (110 mm diameter pipe)

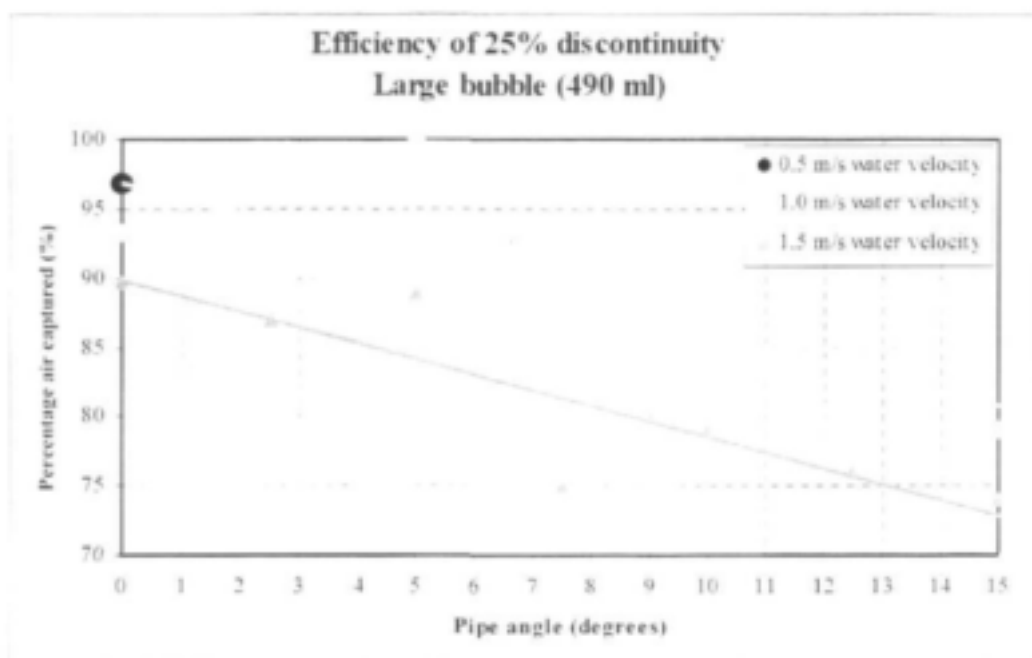


Figure 4.3.19c: 25% Discontinuity with large bubble (110 mm diameter pipe)

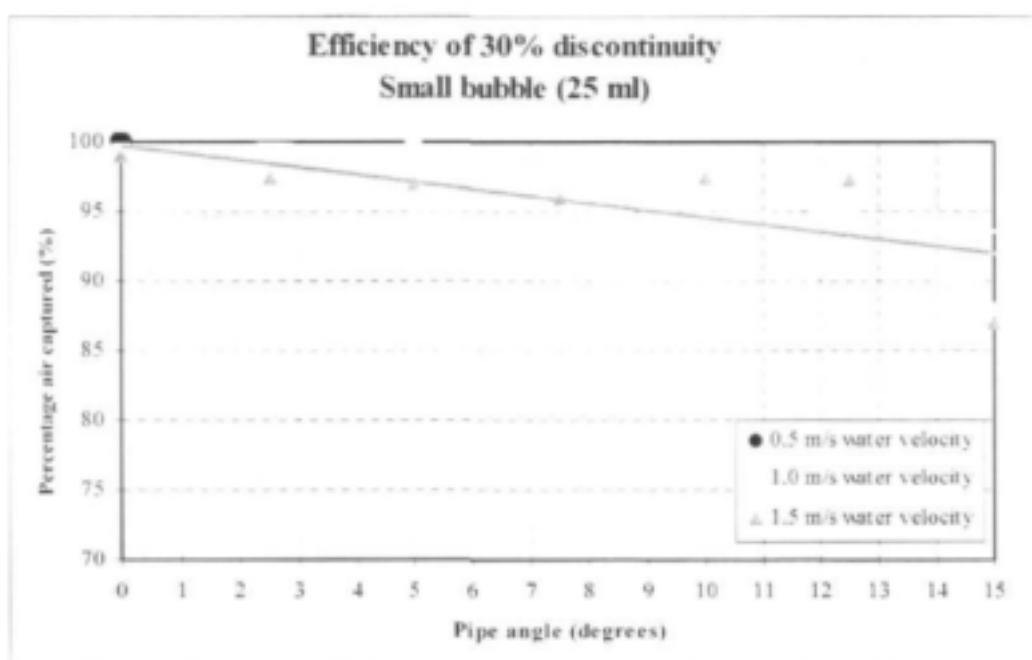


Figure 4.3.20a: 30% Discontinuity with small bubble (110 mm diameter pipe)

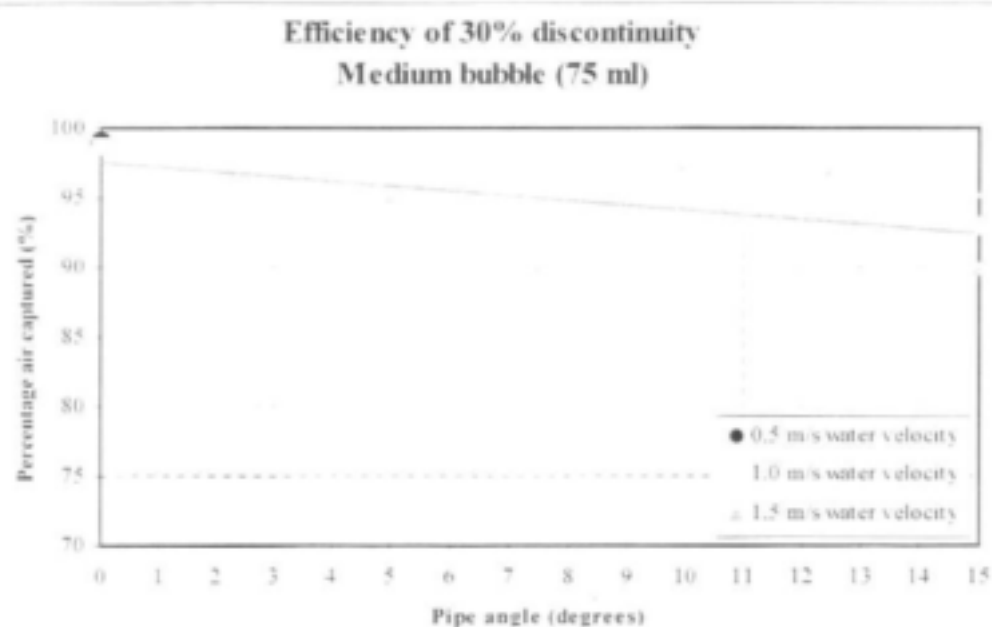


Figure 4.3.20b: 30% Discontinuity with medium bubble (110 mm diameter pipe)

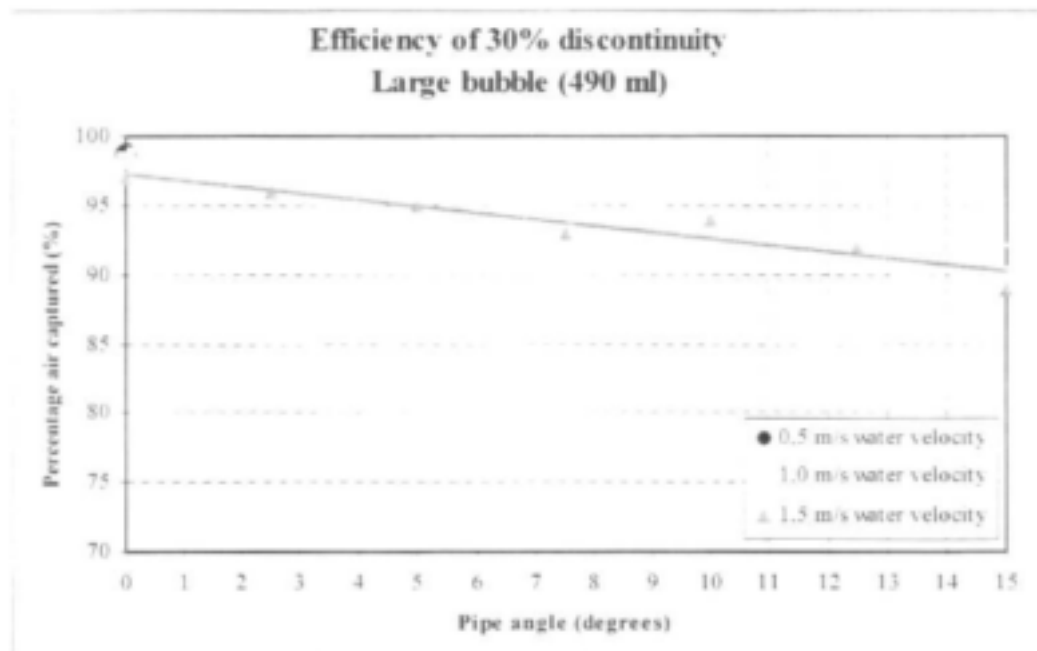


Figure 4.3.20c: 30% Discontinuity with large bubble (110 mm diameter pipe)

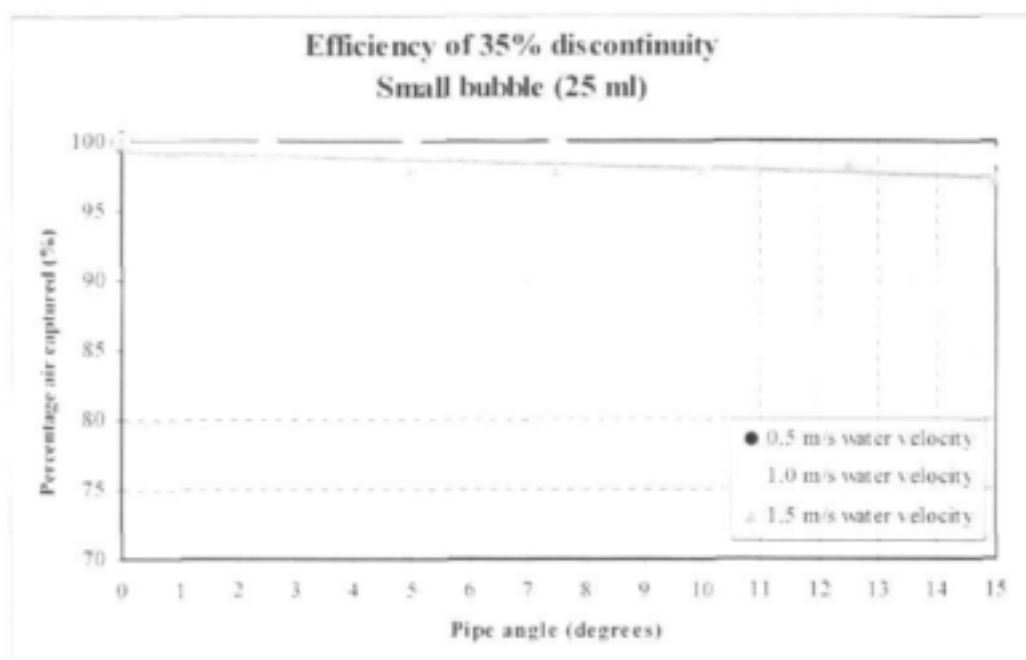


Figure 4.3.21a: 35% Discontinuity with small bubble (110 mm diameter pipe)

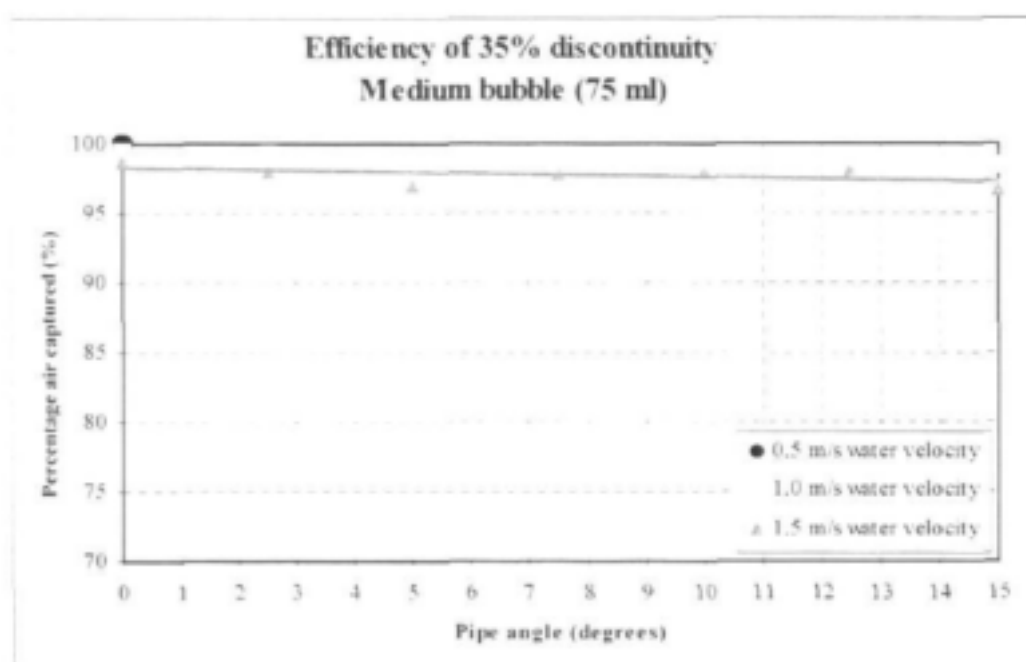


Figure 4.3.21b: 35% Discontinuity with medium bubble (110 mm diameter pipe)

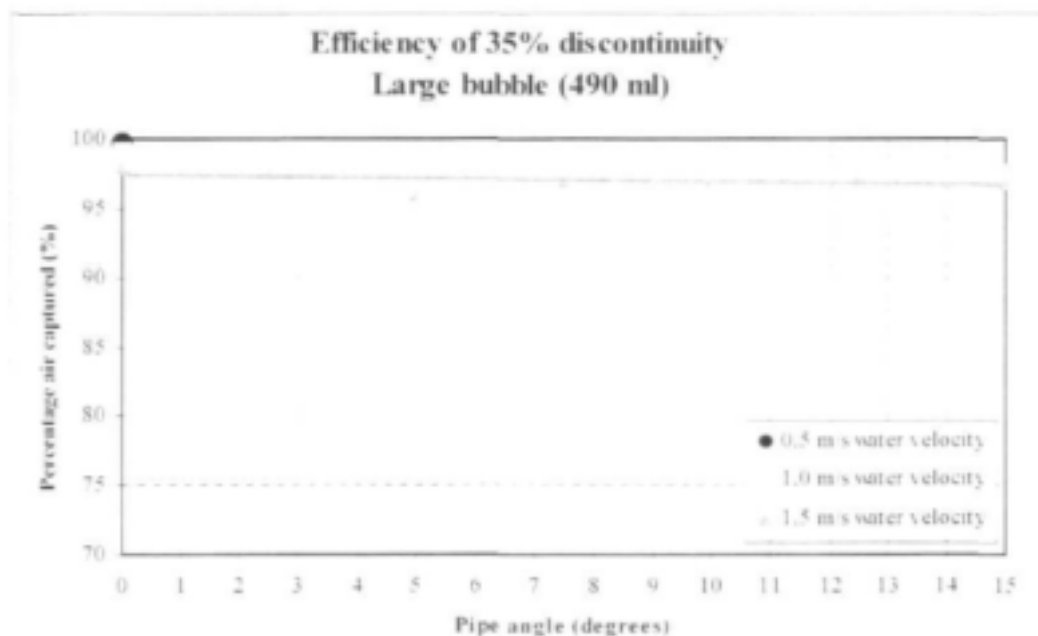


Figure 4.3.21c: 35% Discontinuity with large bubble (110 mm diameter pipe)

4.3.5.2 Results of the tests that were conducted on the 160 mm ND pipeline

In the tests, the discontinuity was modeled as a “standpipe” off take that more closely relate to the stub underneath an air valve for normal operation. This was done because for the tests conducted on the 110 mm diameter pipe the air was captured at a higher rate than the simulated results from the CFD modeling. A 20 mm diameter cylindrical discontinuity, mounted on the 160 mm transparent pipe, was used to capture the air that was induced into the system at different pipe angles (from 0° – 15°) and water velocities (0.5 m/s to 1.5 m/s). The volume of captured air was measured in order to determine the amount of air mechanically removed from the system by means of the 20 mm diameter (12.5%) discontinuity. Similar tests were executed on a 70 mm diameter (44 %) discontinuity.

Figures 4.3.22, 4.3.23 and 4.3.24 show the percentage of air captured at the 20 mm discontinuity at different angles for a small, medium and a large bubble respectively. From these graphs it could be concluded that a higher water velocity leads to a lower percentage of air being captured by the discontinuity. The bubbles move faster past the point of the discontinuity due to the higher water velocity. A low water velocity allows the bubble more time to move into the discontinuity and displacing the water. At gradients steeper than 2.5° a velocity of 0.5 m/s was too low to transport the medium and large air bubbles (Figure 4.3.23 and 4.3.24). At gradients steeper than 5° a velocity of 0.5 m/s was too low to transport small air bubbles (Figure 4.3.22) and hence the graphs will reflect a 0% air captured.

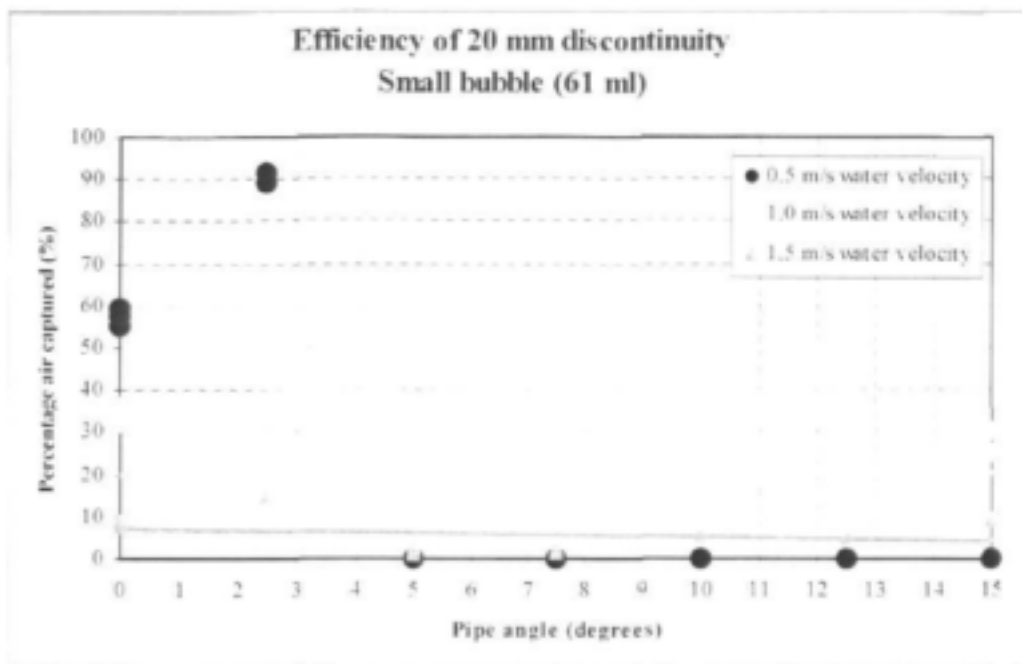


Figure 4.3.22: Efficiency of 20 mm diameter (12,5%) discontinuity with a small bubble (160 mm diameter pipe)

Figure 4.3.22 shows that the 20 mm diameter discontinuity captures on average about 20 % of the small (61 ml) air bubbles at a water velocity of 1 m/s and only 5 % at a velocity of 1,5 m/s. At a velocity of 0,5 m/s, 60 % – 90 % of the bubbles (induced at pipe angles of 2,5° and lower) were captured by the discontinuity. Small bubbles induced at steeper angles could not be transported hydraulically down the system.

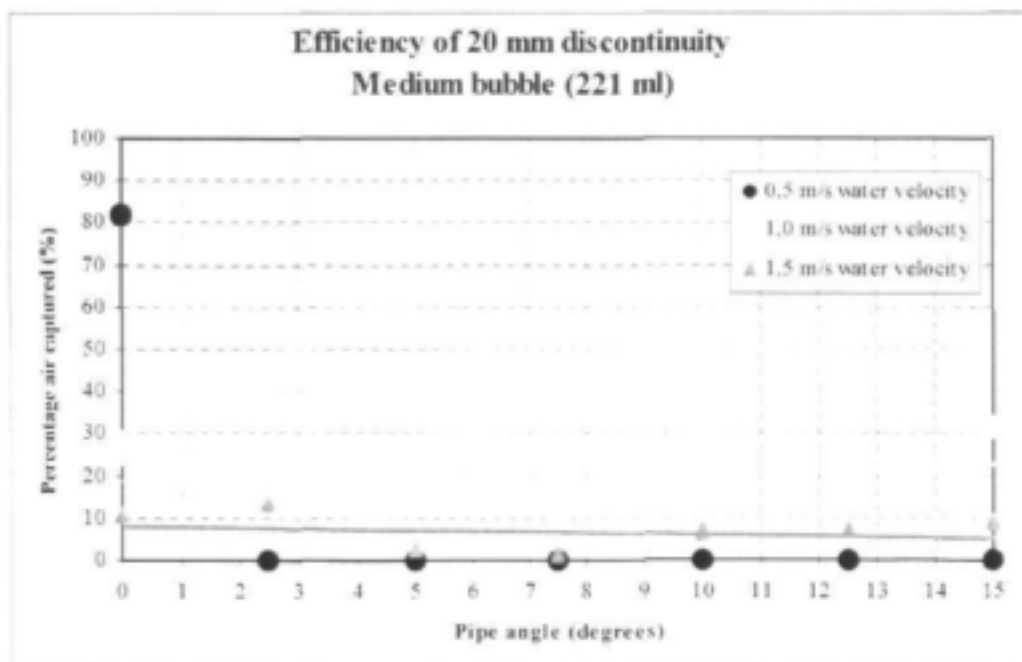


Figure 4.3.23: Efficiency of 20 mm diameter (12,5%) discontinuity with a medium bubble (160 mm diameter pipe)

At a velocity of 1 m/s only 8 % - 35 % of the medium (221 ml) bubbles (**Figure 4.3.23**), induced into the system, were captured. A mere average of 8 % were captured at 1.5 m/s water velocities. For water velocities of 0.5 m/s the medium bubbles could only be transported down the system when the pipe was in a horizontal position. For this case 80 % of the accumulated bubble volume induced into the system was captured by the 20 mm discontinuity.

The large (547 ml) bubble (**Figure 4.3.24**) showed very much the same statistics as that of the medium bubble in the mechanical removal of air by means of the 20 mm discontinuity at water velocities of 0.5 m/s and 1.5 m/s. At a water velocity of 1 m/s, 7 % - 35 % of the induced bubble volume was captured by the discontinuity.

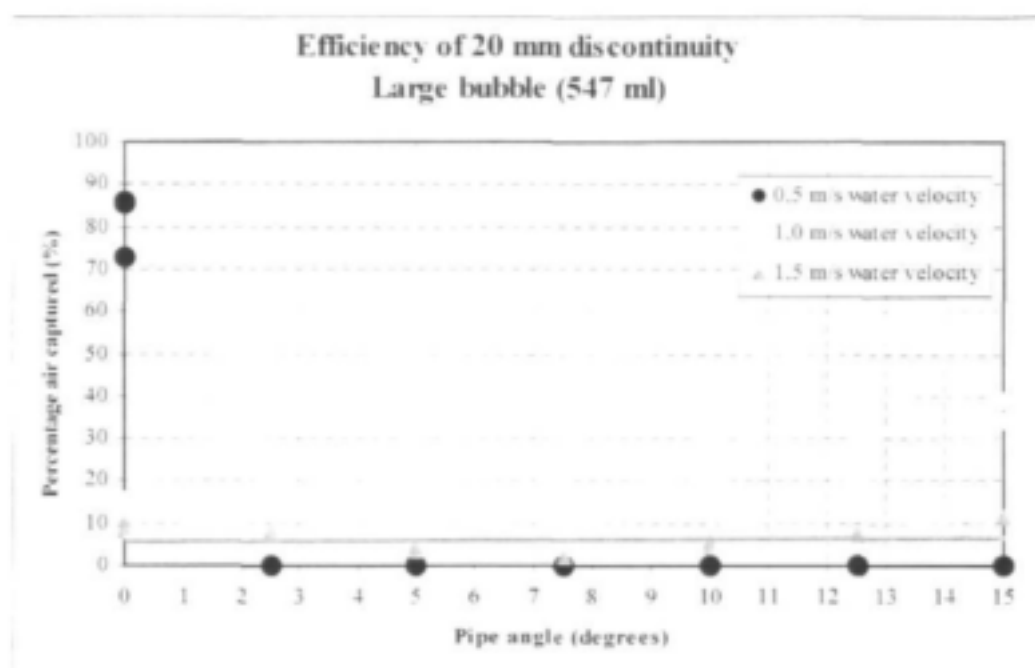


Figure 4.3.24: Efficiency of 20 mm diameter (12,5%) discontinuity with a large bubble (160 mm diameter pipe)

The 20 mm diameter discontinuity (12.5 % of the pipe diameter) captured roughly 30 % of the bubble volumes induced into the system. It would be desirable to have a size discontinuity that captures about 90 % of the induced bubble volume. The 20 mm size for the discontinuity was thus not sufficient for the mechanical removal of air from the pipeline.

The 20 mm diameter size discontinuity was enlarged to 70 mm in order to find the smallest discontinuity required to effectively de-aerate the system.

Figures 4.3.25, 4.3.26 and 4.3.27 shows the graphical presentation of the data for the percentages of air captured at varying gradients with linear lines as best curve fit through the data.

Figure 4.3.25 shows that the small bubble (61 ml) will not be transported down the system for gradients steeper than 5° at a velocity of 0.5 m/s. For the same velocity (0.5 m/s) the medium bubble (221 ml) will only be transported down the system when the pipe is in a horizontal position. A large bubble (547 ml) will not be transported at any angle for a water velocity in the pipe as low as 0.5 m/s. At water velocities of 0.5 m/s, at the positions where the small and medium bubbles were transported down the system, 100% of the bubble volumes were captured in each case.

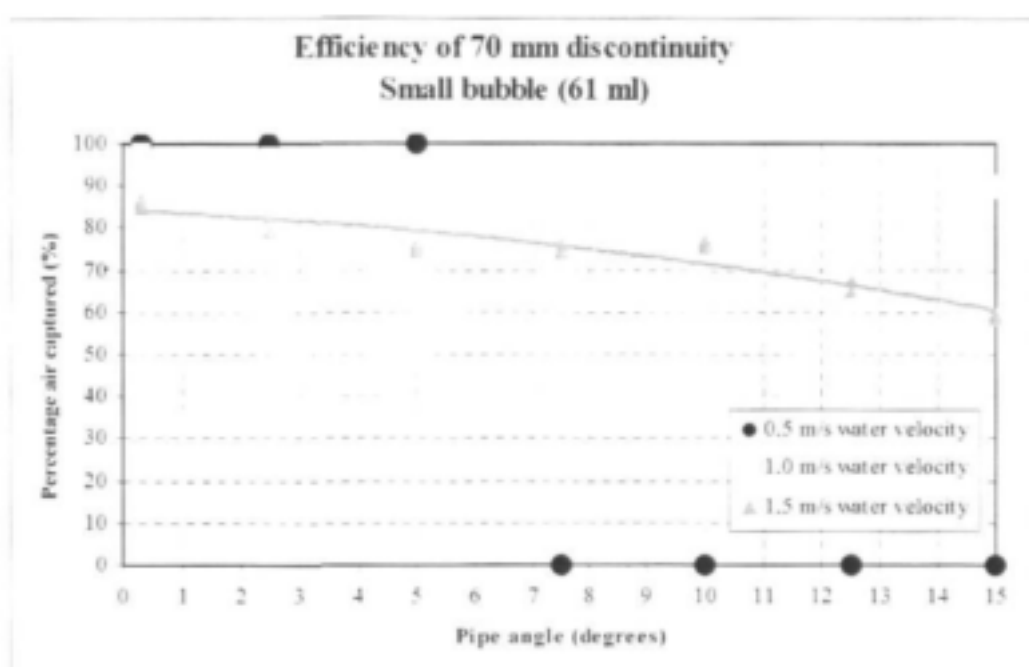


Figure 4.3.25: Efficiency of 70 mm diameter (44%) discontinuity with a small bubble (160 mm diameter pipe)

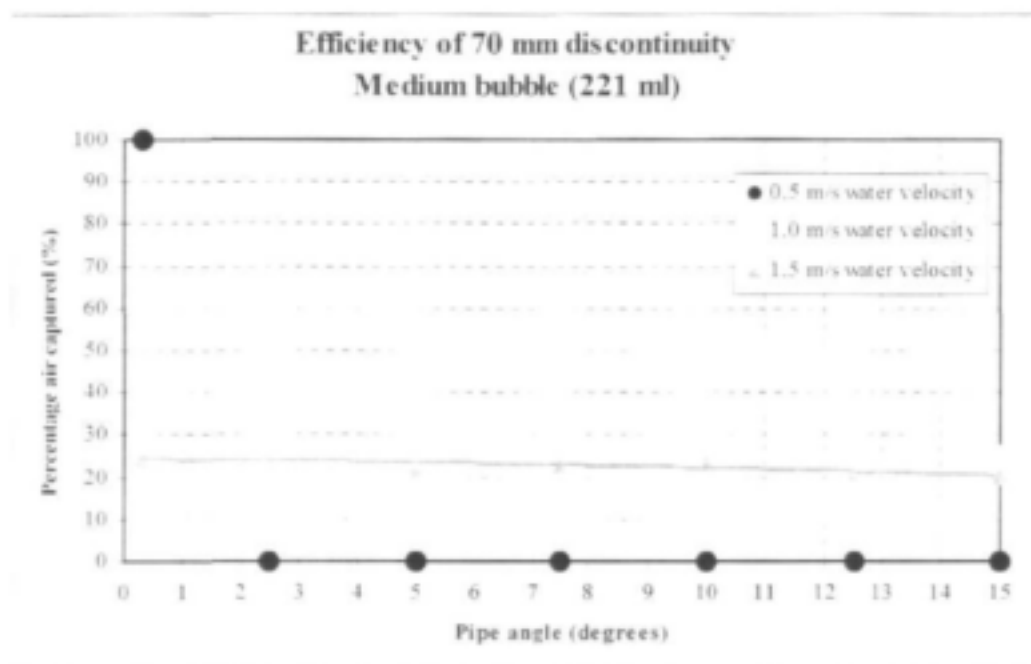


Figure 4.3.26: Efficiency of 70 mm diameter (44%) discontinuity with a medium bubble (160 mm diameter pipe)

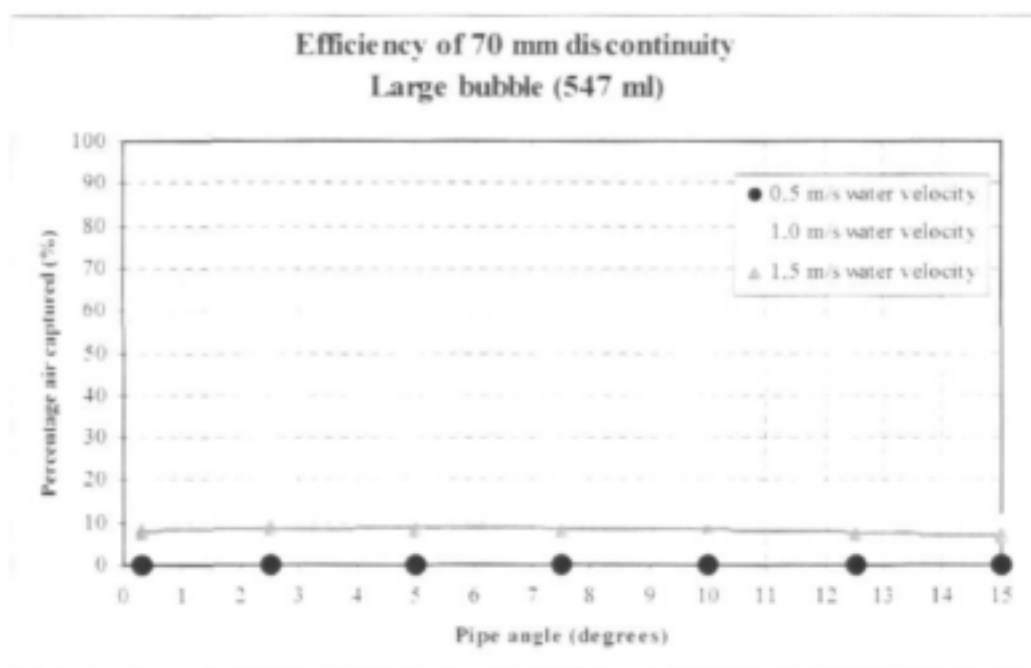


Figure 4.3.27: Efficiency of 70 mm diameter (44%) discontinuity with a large bubble (160 mm diameter pipe)

For the small bubble of 61 ml (Figure 4.3.25), 60 % - 85 % of the bubble volume was captured at 1.5 m/s and 90 % - 95 % of the bubble volume at 1 m/s. For the 221 ml medium bubble (Figure 4.3.26), 20 % - 25 % at 1.5 m/s and 25 % - 30 % at 1 m/s was captured.

An average of only about 8 % of the bubble volume for the 547 ml large bubble (Figure 4.3.27) was captured at 1.5 m/s and about 9 % – 12 % at 1 m/s. The percentage air captured by the discontinuity increases with a decrease in water velocity for reasons as discussed with the 20 mm diameter discontinuity.

These results show that the 70 mm diameter discontinuity (44 % of the pipe diameter) was nearly sufficient for the small bubble, but not for the medium and large bubbles.

4.3.6 Conclusions

Based on the results from the tests conducted on the 110 mm ND pipeline, a 35% discontinuity will be sufficient for any gradient of a pipeline and any velocities up to 1.5 m/s. The tests on the 160 mm pipe, however, indicated that even a 44% diameter would not be sufficient. The main reason for this discrepancy is the resistance in the riser pipe, which was added to the second experimental setup.

The air capturing could be divided into two components:

- Firstly, the discontinuity area should be sufficient to prevent any air bubbles from slipping past the side. This is especially true for large diameter pipelines.
- Secondly, the discontinuity riser should be large enough to allow for air entry and the displacement of the water in the riser. In the experimental work carried out it became evident that for the selected bubble sizes used in the experimental set-ups the discontinuity to capture all the air effectively needs to be more than 45 %.

Typical bubbles in practice would probably be less than 100 ml (normal milliliters).

It is recommended that for pipes larger than 350 mm the criteria of 35% of the diameter be used. Pipes equal to or smaller than 350 mm it is recommended that equal Tees be used. This is for all the positions where the small orifice function is required and the free air should be captured.

It should be noted that all these tests on the various discontinuities were executed at negative pipe slopes only. Attention should be paid to the fact that the bubble velocity would be much higher in the case for positive pipe angles. A higher bubble velocity would result in a lower percentage air captured. The bubble passes the discontinuity quicker and there is less time for the bubble to enter the discontinuity. A larger size discontinuity would thus be required at positive pipe angles.

Air does not always travel at the top of the pipe. If there were a disturbance upstream that creates turbulence the air would cause the air to be mixed through the entire cross section. The air capturing will in this case be less efficient since the air bubbles will not have the opportunity to enter the discontinuity. It is recommended that the air valve should be moved further downstream or just upstream of the disturbance. It is however more cost effective to place air valves in the same chamber as for instance an isolating valve or bend. This, however, might not be the ideal position for capturing free air. The location might however require an air valve for a different function (large orifice function).

4.4 Inverse transients to detect leaks and air bubbles in pipelines

4.4.1 Introduction

Although it is normally accepted that hydraulic transients can be detrimental in pipelines, the phenomena can also be used to determine some of the physical properties of the pipeline. Instead of creating a harmful positive pressure wave, a negative transient can be generated for the purpose of investigating the pipeline characteristics. These so-called inverse transients can also be used to determine some of the hydraulic properties of the system. A transient wave propagating through a pipeline is affected by irregularities and boundary conditions in the pipeline. Common characteristics in a pipeline that will influence the propagation of transient waves are: diameter variations, demands from of supplies to the pipeline, cross connections and dead ends. In the case of a localised air bubble, it is known that it will reflect and transmit a lower transient wave than the approaching wave. An air pocket will also influence the wave celerity.

It can therefore be postulated that inverse transient can be used to determine the presence of irregularities in the pipeline, if the measured wave propagations are different from the calculated wave propagation. The extent of the variation in the wave propagations is proportional to the severity of the irregularities. Jönsson (1999) demonstrated the theory when he showed that the existence of an air pocket, a leak and a malfunctioning check valve could be deduced from the pressure fluctuations of a hydraulic transient.

The basic concept of leak detection, using inverse transients in pipelines is quite straightforward. Transients can be introduced in a pipeline and the propagation recorded by capturing the pressures at various positions along the pipeline. If the characteristics of the induced transient pressure are known (or measured), it can be used to calculate the wave propagation throughout the system. These calculated results could then be compared with the recorded data to determine the possible characteristics of the system that are responsible for the discrepancies in the calculated and recorded pressure-time relationship.

Figure 4.4.1 represents a typical pressure-time plot at a point between the origin of a transient and an irregularity in the pipeline. It reflects that it might be possible to determine the location of the irregularity, if the time can be measured for the reflection of the wave from the irregularity to reach the recording point.

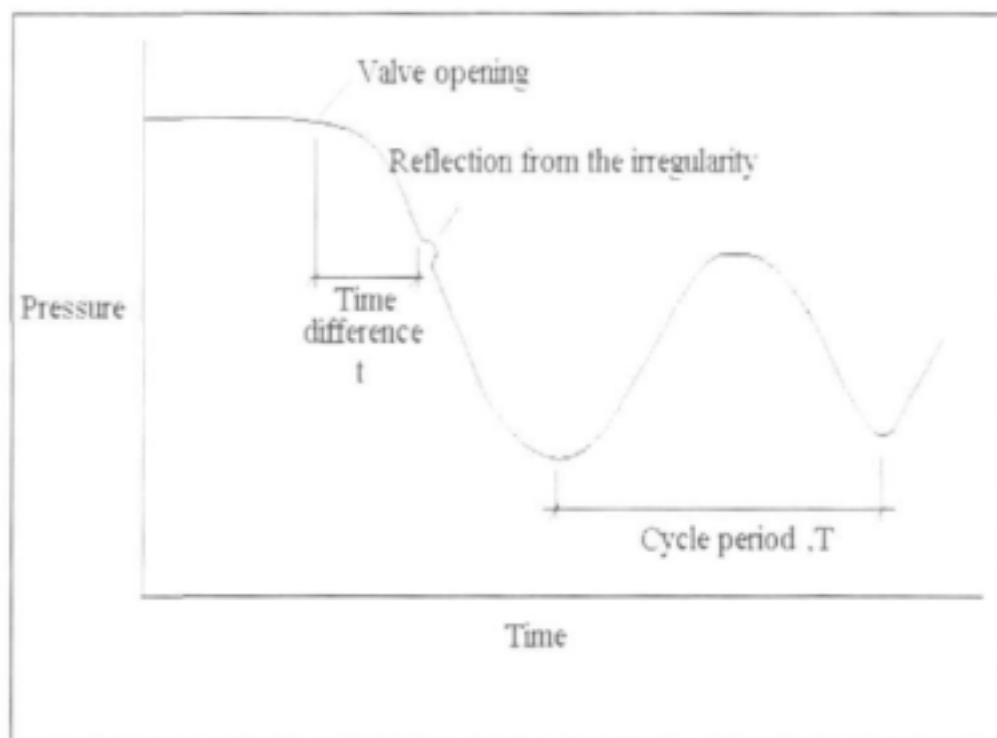


Figure 4.4.1: Pressure variation at measuring point due to the reflected wave

If it is assumed that the distance between the irregularity and the monitoring point is an unknown distance “s” apart, the celerity of the system is known and the time, “t”, which represents the time step between the time the wave reached the measuring point and the time when the reflecting wave reached the measuring point, can be measured, it is possible to determine the position of the irregularity in the system, relative to the position of the measuring point.

On the pressure-time graph, there will be a slight variation in the pressure resulting from the influence of the reflecting wave that reaches the measuring point. By determining the time step “t” between the drop in pressure at the measuring point and the time when the variation from the reflecting wave is recorded, it is possible to estimate the location of the leakage by applying Equation 4.4.1.

$$c = \frac{2s}{t} \dots\dots\dots (4.4.1)$$

Where:

- c = Wave celerity (m/s)
- s = Distance between the irregularity and the measuring point (m)
- t = Time step to experience the reflecting wave at the measuring point (s)

In practice, however, most pipeline systems are much more complex than the simple system discussed so far. Practical water distribution or conveyance systems normally consist of a number of pipes, valves, cross connections and diameter changes, all of which will create reflecting waves when a transient wave is introduced at a specific point in the system. Tracking the movement of the waves is required to determine the location and extend of the irregularities in the system. Any indefinable change in the wave propagations through the system reflects irregularities in the system. Positioning of the location and determination of the extent are complex: hence the uses of search techniques such as genetic algorithms (Tang, Brunone and Karney (2000)) are employed to pinpoint the irregularities.

Two common irregularities in water systems are leaks and localised air bubbles. Within the context of this research these two phenomena were evaluated in an experimental set-up.

4.4.2 Objectives of the experimental work

The objective of the experimental set-up was:

- To simulate a leak of varying flow rates in a pipeline and determine the influence on the wave propagation.
- To determine the influence of a localised air bubble on wave propagation.

The experimental set-ups for both these tests are very similar and hence the description of the experimental set-up, which has been used for both these tests, will be discussed first, followed by a separate discussion of the results for the tests. During the first experimental set-up some problems were experienced with the localised air bubble testing. This necessitated the construction of a second experimental set-up specifically for the localised air bubble.

4.4.3 Experimental set-up for the determination of the influence of leaks and air bubbles on the propagation of transient waves

4.4.3.1 First experimental set-up

The experimental model was set-up on the Experimental Farm, University of Pretoria, and it consisted of a pump, a connection pipe to the experimental pipe that feeds the flow into a channel and back into the pump sump (**Figure 4.4.2** provides a schematic layout of the system).

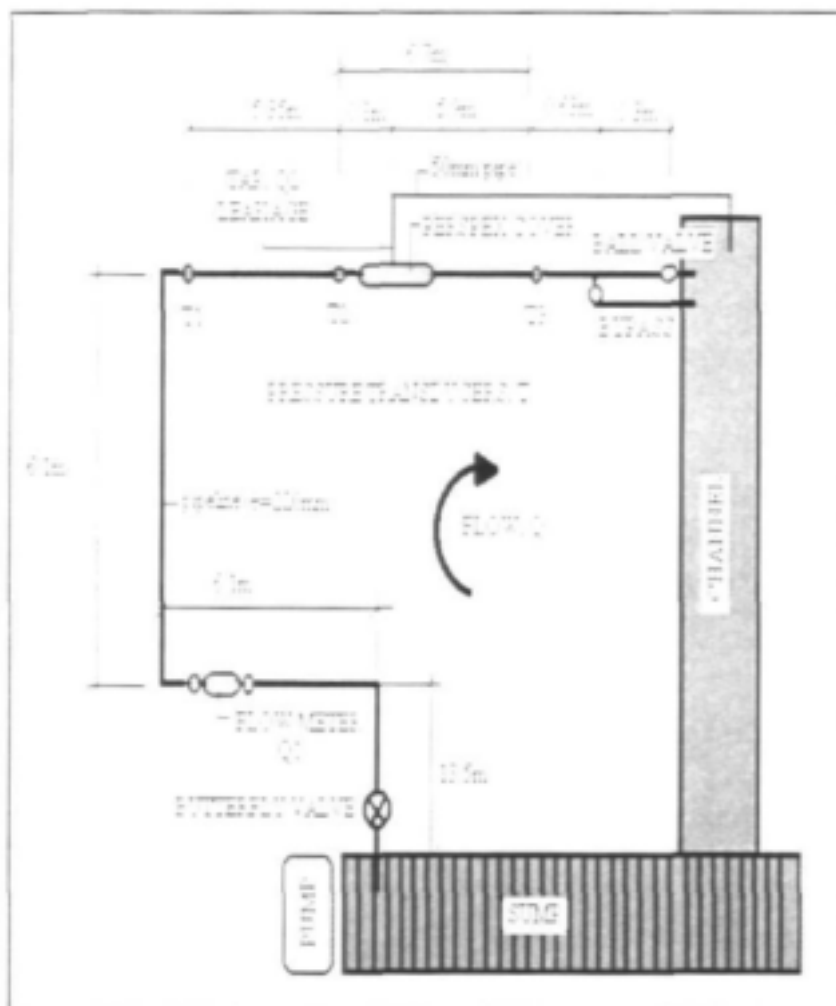


Figure 4.4.2: Layout of set-up for the investigation of inverse transients to detect irregularities in a pipe system

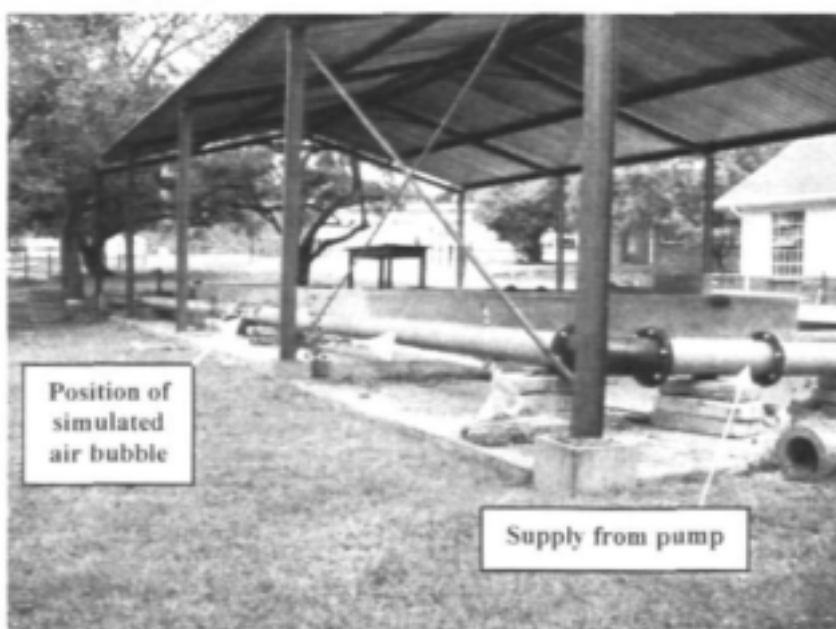


Figure 4.4.3: View of the pipe rig and position of the air bubble

The entire pipe rig was assembled under a shed at the experimental farm of the University of Pretoria. In **Figure 4.4.3** some of the 220 mm diameter (OD) piping can be seen in the foreground and passing into the picture and along the background. The simulated air bubble (**Figure 4.4.5**) and valves for controlling the flow are situated at the far end and are not visible.

Figure 4.4.3 provides a view of the entire length of the 220 mm diameter test section (the flow direction is indicated by an arrow). Concrete supports to the underside of the pipe are shown. The final level adjustment of the pipe is by wooden wedges. The length of pipe, from the 90° bend in the foreground to its end is 13.5 m.

A ball valve was placed at the end of the pipeline (**Figure 4.4.4**) to generate the transient wave, whilst a butterfly valve was installed just downstream from the pump to regulate the flow and pressure. A bypass was placed just in front of the ball valve to create different initial pressures and flow rates for the experiments (**Figure 4.4.4**). The flow from the ball valve and the bypass fed into the channel allowed flow back into the pump sump.



Figure 4.4.4: Ball valve to create the negative surge wave

To create a localised air bubble, the top section of the pipe was cut away and was covered with a Perspex lid (**Figure 4.4.5**). On the inside, wire mesh was welded to the pipe. The Perspex lid was held in place with steel clamps. A layer of latex was put onto the mesh and kept in position by the Perspex lid. An air inlet valve (**Figure 4.4.5**) was fixed to the top of the Perspex cover to allow the bubble to be pressurised and also to release the air when it was required to measure the volume of the air bubble. Air could be manually pumped into the pipe with a compressor and the latex and wire mesh would then function as one of the boundaries of the air bubble (**Figure 4.4.7**). A small off take (tap) (**Figure 4.4.6**), used to simulate a leak, was installed on the side of the pipe just below the Perspex cover. With the tap, the flow rate and thus the size of the leak could be varied. Due to some difficulties to control the size of the air bubble a second experimental layout was modelled where a T-piece was used in which the air bubble was localised. This is described in more detail in paragraph 4.4.3.2.

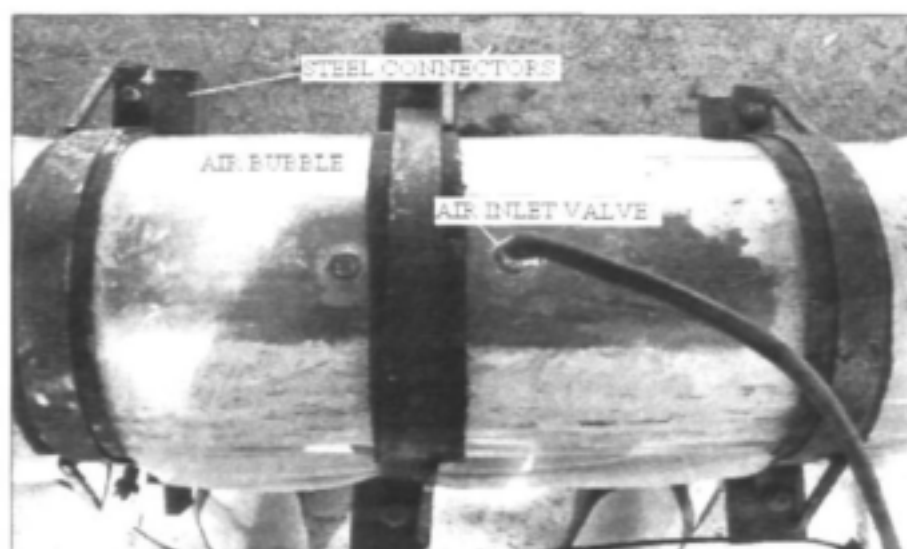


Figure 4.4.5: Air inlet valve, steel connectors and air bubble

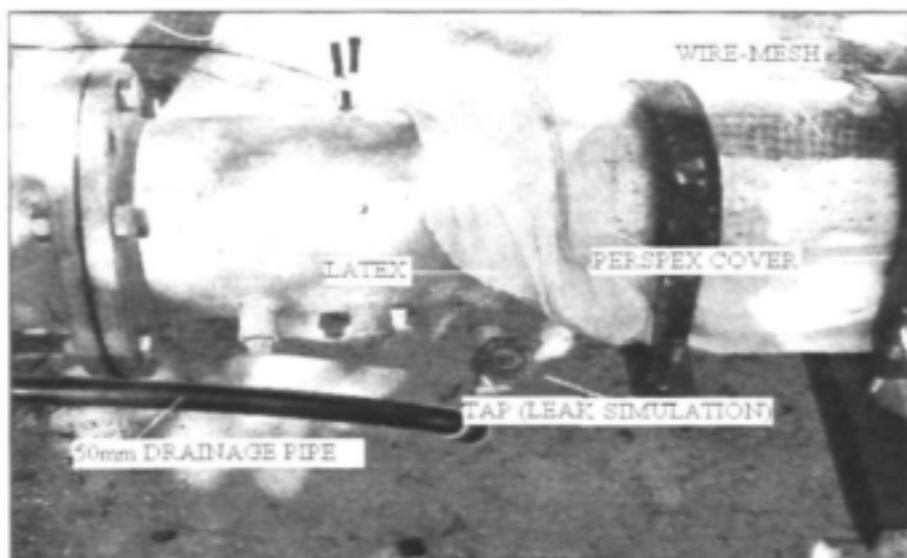


Figure 4.4.6: Simulation of a leak in the pipeline

Pressure transducers (**Figure 4.4.7**) were placed on the system at three points. The first was placed upstream of the leakage point, the second at the leakage point and the third downstream of the leakage point just behind the ball valve. These transducers were used to capture the pressure wave movement of the negative transient that was generated by the fast opening of the ball valve at the end of the line.



Figure 4.4.7: Pressure transducer at the upstream end of the leakage

The pressure transducer type used in the experiments had a working range of 0 to 3 bar absolute pressure. Absolute pressure indicates that atmospheric pressure is measured above the pressure of the fluid in the pipeline, thus values obtained comprised of the atmospheric pressure component and partly of the gauge pressure of the fluid or gas investigated. Absolute pressure transducers were used since the inverse transient wave that was generated could have reached negative gauge pressure values. The transducer output was, in the range of 4 to 20 mA. Included in the circuitry was a 500 Ohm resistor. The voltage output across the resistor was calculated by the equation $V = IR$. Thus for 4 mA, the corresponding voltage is 2 Volts and for 20 mA it is 10 Volts. The transducer output was recorded by a notebook computer fitted with the data acquisition card, as Volts, accurate to 4 decimals and in the operating range of between 2 and 10 Volts. The correlation of transducer output in Volts and absolute pressure is given in the graph below (**Figure 4.4.8**).

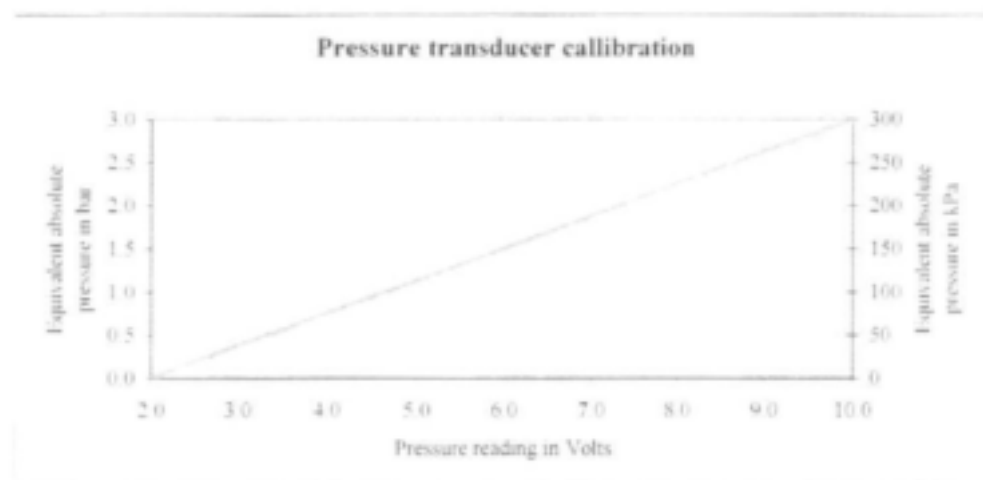


Figure 4.4.8: Pressure transducer calibration, Volts to pressure in bar and kPa

The absolute zero reading of the pressure transducer is 2 Volts. It is desired to equate this to gauge atmospheric pressure. In so doing the adapted transducer readings are gauge pressures of the fluid, the actual pressure of the fluid. The relationship is rectilinear and governed by the equation,

$$P = 37.5 V - 75 \dots \text{for } 2 < V < 10 \dots \dots \dots (4.4.2)$$

Where:

- P = Absolute pressure (kPa)
V = Transducer output (Volts)

Thus for a transducer reading of 2 volts the absolute pressure is 0 kPa.

The following conversion table, **Table 4.4.1**, has been included as a quick reference.

Table 4.4.1: Pressure / head unit conversions

1 kPa	1 kN/m ²
1 bar	100 kPa
1 bar	102.074 m H ₂ O
1 m H ₂ O	9.800 kPa
1 kPa	0.10204 m H ₂ O
1 atm (std)	101.325 kPa

The proper functioning of each transducer was checked from a set of data. Using the records of atmospheric pressures, the following graph was generated.

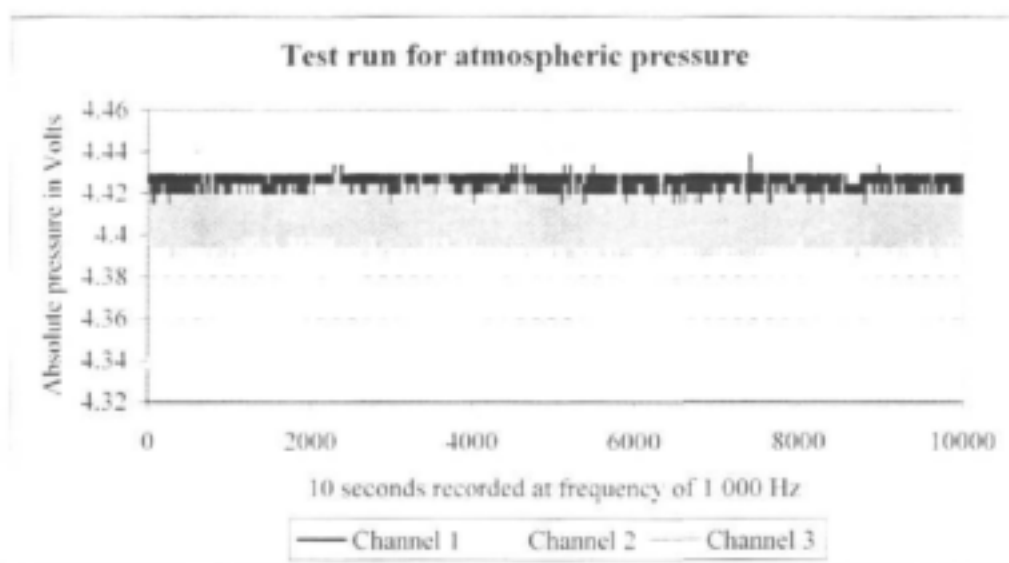


Figure 4.4.9: Plot of 10 000 readings of absolute atmospheric pressure in volts

In **Figure 4.4.9**, it can be seen that the records for each transducer are within a band of width of about 0.02 volts. Fluctuation in records is in the second decimal, thus the equivalent of 0.375 kPa. No peaks occur in the plot and the transducers were of the correct range to be used in the experiments. Peaks were found in data sets for short circuits in the transducer leads, typically when the coaxial connector is bare and in contact with the pipe. By wrapping all connectors with insulation tape, short circuits were prevented.

Separate tests were carried out to reflect the presence of a leak and the presence of a localised air bubble.

Since the pressure transducers weren't all installed at the same height (**Figure 4.4.10 (a)**), the system had to be calibrated to obtain the ability to directly compare the readings from the different pressure transducers. The system was pressurised by closing the leak and the valves at the end of the line. The static pressures at each of the measuring points were determined with the pressure transducers. Three sets of static pressure tests were performed. The first and the last test were performed at almost the same constant pressure in the system whilst the second test was performed at a higher constant pressure.

The measurements of the transducers were captured in volts (V) and the conversion had to be made from volts to pressure. The pressure (in bar) then had to be converted to absolute pressure (in meters). After the pressure was converted to absolute pressure, atmospheric pressure had to be subtracted in order to obtain gauge pressure.

Figure 4.4.8. which depicts the relationship between the voltmeter readings and pressure in bar, was used for the conversion. For values in between, interpolation was used.

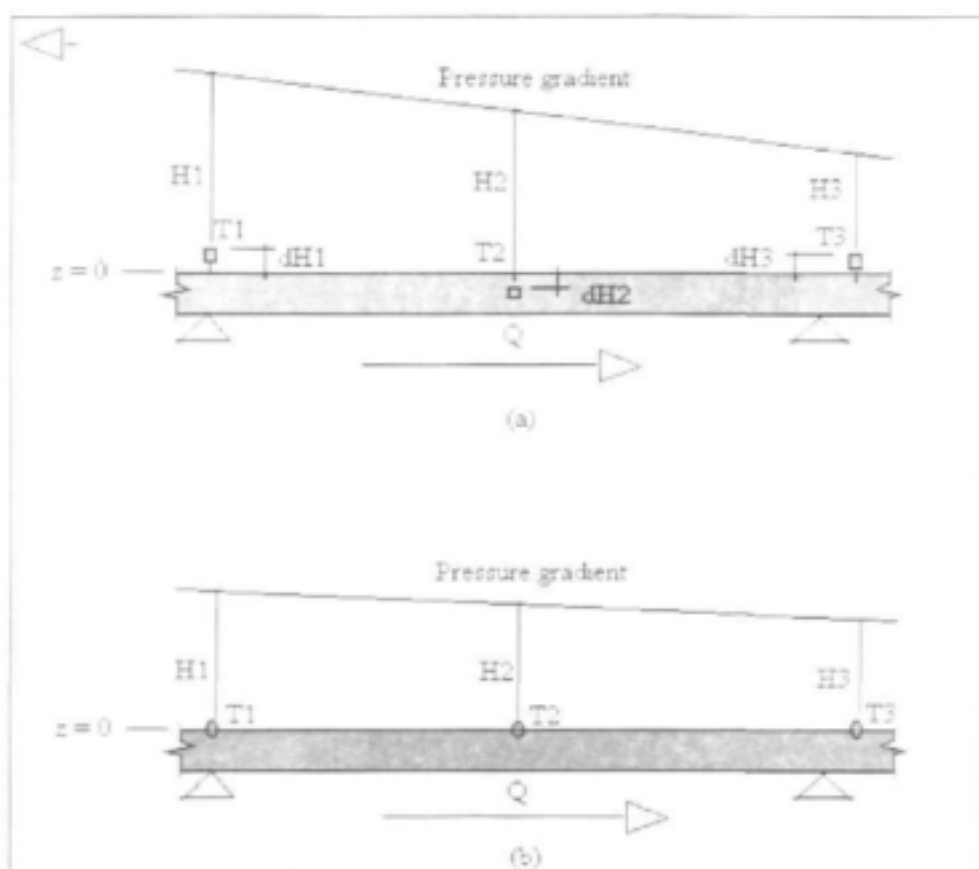


Figure 4.4.10: Placement of the pressure transducers on the same datum level

Since the transducer just upstream of the ball valve and the transducer furthest upstream from the ball valve were approximately at the same height, it was expected that the difference in pressure between them would be zero or very close to zero.

The averages of all the pressures recorded by the transducers in each test were determined. It was decided to take one of the averages of each test as the zero reading for the data set and it was subtracted from the others. This was done for each static pressure test. In **Table 4.4.2**, which is a shortened version of the recorded data set, the difference in pressure for each transducer in each static pressure test can be seen.

Table 4.4.2: Static pressure tests with the calculations of dh

	STAT 1			STAT 2			STAT 3		
	Transducer (Volts)			Transducer (Volts)			Transducer (Volts)		
Time (s)	T1	T2	T3	T1	T2	T3	T1	T2	T3
9.9750	7.2512	7.2464	7.2512	9.2337	9.2386	9.2483	7.1487	7.1536	7.1487
9.9760	7.2512	7.2464	7.2561	9.2386	9.2435	9.2337	7.1536	7.1536	7.1487
9.9770	7.2464	7.2464	7.2512	9.2435	9.2483	9.2386	7.1536	7.1487	7.1487
9.9780	7.2512	7.2464	7.2561	9.2483	9.2386	9.2337	7.1585	7.1536	7.1536
9.9790	7.2464	7.2464	7.2512	9.2483	9.2337	9.2288	7.1585	7.1536	7.1585
9.9800	7.2512	7.2464	7.2561	9.2483	9.2386	9.2337	7.1633	7.1536	7.1585
9.9810	7.2561	7.2512	7.2610	9.2435	9.2483	9.2337	7.1585	7.1585	7.1585
9.9820	7.2512	7.2512	7.2610	9.2386	9.2483	9.2337	7.1585	7.1585	7.1633
9.9830	7.2512	7.2512	7.2610	9.2435	9.2435	9.2386	7.1633	7.1585	7.1682
9.9840	7.2512	7.2464	7.2610	9.2386	9.2435	9.2435	7.1633	7.1585	7.1633
9.9850	7.2512	7.2464	7.2659	9.2435	9.2435	9.2386	7.1633	7.1585	7.1633
9.9860	7.2512	7.2512	7.2610	9.2386	9.2483	9.2483	7.1633	7.1633	7.1682
9.9870	7.2512	7.2512	7.2610	9.2386	9.2483	9.2581	7.1633	7.1633	7.1682
9.9880	7.2512	7.2512	7.2561	9.2435	9.2435	9.2483	7.1633	7.1633	7.1682
9.9890	7.2512	7.2512	7.2610	9.2435	9.2532	9.2483	7.1633	7.1633	7.1682
9.9900	7.2464	7.2512	7.2561	9.2483	9.2532	9.2532	7.1633	7.1633	7.1731
9.9910	7.2512	7.2512	7.2512	9.2435	9.2483	9.2435	7.1682	7.1682	7.1633
9.9920	7.2512	7.2512	7.2561	9.2483	9.2435	9.2386	7.1682	7.1682	7.1731
9.9930	7.2512	7.2512	7.2512	9.2483	9.2483	9.2386	7.1682	7.1682	7.1731
9.9940	7.2512	7.2512	7.2512	9.2483	9.2532	9.2386	7.1682	7.1731	7.1731
9.9950	7.2561	7.2512	7.2610	9.2483	9.2532	9.2435	7.1682	7.1682	7.1682
9.9960	7.2512	7.2512	7.2610	9.2483	9.2483	9.2435	7.1682	7.1682	7.1731
9.9970	7.2512	7.2464	7.2610	9.2435	9.2386	9.2386	7.1731	7.1731	7.1682
9.9980	7.2512	7.2464	7.2610	9.2435	9.2483	9.2483	7.1780	7.1682	7.1731
9.9990	7.2512	7.2464	7.2659	9.2386	9.2532	9.2483	7.1780	7.1731	7.1682
10.0000	7.2561	7.2464	7.2659	9.2386	9.2483	9.2435	7.1731	7.1731	7.1731
Total	72629.2	72633.8	72718.8	92394.2	92449.3	92422.0	72432.8	72451.3	72427.1
Ave.	7.2629	7.2634	7.2719	9.2394	9.2449	9.2422	7.2433	7.2451	7.2427
dh	0	0.0005	0.009	0	0.006	0.003	0	0.002	-0.001

The recordings of the last static pressure test were used in calculations since, the difference in pressure between the transducer just upstream of the ball valve and the transducer furthest upstream from the ball valve was zero, as was expected. By subtracting this difference in pressure from the pressures that were recorded in the tests, the difference in height between the pressure transducers was eliminated (**Figure 4.4.10(b)**).

The data acquisition program used to capture the data from the pressure transducers was set at a frequency of 1000 Hz with a time frame of 10 seconds. This resulted in 10 000 readings being taken in each time frame of each test, for each pressure transducer.

The bubble as constructed in the first experimental set-up provided some problems. When it was initially attempted to place a localised air bubble in the pipe the force of the water tore the bubble. The wire mesh was placed in the pipe to function as a boundary for the bubble but this caused a too great restriction on the expansion capabilities that the air has during a pressure change. A second experimental set-up was subsequently made specifically for a localised air bubble.

4.4.3.2 Second experimental set-up

The same layout as indicated in **Figure 4.4.2** was used for the second experimental set-up but instead of the air bubble underneath the Perspex cover, a T-piece was installed in which various volumes of air could be induced, see **Figure 4.4.11**.



Figure 4.4.11: T-piece for inducing various volumes of air



Figure 4.4.12: View of system layout looking upstream from the ball valve at the end of the pipe

With the experimental set-up shown in Figure 4.4.12, it was possible to introduce different volumes of air into the T-piece and then determine what the effect of that volume was on the propagation of the induced wave. The same data capturing equipment as used in the first experimental set-up was again used and thus the description thereof is not repeated.

The pressure transducer, for position 2, was placed at the T-piece at level 5 (see **Figure = 4.4.13**). Air would be introduced at level 4 until the entire T-piece was filled with air. The air would then be bled off at levels 1, 2 or 3 until water spurts out at the selected level. The volume of air would then be the volume above the level at which the excess air was released. This allowed three volumes of air to be tested referred to as, large (level 3), medium (level 2) and small (level 1) bubble. The pressure from the pressure transducer was used to convert the compressed volume of air into normal litres of air. Releasing the volume of air through the ball valve at level 0 and measuring it by a simple displacement test in a separate container made was a further check.

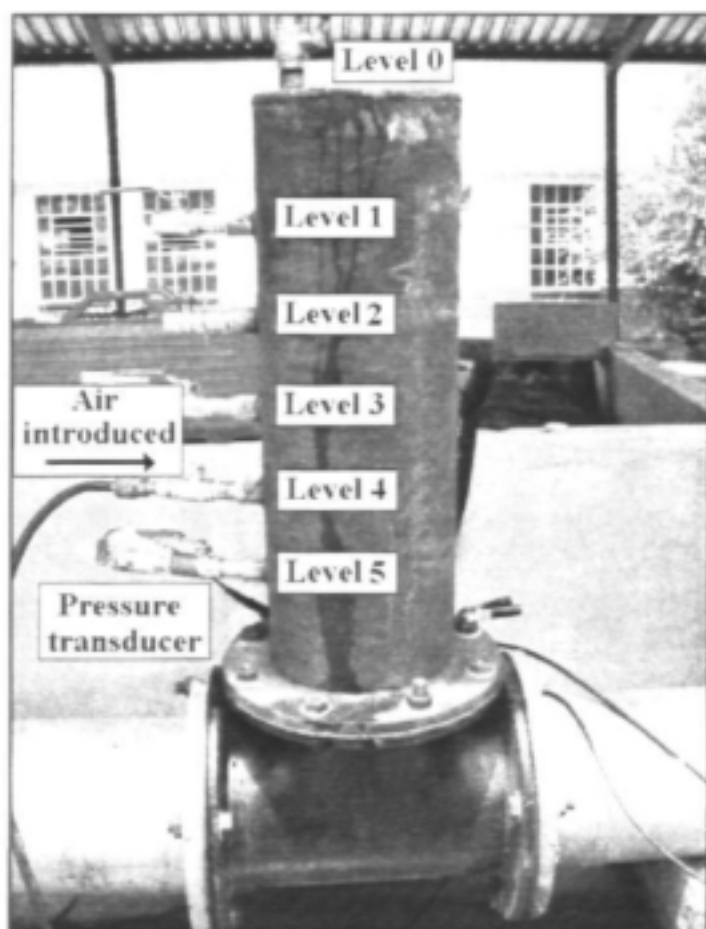


Figure 4.4.13: T-piece with introduced air

Various tests were conducted by inducing various volumes of air and testing at different flow rates.

Similar to the first experimental set-up, the transducers were all "placed" at the same height by adding a correction factor and the readings were converted to gauge pressure in meters. A plot was then made of pressure versus time for each transducer individually.

In the paragraphs that follow, attention will first be given to the experiments that were conducted with a leak in the system. This is followed by the discussion of the modelling of the influence of a localised air bubble on the transient wave propagation.

4.4.4 Experiments executed to determine the influence of leaks on the propagation of transient waves

4.4.4.1 Methodology

In order to induce an inverse transient in the pipeline, the ball valve at the end of the pipeline had to be opened for a short period of time while the pipeline was under pressure. The opening of the ball valve immediately increased the flow rate in the pipeline, with the associated negative transient wave that travelled upstream in the pipe.

During the filling of the pipe, both the ball valve and the bypass at the end of the pipeline were left open to ensure that all the air was removed from the pipeline and that the pipeline was filled completely. After the pipeline was de-aerated, the ball valve was closed slowly whilst the bypass was left open. This procedure was followed for two reasons:

- Rapid closure of the ball valve would have caused a positive transient to propagate through the pipeline, which in turn could have damaged the pipeline, causing the pipe to kick, which was unacceptable because the pipeline was not anchored to the ground.
- The bypass was used to regulate the pressure in the pipeline. The transducers' used could withstand pressures of 3 bar (30 m absolute pressure). For this reason, the pressure in the pipeline had to be kept below this limit in order to protect the pressure transducers.

After the ball valve was completely closed, the pressure in the pipeline was monitored by means of the data acquisition program, "Quatech". The program default setting was changed to allow the data to be captured at a frequency of a 1 000 Hz. Data for each of the tests were recorded for a period of at least 10 seconds.

At the start of the tests, the ball valve was closed and the tap at the off-take was opened to create flow that represented a leak in the pipeline. The flow through the leak was varied and determined by means of the mass balance procedure discussed below.

A small drainage pipe ($\theta = 50$ mm) was connected to the tap with the outlet of the pipe releasing back into the channel. The flow rate through the leak was measured manually by

capturing the flow in a container for a period of time. The time it took to fill up the container was measured by means of a stopwatch and the water captured in the container was weighed. With the density of water ($\rho = 1000 \text{ kg/m}^3$) and the weight of the water that leaked out known, it was possible to calculate the volume of water that leaked out through the tap with the following equation:

$$\rho = \frac{m}{V} \dots\dots\dots (4.4.3)$$

Where:

ρ = Density of water (kg/m^3)

m = Mass of water (kg)

V = Volume of water (m^3)

The volume and the time interval were then used to determine the flow rate through the leak using equation 4.4.4.

$$Q = \frac{V}{t} \dots\dots\dots (4.4.4)$$

Where:

Q = Flow rate (m^3/s)

V = Volume of water (m^3)

t = Time interval (s)

In **Table 4.4.3** the volume and the flow rate of the leak from the pipe for one of the tests (NEG I) is reflected. Two readings were taken to determine the flow through the leak and the average of the two was used.

Table 4.4.3: Procedure for the calculation of the leakage flow rate

NEG I					
Mass (kg)	Density (kg/m^3)	Volume (m)	Time (s)	Q (m^3/s)	Q (l/s)
15.25	1000	0.01525	13.30	0.001147	0.114662
17.20	1000	0.01720	14.73	0.001168	0.116768
Average					0.115715

The flow in the pipeline was also varied by means of the butterfly valve located just after the pump and was measured with an ultrasonic flow meter (**Figure 4.4.14**).

After the above-mentioned data was collected, a negative transient was induced in the pipeline. The data acquisition program was set up to capture the pressure in the pipeline as recorded by the pressure transducers. The program had to be triggered before any data was to be collected. When the data acquisition for the specific test was started, a short time-delay was allowed before the ball valve was opened. This was done to record the pressures in the pipeline without any transients being present. The pressures associated with the transient were recorded by the three pressure transducers and acquired with the data acquisition program.

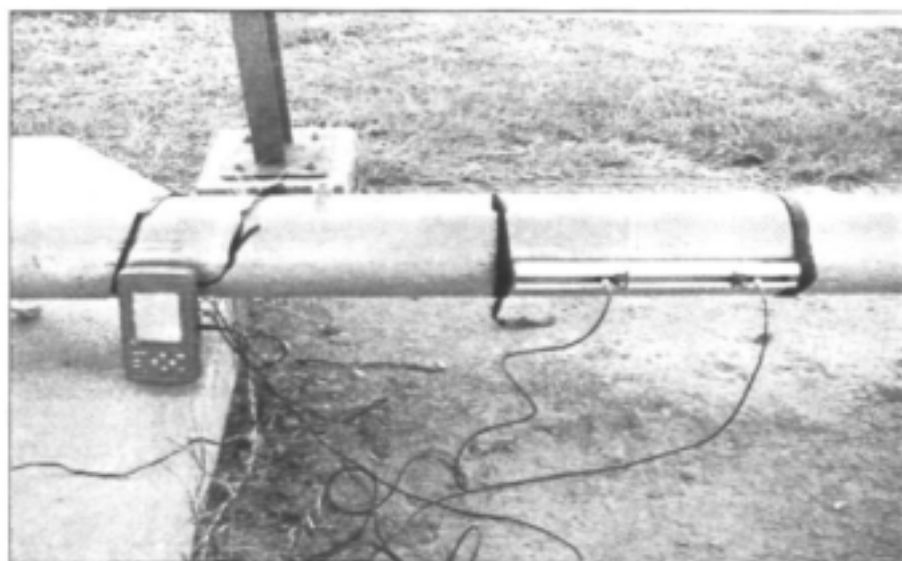


Figure 4.4.14: Flow meter used to determine the flow in the pipe

These graphs were then superimposed on the same axis. **Table 4.4.4** shows a typical worksheet for the work procedure that was followed for test NEG I-1 to eventually obtain the values used in order to draw up the graph for the data set. **Figure 4.4.15** shows the graph generated from the data listed. Again it must be noted that **Table 4.4.4** is a shortened table due to the number of readings that were taken within the time frame of 10 seconds.

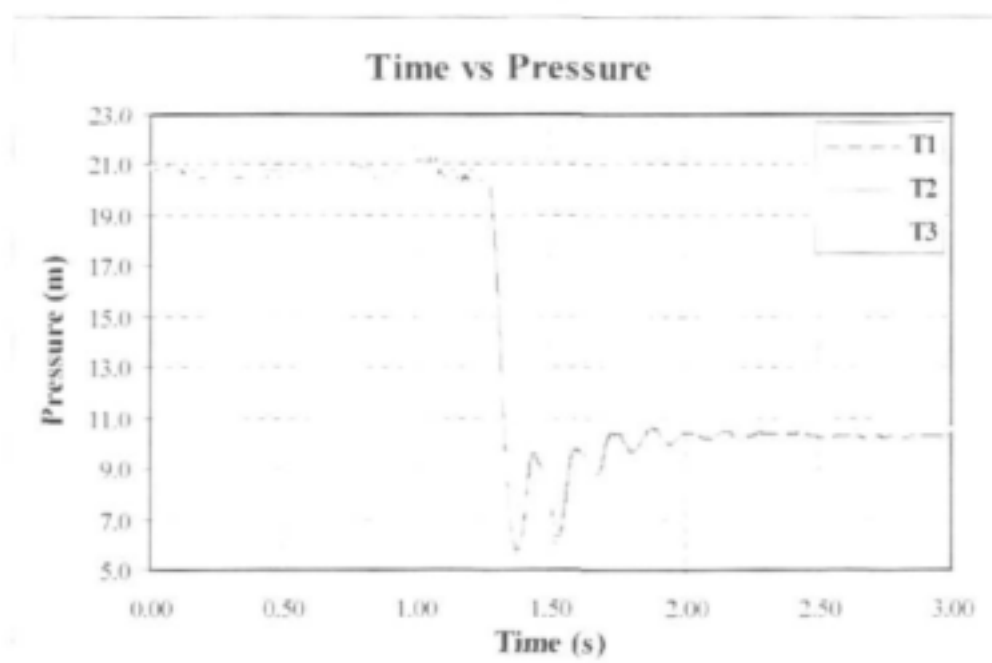


Figure 4.4.15: Pressure versus time for test NEG I – test run 1

In **Figure 4.4.15** the constant pressures recorded in the pipeline (from $t = 0$ s to $t = 1.3$ s) before the ball valve was opened, can be seen. Once the data acquisition software triggered to commence recording, a short time delay (in this case 1.3 s) was allowed before the ball valve was opened. The constant pressures recorded relate to the time before the opening of the ball valve. After the ball valve was opened, the negative transient propagated through the pipeline, resulting in the sudden drop in pressure. The rise in pressure at $t = \pm 1.37$ s relates to the reflection of the transient (positive) from the pump.

The oscillations in the pressures (measured in Volt) on the graph show the propagation of the transient through the pipeline. A rise in pressure is associated with the transient (positive) that is reflected back from the pump and a drop in pressure relates to the transient (negative) that is reflected from the open ball valve, which functions as a control (atmospheric pressure). In a very short time, the pressures reach a new equilibrium, which can be attributed to pipe friction and deformation energy used during the transient movement. The pipe system itself is also relatively short.

Table 4.4.4: Typical worksheet for the manipulation of captured data

Negative pressure test I; Results

NEG 1

[No air bubble with leakage: $q = 1.147$ l/s]

[$Q = 22.14$ l/s ; $V = 0.713$ m/s]

dH	0.0000	0.0018	0.0000
----	--------	--------	--------

Time(s)	Pressure transducer(Volts)			Pressure transducer on the same height (Volts)			Pressure transducer on the same height (meters)			Time(s)	Moving average		
	T1	T2	T3	T1	T2	T3	T1	T2	T3		T1	T2	T3
0.001	8.7845	8.7112	8.7112	8.7845	8.7094	8.7112	21.0209	20.8196	20.7609	0.001	20.9386	21.0122	20.8797
0.002	8.7991	8.7112	8.7063	8.7991	8.7094	8.7063	21.0756	20.8196	20.7425	0.002	20.9416	21.0258	20.8844
0.003	8.8040	8.7112	8.7112	8.8040	8.7094	8.7112	21.0940	20.8196	20.7609	0.003	20.9410	21.0400	20.8892
0.004	8.7991	8.7405	8.7161	8.7991	8.7387	8.7161	21.0756	20.9295	20.7793	0.004	20.9380	21.0530	20.8933
0.005	8.7942	8.6917	8.7112	8.7942	8.6899	8.7112	21.0573	20.7465	20.7609	0.005	20.9351	21.0542	20.8974
0.006	8.7893	8.7356	8.7161	8.7893	8.7338	8.7161	21.0389	20.9111	20.7793	0.006	20.9327	21.0677	20.9039
0.007	8.7796	8.7210	8.7161	8.7796	8.7192	8.7161	21.0025	20.8563	20.7793	0.007	20.9309	21.0731	20.9104
0.008	8.7698	8.6868	8.7210	8.7698	8.6850	8.7210	20.9658	20.7281	20.7977	0.008	20.9303	21.0807	20.9193
0.009	8.7649	8.7503	8.7259	8.7649	8.7485	8.7259	20.9474	20.9662	20.8160	0.009	20.9315	21.0985	20.9287
0.010	8.7649	8.7405	8.7308	8.7649	8.7387	8.7308	20.9474	20.9295	20.8344	0.010	20.9339	21.1038	20.9382
0.011	8.7649	8.7454	8.7405	8.7649	8.7436	8.7405	20.9474	20.9478	20.8708	0.011	20.9363	21.1138	20.9470
0.012	8.7649	8.7356	8.7454	8.7649	8.7338	8.7454	20.9474	20.9111	20.8892	0.012	20.9386	21.1215	20.9553
0.013	8.7649	8.7747	8.7454	8.7649	8.7729	8.7454	20.9474	21.0577	20.8892	0.013	20.9404	21.1256	20.9624
0.014	8.7649	8.7552	8.7454	8.7649	8.7534	8.7454	20.9474	20.9846	20.8892	0.014	20.9404	21.1327	20.9689
0.015	8.7649	8.7356	8.7454	8.7649	8.7338	8.7454	20.9474	20.9111	20.8892	0.015	20.9381	21.1451	20.9736
0.016	8.7649	8.8138	8.7503	8.7649	8.8120	8.7503	20.9474	21.2043	20.9075	0.016	20.9316	21.1498	20.9766
0.017	8.7552	8.7893	8.7454	8.7552	8.7875	8.7454	20.9110	21.1125	20.8892	0.017	20.9215	21.1457	20.9778
0.018	8.7405	8.7356	8.7454	8.7405	8.7338	8.7454	20.8559	20.9111	20.8892	0.018	20.9085	21.1498	20.9784
0.019	8.7210	8.7845	8.7503	8.7210	8.7827	8.7503	20.7828	21.0945	20.9075	0.019	20.8938	21.1481	20.9778
0.020	8.7015	8.8382	8.7552	8.7015	8.8364	8.7552	20.7096	21.2958	20.9259	0.020	20.8808	21.1487	20.9754

In order to determine the location of the leak it was necessary to investigate the graphs drawn from the experimental data (as in **Figure 4.4.15**). On the part of the graph where the pressure dropped due to the rapid opening of the ball valve there would, in theory, be a slight variation, as part of the wave is reflected back from the leak. If this was the case, it would have been possible to estimate the time difference between the drop in pressure and the plateau from the graphs and since the celerity of each transient was known, it would have been possible, with the use of Equation 4.4.1, to estimate where the leak was located in the pipeline, relative to the pressure measuring point.

To explain the procedure that one would follow to estimate the location of a leak in a simple pipeline (horizontal pipeline with no networking) **Figure 4.4.16** was drawn for illustration purposes.

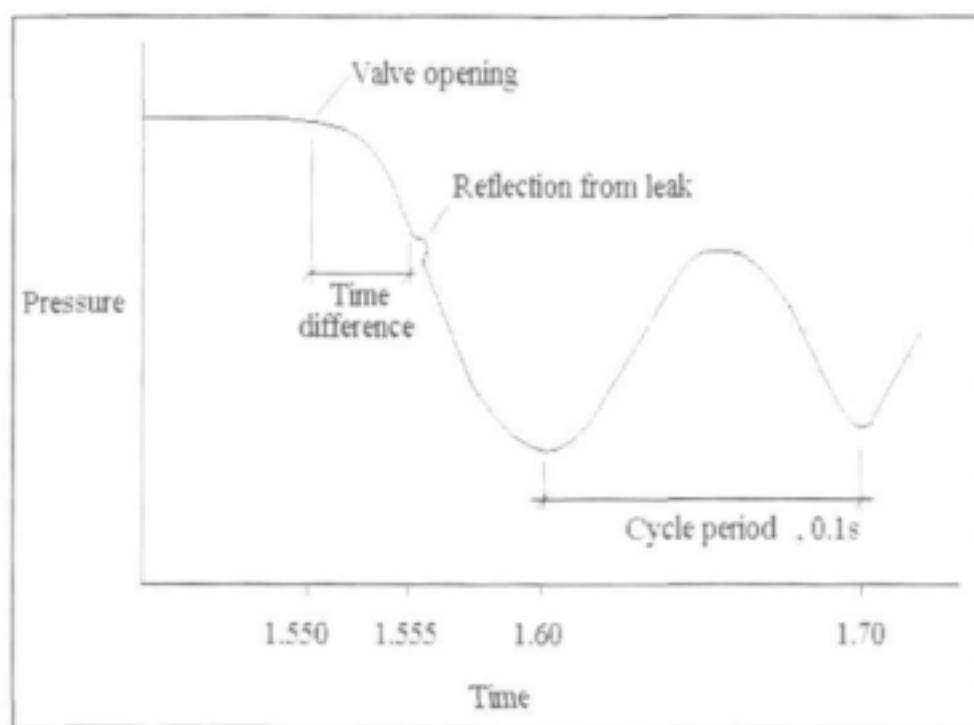


Figure 4.4.16: Pressure variation indicating influence of leak on propagation of transient

If, for argument sake, the pipeline had a length of 25 m with a leak at 5 m from the point where the transient originated from and a cycle period of 0.1 s for the transient, the wave celerity can be determined:

$$c = \frac{4L}{t} = \frac{(4)(25)}{0.1} = 1000 \text{ m/s}$$

The difference in time between the drop in pressure and the reflected wave from the leak is calculated as 0.005 seconds. With the use of Equation 4.4.3 the location of the leak, with the origin of the transient as reference point, can be determined as follows:

$$c = \frac{2s}{t} \quad s = 1000 \times \frac{0.005}{2} = 2.5m$$

The location of the leak, however, could not be determined from the data obtained for this experimental set-up, due to the limited length of the line and the rate of 1 000 Hz at which the data was recorded.

By measuring the pressures for a certain experimental set-up, it is possible, however, to establish the influence of a leak on the propagating transient wave. If the pressure fluctuations with a leak are compared to those for a system with no leakage, the influence of the size of the leak on the transient movement can be obtained.

4.4.4.2 Results from the tests that were conducted on a leak in a pipeline

Nine tests (test A – test I) were carried out and each test was repeated five times. This was done to compensate for possible faulty test results. A summary of the tests and their parameters are shown in Table 4.4.5.

Table 4.4.5: Test summary of the inverse transient test conducted for a leak in the pipe

Test	Ball valve opening	Bypass valve opening	Pipe flow, Q_1			Out flow, Q_2 (Leak)	
			Velocity (m/s)	Q (m ³ /s)	Q (l/s)	Q (m ³ /s)	Q (l/s)
A	80	Closed	0.45600	0.1430	14,300	None	None
B	80	Closed	0.09000	0.0300	3,000	0.00046	0.04567
C	80	Closed	0.02342	0.0075	0.754	0.00078	0.07758
D	80	Closed	0.02641	0.0085	0.850	0.00078	0.07758
E	80	Closed	0.02641	0.0085	0.850	0.00093	0.09282
F	100	Closed	0.03200	0.0104	1,037	0.00119	0.11941
G	100	Open	0.30000	0.0940	9,400	0.00094	0.09360
H	100	Open	0.53000	0.1650	16,500	0.00032	0.03216
I	100	Open	0.71300	0.2214	22,140	0.00116	0.11577

The pressure reduction of the propagating inverse transient was calculated for all the tests.

Table 4.4.6 gives a summary of the results.

Table 4.4.6: Summary of the tests conducted on the influence of a leak on the propagation of a pressure wave

Test	NEG1000		NEG1001		NEG1002		NEG1003		NEG1004	
	Ds	Us	Ds	Us	Ds	Us	Ds	Us	Ds	Us
Avg Start	7.190	7.197	7.244	7.232	7.253	7.247	7.237	7.239	7.203	7.204
Avg End	7.179	7.201	7.221	7.239	7.218	7.219	7.251	7.264	7.279	7.276
Min	4.634	5.049	4.546	4.722	4.607	4.820	4.429	4.717	4.449	4.595
Delta	9.612	8.101	10.164	9.474	10.239	9.034	10.697	9.588	10.692	10.091

Test	NEG2000		NEG2001		NEG2002		NEG2003		NEG2004	
	Ds	Us	Ds	Us	Ds	Us	Ds	Us	Ds	Us
Avg Start	7.223	7.187	7.196	7.176	7.166	7.163	7.204	7.183	7.208	7.196
Avg End	7.190	7.181	7.220	7.212	7.164	7.174	7.167	7.167	7.181	7.180
Min	4.527	4.712	4.580	4.893	4.434	4.658	4.532	4.893	4.385	4.610
Delta	10.097	9.294	9.971	8.729	10.319	9.474	9.955	8.562	10.561	9.711

Test	NEG3000		NEG3001		NEG3002		NEG3003		NEG3004	
	Ds	Us	Ds	Us	Ds	Us	Ds	Us	Ds	Us
Avg Start	7.173	7.150	7.196	7.176	7.185	7.158	7.182	7.187	7.201	7.177
Avg End	7.211	7.191	7.191	7.190	7.179	7.192	7.193	7.187	7.169	7.190
Min	4.527	4.854	4.474	4.639	4.341	4.517	4.382	4.629	4.512	4.888
Delta	10.139	8.799	10.455	9.632	10.495	10.072	10.625	9.627	10.038	8.664

Test	NEG4000		NEG4001		NEG4002		NEG4003		NEG4004	
	Ds	Us	Ds	Us	Ds	Us	Ds	Us	Ds	Us
Avg Start	7.206	7.185	7.177	7.158	7.179	7.155	7.219	7.194	7.198	7.178
Avg End	7.170	7.181	7.123	7.117	7.169	7.154	7.160	7.140	7.090	7.107
Min	4.385	4.610	4.375	4.580	4.380	4.556	4.507	4.756	4.390	4.688
Delta	10.522	9.603	10.380	9.550	10.537	9.744	10.021	8.972	10.201	9.100

Test	NEG5000		NEG5001		NEG5002		NEG5003		NEG5004	
	Ds	Us	Ds	Us	Ds	Us	Ds	Us	Ds	Us
Avg Start	7.216	7.164	7.217	7.196	7.223	7.194	7.218	7.191	7.208	7.180
Avg End	7.207	7.187	7.148	7.136	7.192	7.168	7.199	7.178	7.210	7.199
Min	4.405	4.658	4.290	4.396	4.372	4.492	4.341	4.541	4.312	4.434
Delta	10.588	9.518	10.788	10.317	10.881	10.072	10.795	9.915	10.947	10.408

Test	NEG6000		NEG6001		NEG6002		NEG6003		NEG6004	
	Ds	Us	Ds	Us	Ds	Us	Ds	Us	Ds	Us
Avg Start	7.933	7.915	7.929	7.909	7.926	7.903	7.960	7.942	7.932	7.930
Avg End	7.913	7.900	7.915	7.908	7.962	7.937	7.953	7.939	7.939	7.927
Min	4.590	4.678	4.475	4.712	4.590	4.971	4.507	4.668	4.556	4.858
Delta	13.308	12.130	13.032	12.631	12.777	11.168	13.018	11.192	13.537	12.302

Test	NEG7000		NEG7001		NEG7002		NEG7003		NEG7004	
	Ds	Us	Ds	Us	Ds	Us	Ds	Us	Ds	Us
Avg Start	7.268	7.238	7.258	7.238	7.266	7.227	7.293	7.255	7.309	7.260
Avg End	7.269	7.249	7.258	7.243	7.246	7.212	7.303	7.266	7.271	7.238
Min	4.532	4.790	4.751	4.932	4.605	4.790	4.595	4.824	4.585	4.732
Delta	10.340	9.257	9.409	8.698	9.978	9.117	10.227	9.180	10.145	9.430

Test	NEG8000		NEG8001		NEG8002		NEG8003		NEG8004	
	Ds	Us	Ds	Us	Ds	Us	Ds	Us	Ds	Us
Avg Start	6.635	6.603	6.663	6.636	6.647	6.619	6.646	6.616	6.643	6.611
Avg End	6.605	6.586	6.621	6.609	6.645	6.618	6.610	6.592	6.608	6.594
Min	4.644	4.776	4.741	4.820	4.761	4.868	4.741	4.878	4.683	4.781
Delta	7.408	6.817	7.150	6.736	7.119	6.584	7.059	6.453	7.274	6.826

Test	NEG9000		NEG9001		NEG9002		NEG9003		NEG9004	
	Ds	Us	Ds	Us	Ds	Us	Ds	Us	Ds	Us
Avg Start	8.700	8.695	8.668	8.670	8.676	8.695	8.662	8.683	8.658	8.683
Avg End	8.690	8.701	8.663	8.678	8.693	8.688	8.696	8.717	8.664	8.668
Min	4.683	4.810	4.908	5.059	5.069	5.445	5.010	5.238	5.298	5.616
Delta	15.139	14.648	14.189	13.615	13.578	12.210	13.926	13.108	12.715	11.605

Figure 4.4.17 is a graphical presentation of the changes in pressure on the downstream side of the leak (Delta Ds) divided by the change in pressure upstream from the leak (Delta Us) as a function of the proportion of the leak flow (Q_{leak}) divided by the flow (Q)

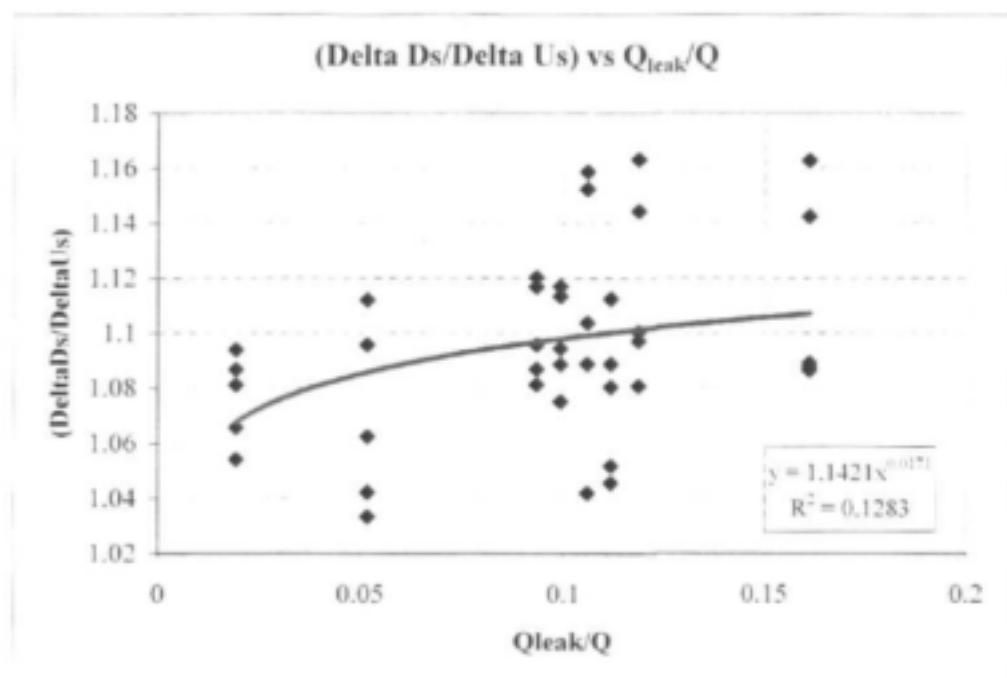


Figure 4.4.17: Graphical presentation of (Delta Ds)/ (Delta Us) against (Q_{leak})/ (Q)

The results are scattered, probably due to the magnitude of the negative pressure that was induced as well as the different opening times of the fast acting valve that was operated by hand. One of the data sets was discarded and the average values for the 40 remaining tests were used to derive a relationship between the change in pressure and the leakage.

Figure 4.4.18 provides a relationship of the change in the pressure fluctuation as a function of the proportion of the leakage.

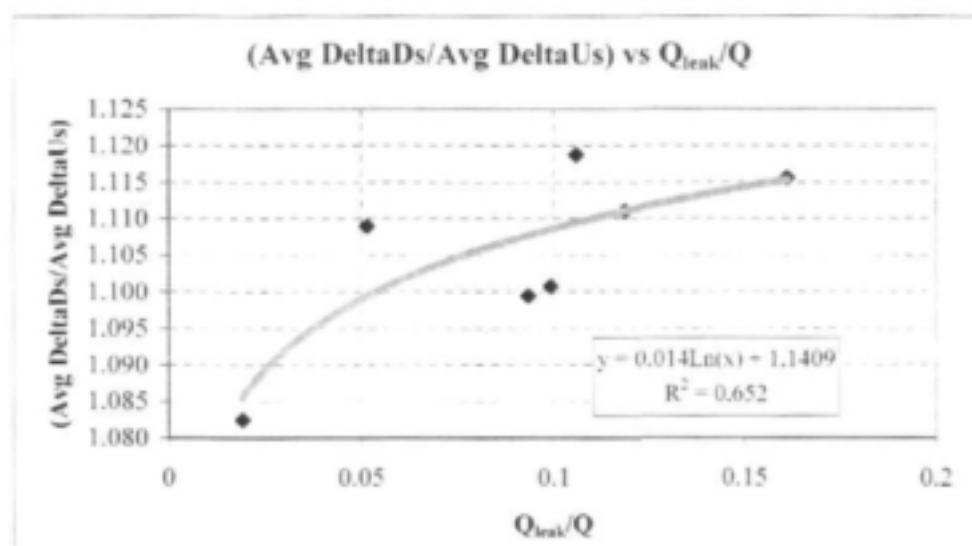


Figure 4.4.18: Change in the pressure fluctuation as a function of the proportion of the leakage

4.4.4.4 Conclusion

Based on the preliminary findings the experimental results indicate the potential use of inverse transients to determine leaks in a pipeline. It is recommended that these tests be repeated for other system layouts to provide a guideline for the use of this technique to determine the potential leakage and the location thereof as well as the position of air in a system.

4.4.5 Experiments executed to determine the influence of a localised air bubble on the propagation of transient waves

4.4.5.1 Methodology

The methodology used in determining the influence of a localised air bubble in a system was similar to that of the leak in the pipeline. The ball valve at the end of the pipeline had to be opened for a short period of time while the pipeline was under pressure. The opening of the ball valve increased the flow rate in the pipeline, with the associated negative transient wave that travelled upstream in the pipe.

During the filling of the pipe, both the ball valve and the bypass at the end of the pipeline were left open to ensure that all the air was removed from the pipeline and that the pipeline was filled completely. After the pipeline was de-aerated, the ball valve was closed slowly as well as the bypass valve. The bleed-off valve on top of the T-piece was used to ensure that the T-piece was properly de-aerated.

A number of inverse transients were generated in the system, which contained no air. Each test consisted of five runs. During the test the flow rate was measured with the ultrasonic flow meter, see **Figure 4.4.14**, (if there was flow), and the pressure was recorded for a 10 second period at all three pressure transducers. In the testing period after approximately 2 seconds the ball valve at the end of the line would be opened rapidly to create the negative transient wave.

Table 4.4.7 is a summary of the tests conducted with this experimental layout.

Table 4.4.7: Test summary of the inverse transient tests conducted for a localised air bubble

Test	By pass valve	Air status
A	Completely closed	No air
B	Partially open	No air
C	Fully open	No air
D	Completely closed	Large air bubble (level 3)
E	Partially open	Large air bubble (level 3)
F	Fully open	Large air bubble (level 3)
G	Completely closed	Medium air bubble (level 2)
H	Partially open	Medium air bubble (level 2)
I	Fully open	Medium air bubble (level 2)
J	Completely closed	Small air bubble (level 1)
K	Partially open	Small air bubble (level 1)
L	Fully open	Small air bubble (level 1)

A typical data graph obtained from a test is shown in **Figure 4.4.19**.

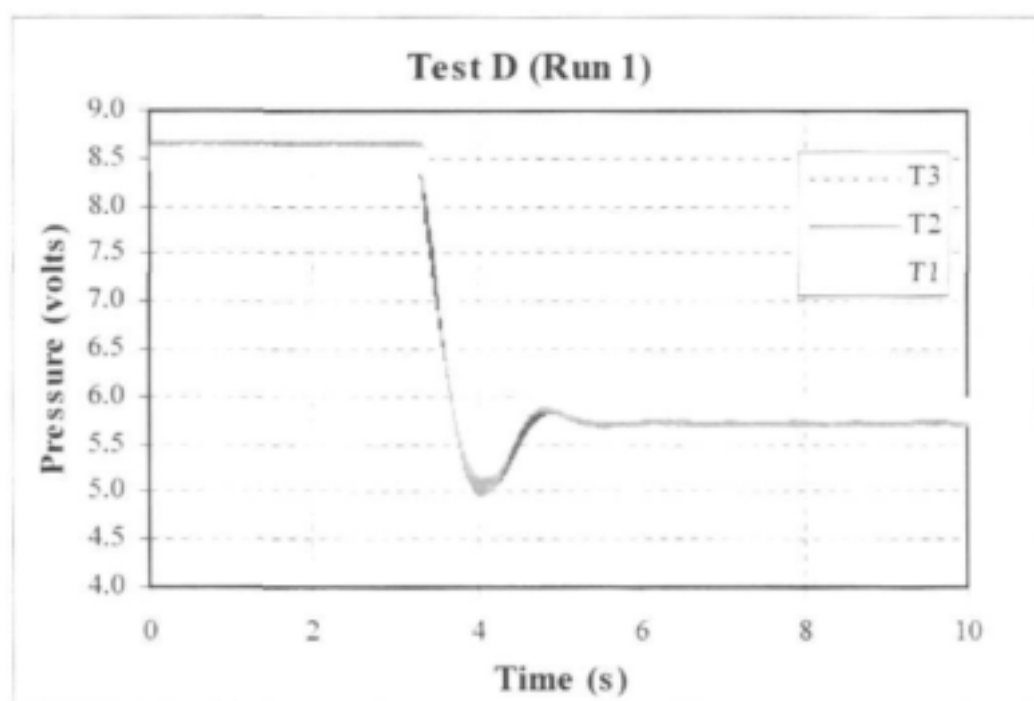


Figure 4.4.19: Pressure versus time for Test D – Test Run 1

In **Figure 4.4.19** the constant pressures recorded in the pipeline (from $t = 0$ s to $t = 2$ s) before the ball valve was opened, can be seen. Once the data acquisition software triggered to commence recording, a short time delay (in this case 3 s) was allowed before the ball valve was opened. The constant pressures recorded relate to the time before the opening of the ball valve. After the ball valve was opened, the negative transient propagated through the pipeline, resulting in the sudden drop in pressure. The rise in pressure relates to the reflection of the transient (positive) from the pump.

In a very short time, the pressures reach a new equilibrium, which can be attributed to pipe friction and deformation energy used during the transient movement. The relative short pipe system also contributed to reaching the equilibrium in a short time span.

The location of the bubble, however, could not be determined from the data obtained for this experimental set-up, due to the limited length of the pipeline and the rate of 1 000 Hz at which the data was recorded.

By measuring the pressures for a certain experimental set-up, it was possible, however, to establish the influence of the bubble on the propagating transient wave. The pressure fluctuations with a bubble were compared to those for a system with no air to obtain the influence of the air on the transient movement.

4.4.5.2 Results from the tests that were conducted on a localised air bubble in a pipeline

A summary of the tests and their parameters are shown in **Table 4.4.7**.

The pressure reduction of the propagating inverse transient was calculated for all the tests. The most significant comparison is showed below for a system with no air and a system with a large volume of air with the bypass closed. This would result in the largest pressure drop in the system.

By comparing **Figure 4.4.20** and **Figure 4.4.21**, the pressure drop due to the induced negative wave, it can be clearly seen that the propagation of the wave through the two systems is different. This is illustrated even more clearly in **Figures 4.4.22** and **4.4.23**, which is the average value of the pressure drop at the three measuring points.

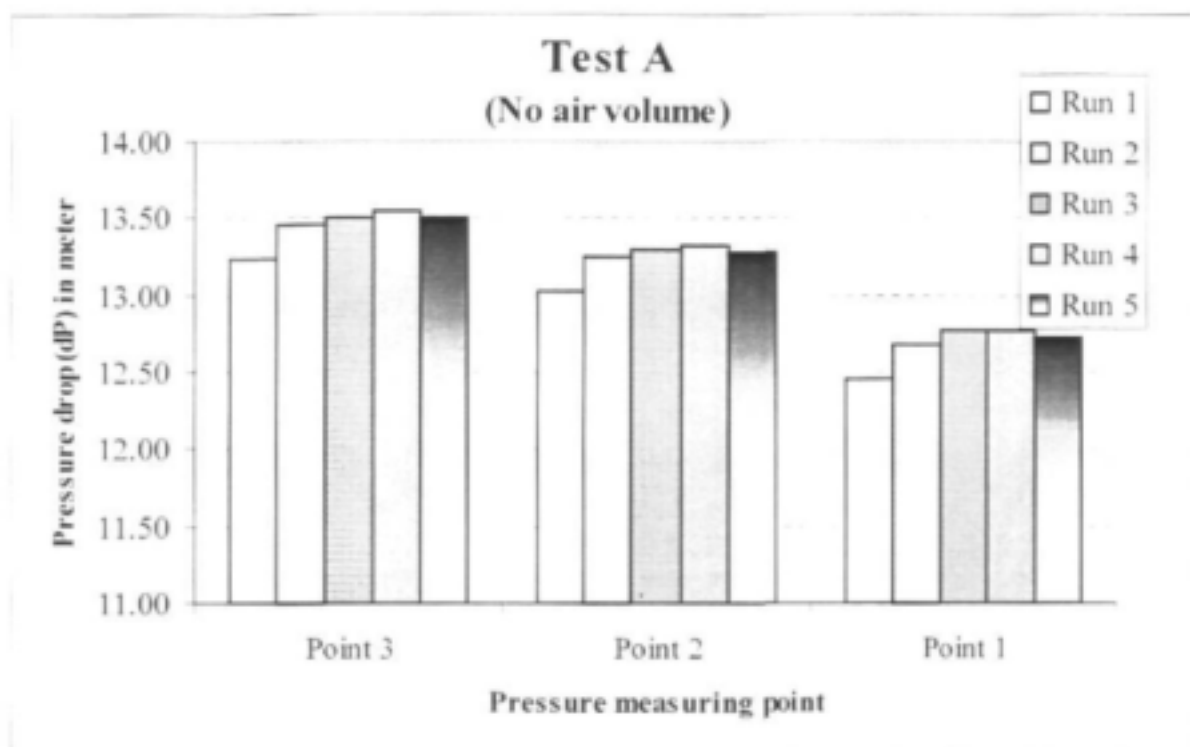


Figure 4.4.20: Pressure drop with no air

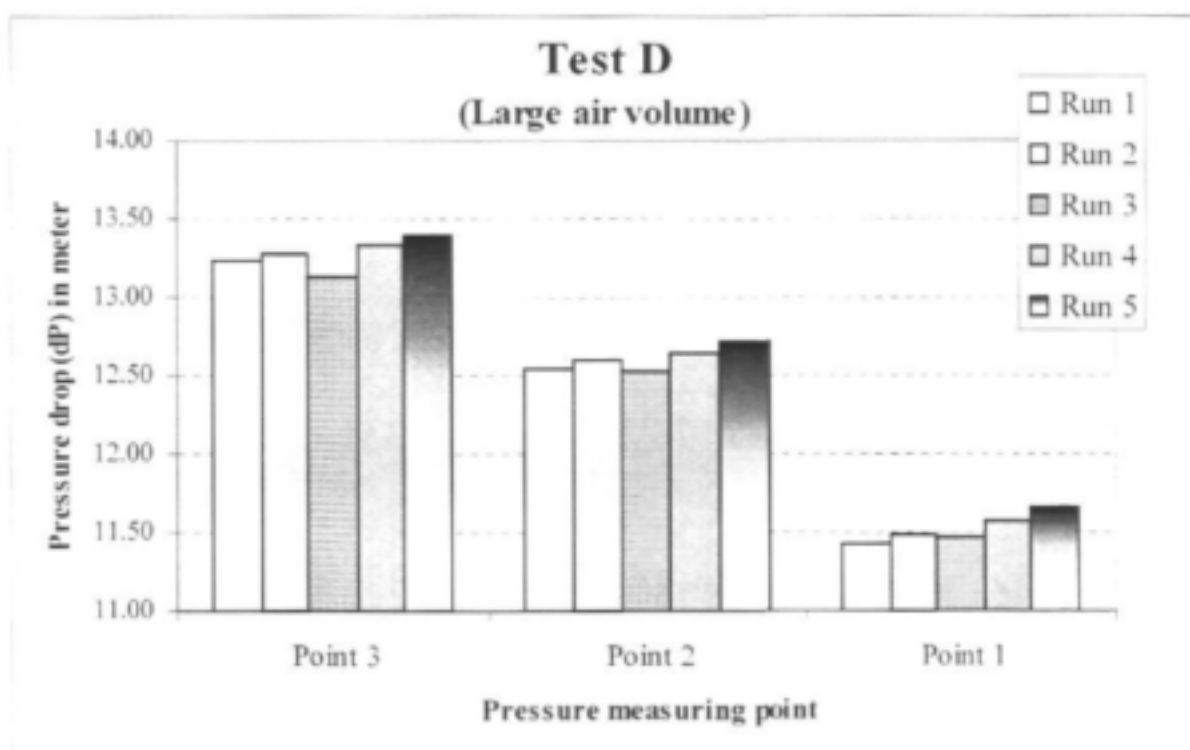


Figure 4.4.21: Pressure drop with large volume of air

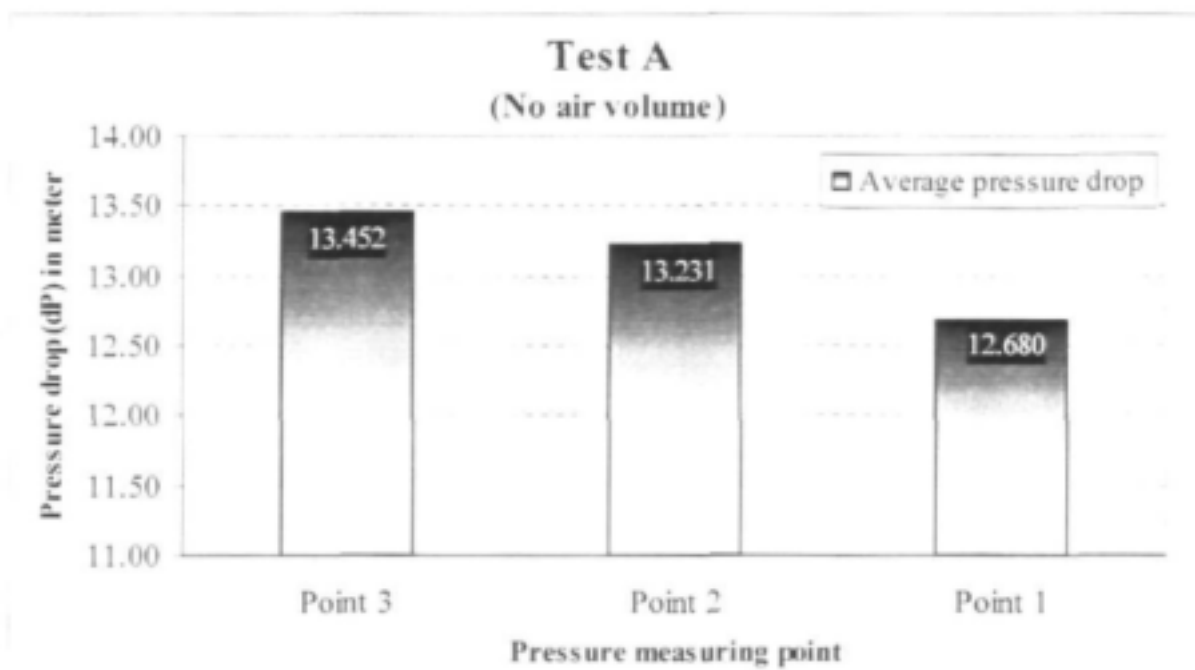


Figure 4.4.22: Average pressure drop with no air

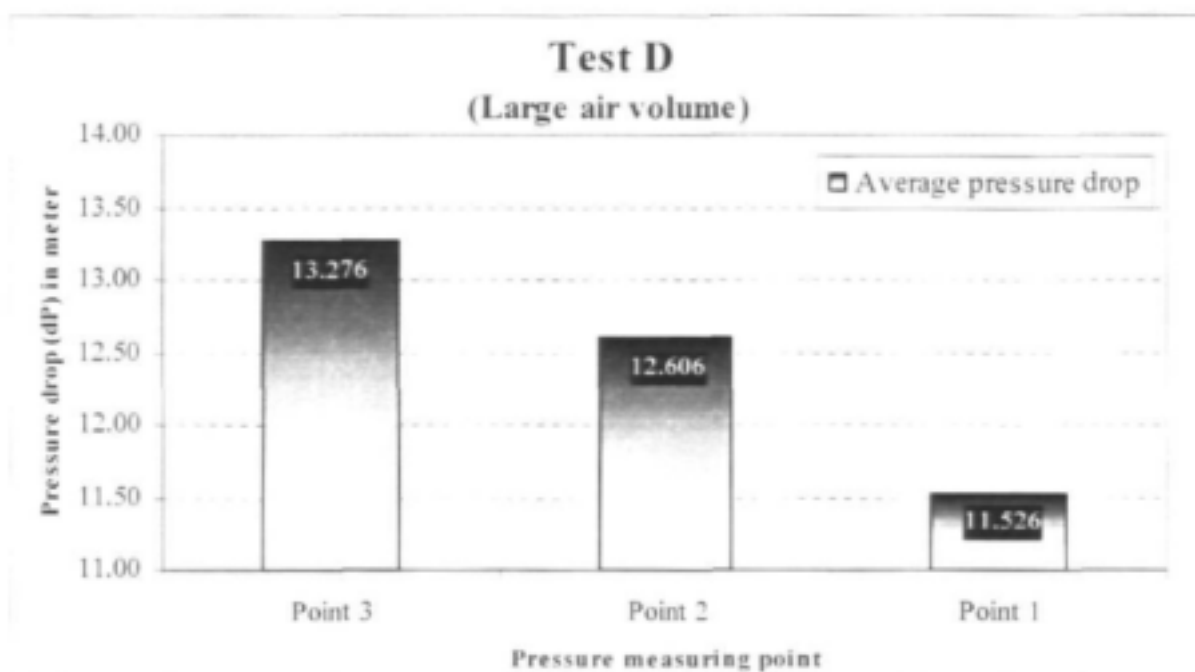


Figure 4.4.23: Average pressure drop with large volume of air

As indicated in **Figure 4.4.22** the pressure drop between Point 3, point closest to the end of the pipe, and Point 1 is 0.772 m. Point 3 relates to T3 in **Figure 4.4.10**, the first experimental set-up. The pressure drop as shown in **Figure 4.4.23** between Point 3 and Point 1 is 1.75 m. This reduction in pressure drop as the wave propagates through the system is directly the result of the presence of free air at Point 2. The pressure drop in the system is also a function of the initial pressure (P_0). The initial pressure for Test A was higher than that of Test D.

This is the reason for the lower pressure drop for Test D compared to Test A. It must also be mentioned that the way in which the valve is opened differs between the tests, but due to the comparison nature of these tests this aspect was not investigated in depth. To compensate for this lower initial pressure the results are presented as a relationship of the drop in pressure (dP) divided by the initial pressure (P_0). This relationship is a non-dimensional relationship shown in **Figure 4.4.24**.

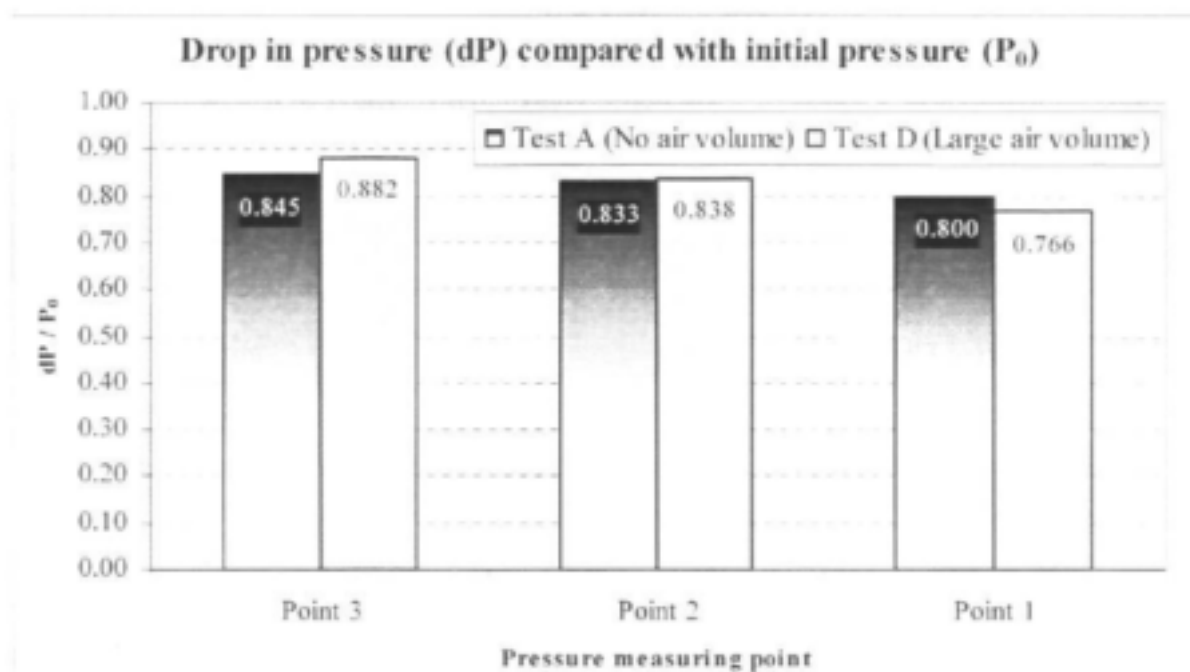


Figure 4.4.24: A non-dimensional relationship of dP/P_0 (Test A and D)

As can be seen in **Figure 4.4.24** the non-dimensional relationship indicates that the pressure drop divided by the initial pressure, for Test A, as the wave propagates through the system changes from 0.845 to 0.800 whilst for Test D it changes from 0.882 to 0.766 which is a steeper gradient.

4.4.5.3 Conclusion

Based on the preliminary findings the experimental results indicate the potential use of inverse transients to quantify the volume of air in a pipeline as well as to determine the location of localised air pockets in a pipeline. It is recommended that these tests be repeated for other system layouts to provide a guideline for the use of this technique to determine the potential amount of air and the location thereof in a system.

4.5 Energy loss associated with localised air bubble and the formation of stable air bubble at the apex point

4.5.1 Introduction

Pipelines are constructed to follow the general profile of the ground for economical and hydraulic reasons. High and low points are therefore created and bends are used to accommodate the change in horizontal and vertical direction. In this report, the term "Bend" will refer to the vertical directional change and will be replaced with "Apex" as it describes the highest point in a section of a pipeline, where air will accumulate due to buoyancy forces.

Energy loss in pipelines can be contributed to a number of physical factors of which friction at the pipe wall is the most important. Literature refers to energy loss at transitions, junctions, valves and apex points as secondary or minor losses.

Actual energy loss caused by localised air at apex points of pipelines, as well as the size of the localized air bubble have not been well documented.

As a water column moves through an apex point, centrifugal forces on fluid particles will cause a change in the flow pattern. This directional change causes turbulence, momentum transfer resulting in a loss of energy from the system. Air is constantly transported in solution and with a decrease in pressure, air can be released from solution. Air in a pipeline then congregates at the apex point and as more air is released from solution, a visible air bubble will form. The bubble is kept in place at the apex, due to buoyancy forces, and as more air is introduced, the cross-sectional area of the pipe decreases due to the growth of the bubble, whilst the velocity increases.

This phenomenon decreases the capacity of the pipe and the result is a lower discharge and loss of revenue due to higher costs associated with longer pump cycles.

4.5.2 Objective of the experimental work

The experimental investigation focused on the following aspects:

- Secondary energy losses caused by various vertical angles at the apex points;
- Secondary energy losses caused by various air bubble sizes trapped at the apex point;
- Determination of the behaviour of various bubble sizes at the apex for different apex angles and velocities;
- Deriving secondary loss coefficients, K-values, to be able to calculate the secondary energy losses using the standard formulae.

4.5.3 Methodology

A 110 mm diameter transparent uPVC pipe was used in the experiment. The pipe was cut to form an angle that could be used to simulate an apex point. The apex angle that was identified for investigation was $11\frac{1}{4}^{\circ}$.

A steel frame was constructed and 110 mm diameter pipe saddles were bolted to the frame. The pipe was fixed in the required position with the use of the pipe saddles. This set-up made it possible to pivot the pipe around the apex, thus simulating typical installations. Illustrated in **Figure 4.5.1** reflects a section of the set-up showing the position where the pressure transducer have been installed and the arrangement to change the slope of the system.



Figure 4.5.1: Typical set-up of experiment

Figure 4.5.2 provides the detail pf the reference points and layout of the experimental set-up.

The following equipment was used in the experiments:

- Pressure Transducers (PT)
- Data Acquisition Equipment (DAE)
- Air Induction Box (AIB)
- Ultrasonic Flow Meter (UFM)

Tests were carried out in three stages. The first was to determine secondary losses due to an apex alone without any air at the apex. The second stage was to add an air bubble, with the aid of the AIB, and to measure the secondary energy loss across the bubble. A stationary situation was created in the third stage, where the flow was stopped and the pressure measured. This was done to be able to relate the pressures from the different pressure transducers to one another. At stages one and two, pressure losses were measured at the up- and downstream pressure transducers (PT's) and velocities were varied between 0.2 and 1.1 m/s to obtain a representative range of results. The pressure transducer (PT) output signal was captured through the data acquisition system (DAE), which records the pressures at a rate of 1000 readings per second over a period of 10 seconds. The velocity is recorded manually with the aid of the UFM and controlled by isolating valves situated up- and downstream of the pipe. The bubble behaviour could also be recorded with the varied velocity. For every measured velocity, three sets of readings were taken to ensure that measuring errors could be identified and discarded from the data sets.

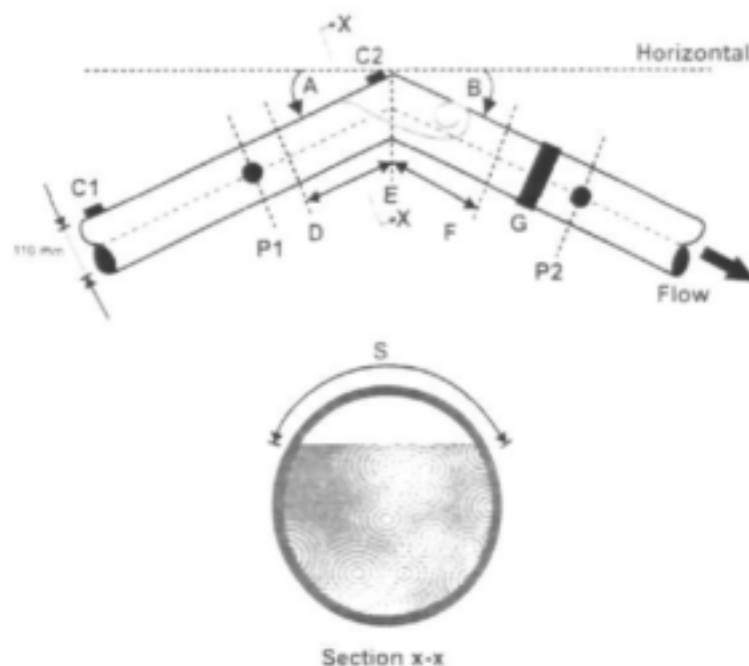


Figure 4.5.2: Schematic layout of the pipe, reflecting the different elements of the set-up

4.5.4 Experimental set-up

The experiments were performed at the Department of Water Affairs and Forestry Laboratories in Pretoria West. A constant head tank receives water from pumps. The water from the constant head tank is transported through a pipe system to the experimental set-up. Water flows from the experimental set-up into the pump sump thus creating a closed system. **Figure 4.5.3** contains the schematic layout of the experimental set-up.

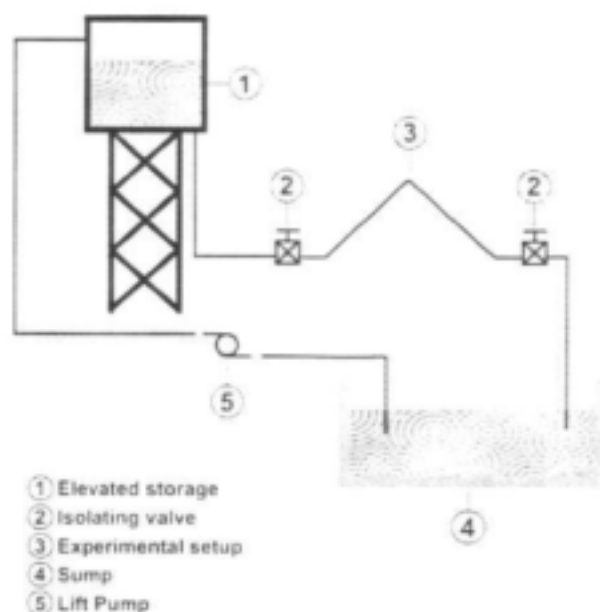


Figure 4.5.3: Schematic layout of the experimental set-up

Two pressure transducers were used to record the pressures upstream and downstream of the apex point of node 3 in **Figure 4.5.3**. The details of the transducers are given in **Table 4.5.1** below. The relationship, Equation 4.5.1 for the determination of the pressure is:

$$P = aV + b \dots\dots\dots (4.5.1)$$

Where:

- P = pressure (m)
- V = Volt reading from the pressure transducer (volt)
- a, b = Calibration constants.

Table 4.5.1: Details of the pressure transducers

Position of transducer	Transducer		Calibration constants	
	Pressure range	ID	a	b
Upstream from apex	1.6 BarA	PT No. 9	0.2	-0.4
Downstream from apex		PT No. 10	0.2	-0.4

4.5.5 Results obtained from the experimental work

4.5.5.1 Results for the experimental set-up with an angle of 11,25° at the Apex

4.5.5.1.1 Secondary losses due to Apex – 11,25 ° - No air in the system

The first experimental set-up was on the 11,25° Apex. The upstream section had a 0,0° gradient and this resulted in an 11,25° gradient on the downstream section.

The first step was to measure the secondary losses through the apex without any air and derive a relationship for the energy loss between the measuring points from the measured data. For each flow rate 3 measurements were taken and the averaged values were calculated.

The averaged results for the 11,25° angle set-up are reflected in **Table 4.5.2** and graphically presented in **Figure 4.5.4**.

Table 4.5.2: Averaged results for the evaluation of a 11,25° angle at the APEX – No air bubble

Test *	Velocity (m/s)	Secondary loss ΔP (mm)
1	0,284	4,629
2	0,287	3,225
3	0,754	18,153
4	0,747	16,519
5	0,971	29,858
6	0,967	30,068
7	1,119	45,715
8	1,129	42,164

Note: * Average values of three tests

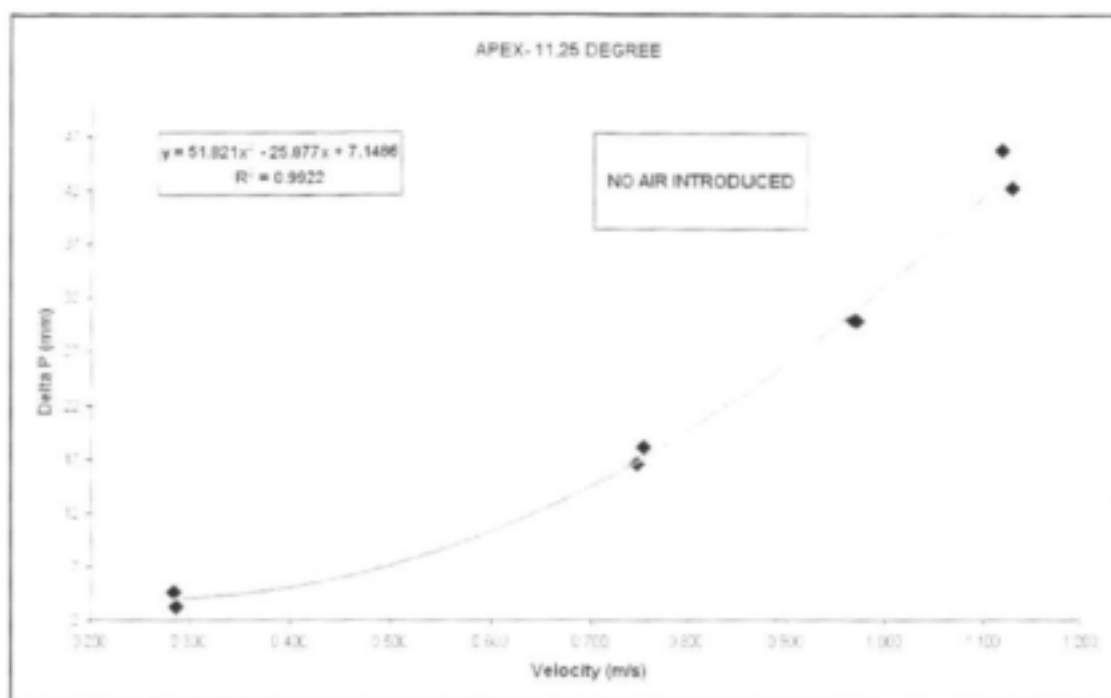


Figure 4.5.4: Secondary energy loss across the 11.25° apex with an upstream horizontal leg (No air in the system)

The formula derived from the experimental data is as follows:

$$\Delta P = 51.821V^2 - 25.877V + 7.1486 \quad \text{Accuracy : } R^2 = 0.9922 \dots \dots \dots (4.5.2)$$

Where:

- ΔP = Energy loss (mm)
 V = Flow velocity (m/s)

4.5.5.1.2 Secondary losses at Apex with an air bubble introduced – 11.25°

Air was added in the second step with the use of the AIB. The air bubble moved upstream and formed a long flat slug of air at low velocities. The air that was introduced measured with the AIB. The results of the second set-up are summarised in **Table 4.5.3**.

Table 4.5.3: Averaged results for the evaluation of a 11,25° angle at the APEX – With an air bubble

Test	Velocity (m/s)	Measured (S) (mm)	Measured (D-E and E-F) L_B (mm)	Secondary loss ΔP (mm)	Comments
1	0.342	85	850	34.237	Air bubble moving upstream. Air volume 1,919 litres
2	0.342	65	825	27.669	Air bubble moving upstream. Air volume 1,416 litres
3	0.491	65	825	38.585	
4	0.495	65	825	30.711	
5	0.387	170	1400	99.922	Air bubble moving upstream. Air volume 6,283 litres
6	0.388	220	825	91.246	Air bubble stabilised upstream of point P1. Air volume = 13,5 litres Observed a hydraulic jump.

With the upstream section horizontal and a flow velocity less than 0,495 m/s, the air bubble tends to move upstream. In these cases, the length of the air bubble depends on the air volume, flow velocity and pressure within the pipe.

In tests 5 and 6, a small amount of air was transported downstream.

At a velocity of less than 0,6 m/s, the air bubble was stable at the apex point, although occasionally small quantity of air was, transported downstream. At a velocity of about 0,68 m/s, the air bubble eventually started to move downstream, until a point was reached where buoyancy forces were greater than the drag force of the water on the air bubble and the bubble started to move upstream again. This indicated that the **flow velocity should be larger than 0,6 m/s for a slope of 11,25°** to transport the air bubble hydraulically. This aspect is discussed in more detail in Section 4.2 of this report.

In test 6 a hydraulic jump was observed at the tail of the air bubble. In this test, the induced air bubble had a volume of about 13.5 litres. The shape of the jump can be described as follows:

- Length of air bubble was measured at two sections (Refer to Figure 4.5.2).
D-E = 825 mm and E-F = 1500 mm.

Two circumferential measurements (S) of the bubble were taken downstream of the air inlet C2, respectively 300 mm and 1500 mm downstream from C2. At these positions the circumferential dimensions was S1=210 mm and S2= 230 mm.

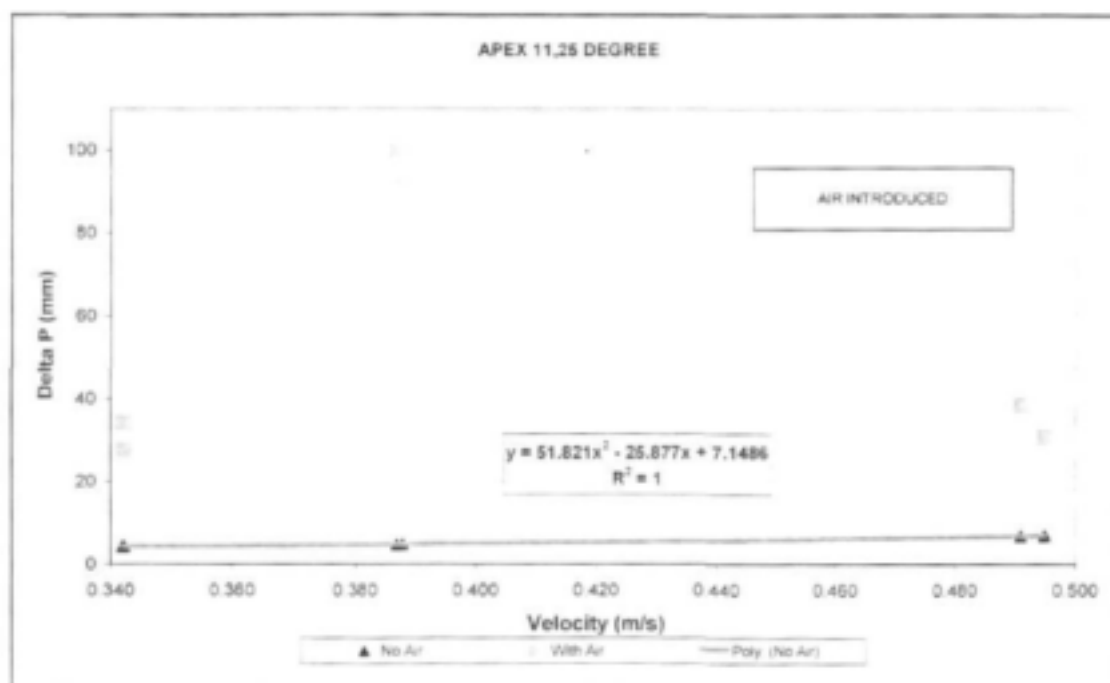


Figure 4.5.5: Secondary energy loss across the 11.25° with and without air

Figure 4.5.5 illustrates the difference in the pressure drop across the 11.25° bend for the two tests, one with no air bubble and the other with an air bubble. The measured pressure drop is compared with the calculated pressure drop for a similar flow rate, using the relationship expressed as Equation 4.5.2. The size of the air bubble influences the secondary loss, which was measured between 27 and 100 mm. Because the form and size of the air bubble influenced the pressure drop, it should be reflected in the results. The difference in the secondary loss between the tests was defined as the contribution from the air bubble. The secondary loss associated with the air bubble was then used to back-calculate the secondary loss coefficient, k .

$$h = k \frac{v^2}{2g}$$

The results are summarised in Table 4.5.4.

Table 4.5.4: Results for energy loss associated with a 11,25° angle at the APEX – With and without an air bubble

Test	Velocity (m/s)	Secondary loss measured ΔP (mm)		B-A (m)	k-Value
		No air (A)	With air (B)		
1	0,342	4,359	34,237	0,0299	5,011
2	0,342	4,360	27,669	0,0233	3,907
3	0,491	6,936	38,585	0,0316	2,575
4	0,495	7,036	30,711	0,0237	1,895
5	0,387	4,895	99,922	0,0950	12,448
6	0,388	4,909	91,246	0,0863	11,252

The results of the calculated secondary loss coefficients, k, are illustrated in Figure 4.5.6.

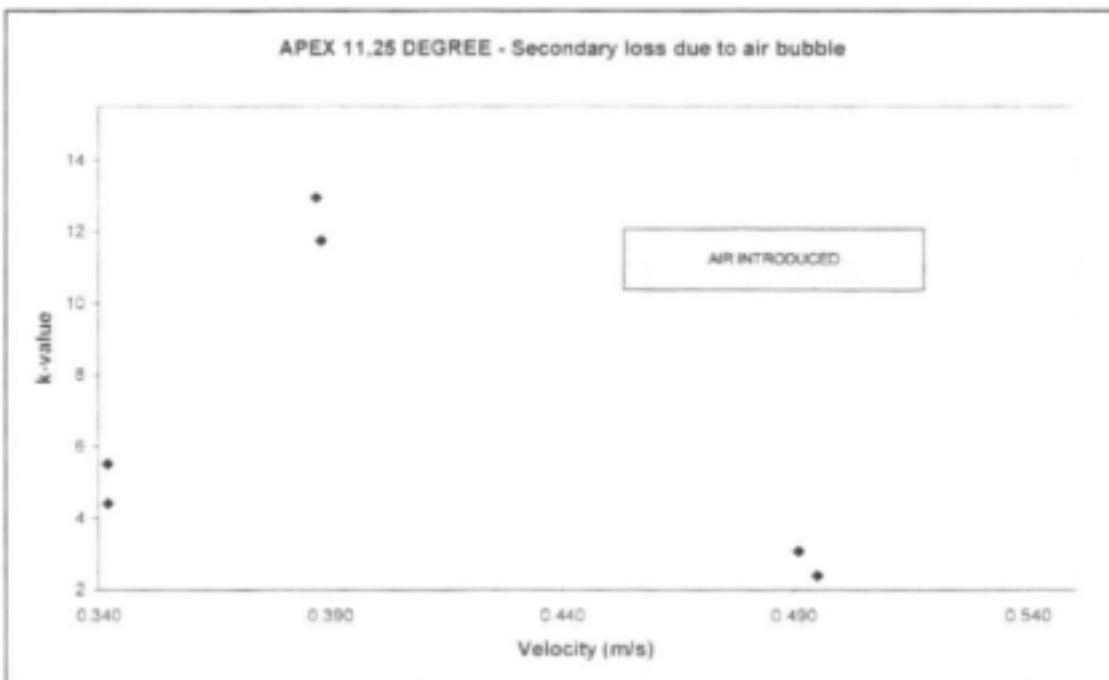


Figure 4.5.6: Graphical presentation of the calculated secondary loss coefficient, k, for an angle of 11,25° at the apex, due to an air bubble

The relationship between air volume and secondary energy loss are shown in **Figures 4.5.7 to 4.5.10**. From **Figure 4.5.7** it is observed that the bigger the bubble volume the higher the secondary loss.

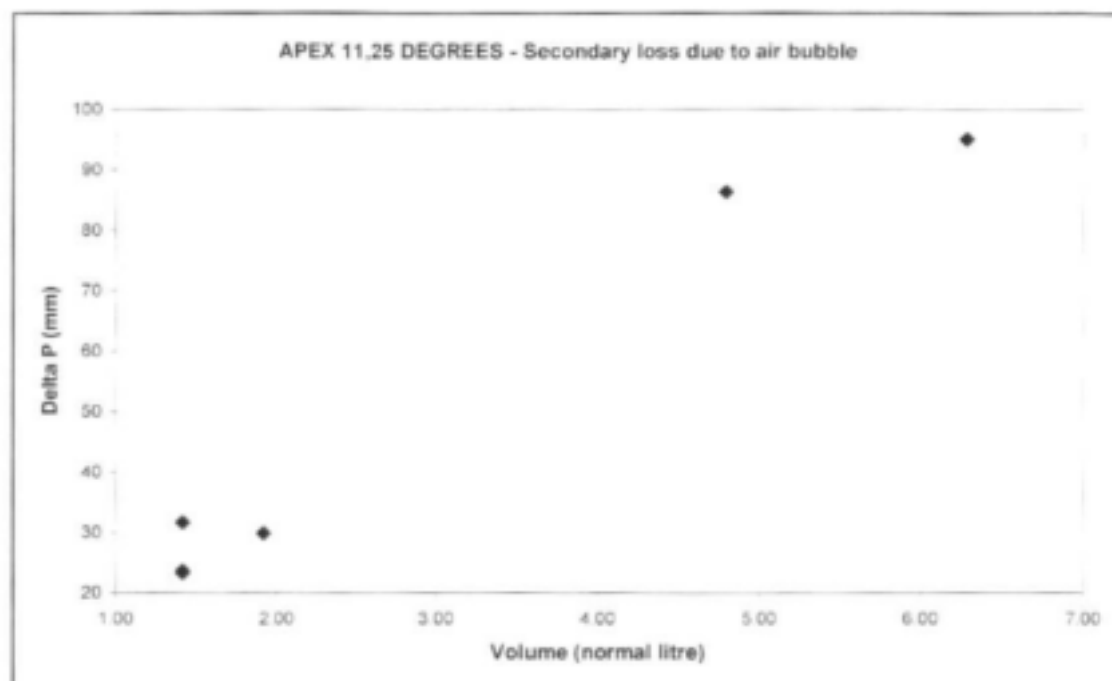


Figure 4.5.7: Graphical presentation of the measured secondary energy loss against bubble volume

In **Figure 4.5.8** it can be observed that at different velocities, bubbles of equal volume may cause different secondary energy losses. The cause of this relationship could be that at the higher velocity, the form of the bubble changes due to the higher shear on the bubble.



Figure 4.5.8: Graphical presentation of bubble volume measured against secondary energy loss

The air bubble's shape was measured in terms of circumference dimension (S) and length, (L_B) (**Figure 4.5.2**) which are included in the graphical presentation to establish the influence of the air volume characteristic on the air bubble. In **Figure 4.5.9** the

$$\sqrt{S \times L_B}$$

which represents the bubble volume characteristic, is shown, while in **Figure 4.5.10** the air bubble volume characteristic is presented by

$$\sqrt{S \times L_B} L_B$$

to accentuate the influence of the bubble's length, L_B .

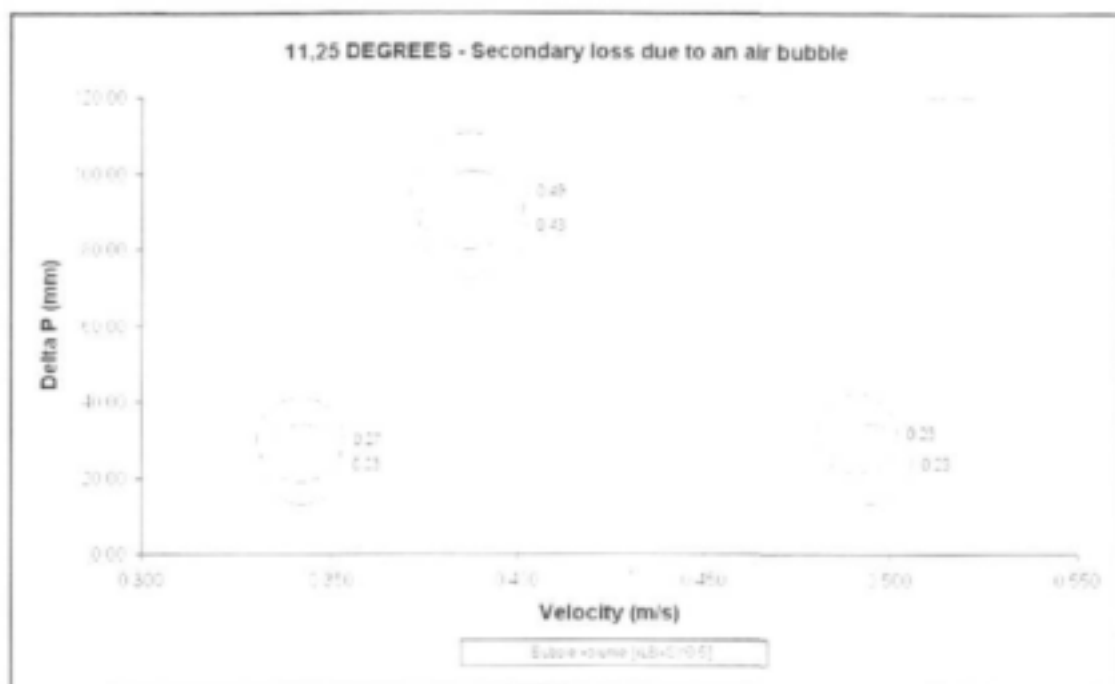


Figure 4.5.9: Graphical presentation of bubble volume represented by $(S * L_B)^{0.5}$ against secondary energy loss

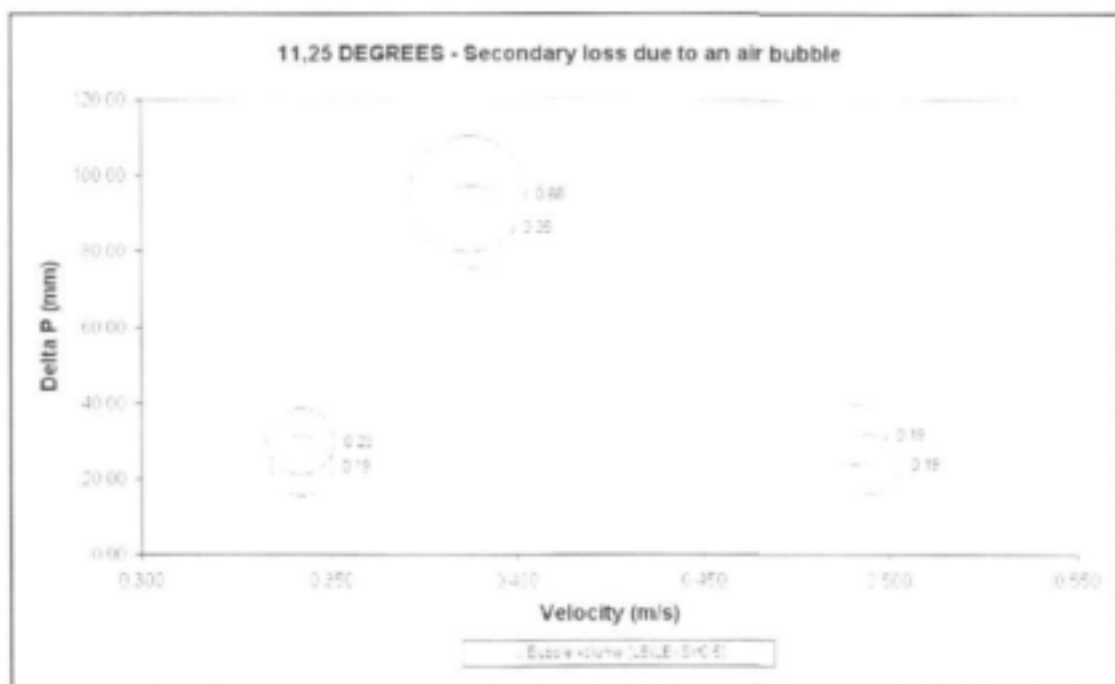


Figure 4.5.10: Graphical presentation of bubble volume represented by $(S * L_B)^{0.5} * L_B$ against secondary energy loss

The results did not provide any conclusive relationship between the energy loss, velocity and the air bubble size, due mainly to the dynamics of the air bubble's form.

It is however clear that larger air bubbles, result in a larger secondary losses and that the cross sectional area (presented by S) of the air bubble has a bigger influence on the energy loss in comparison to the length of the bubble, L_B .

4.5.5.1.3 Secondary losses due to specific air bubble size

The secondary loss caused by an air bubble at an apex was further investigated by establishing the loss associated with a stationary air bubble of a specific constant air volume. The set-up was tested for an upstream slope of 5° and an angle of 11.25° . A "stable air bubble" was defined as the air bubble that will remain in position at the specific flow velocity and system layout. This is associated with the condition that no air is progressing downstream or upstream from the bubble. For the various tests, the velocity was kept constant, while the air bubble size was increased with the objective to define the maximum air bubble size and the associated energy loss. Air was introduced at point C2 with the use of the AIB. The bubble shape was recorded. The results are summarised in **Table 4.5.5**.

Table 4.5.5: Results for the secondary energy loss associated with an 11.25° angle at the APEX and a constant velocity and different air bubble sizes

Test	Velocity (m/s)	Measured (S) (mm)	Measured (D-E and E-F) L_B (mm)	Secondary loss ΔP (mm)	Comments
1	0.789	150	340	50.400	Air bubble stable at apex. Air volume 1.346 litre
2	0.793	150	340	52.531	
3	0.796	150	340	54.339	
4	0.781	140	200	33.084	Air bubble stable at apex. Air volume 0.739 litre
5	0.780	140	200	31.933	
6 to 9	Data rejected				
10	0.778	120	140	20.123	Air bubble stable at apex. Air volume 0.443 litre

These results were plotted on the same graph as the results obtained for the losses without the air bubble, shown in **Figure 4.5.5** and reflected in **Table 4.5.6**.

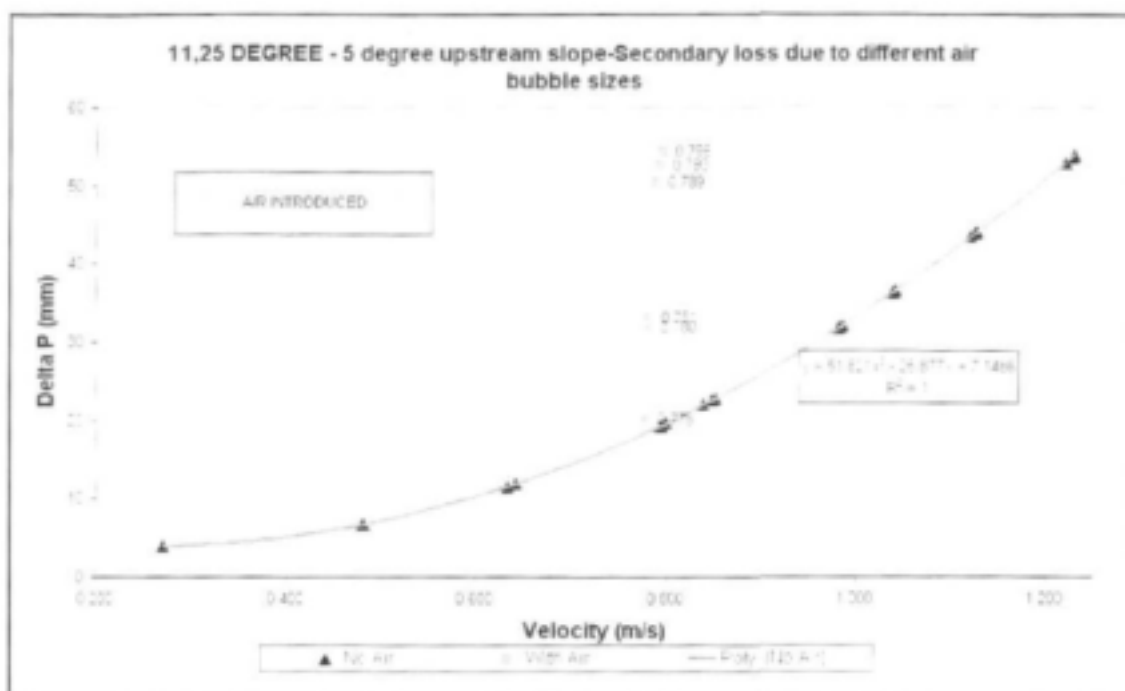


Figure 4.5.11: Secondary losses for different air bubble sizes at a constant velocity

Table 4.5.6: Results for energy loss associated with an 11,25° angle at the APEX – With and without an air bubble. Upstream leg at a positive 5° slope

Test	Velocity (m/s)	Secondary loss ΔP (mm)		Hf=(B - A) (m)	k-Value
		No air (A)	With air (B)		
1	0.789	18.99	50.002	0.0310	1.003
2	0.793	19.215	52.531	0.0333	1.039
3	0.796	19.385	54.339	0.0349	1.082
4	0.781	18.547	33.084	0.0145	0.467
5	0.780	18.492	31.933	0.0134	0.433
6*	0.778	18.382	16.196	-0.0022	0.0
7*	0.778	18.404	16.596	-0.0018	-0.0580
8*	0.775	18.218	15.960	-0.0023	-0.0730
9*	0.783	18.658	18.029	-0.0006	-0.0200
10	0.778	18.382	20.123	0.0017	0.056

Note: * These test data were discarded – possible data errors

The relationship between secondary energy loss, velocity and bubble volume was observed and illustrated in **Figure 4.5.12**. At a constant bubble volume, the increase in velocity resulted in higher secondary energy losses.

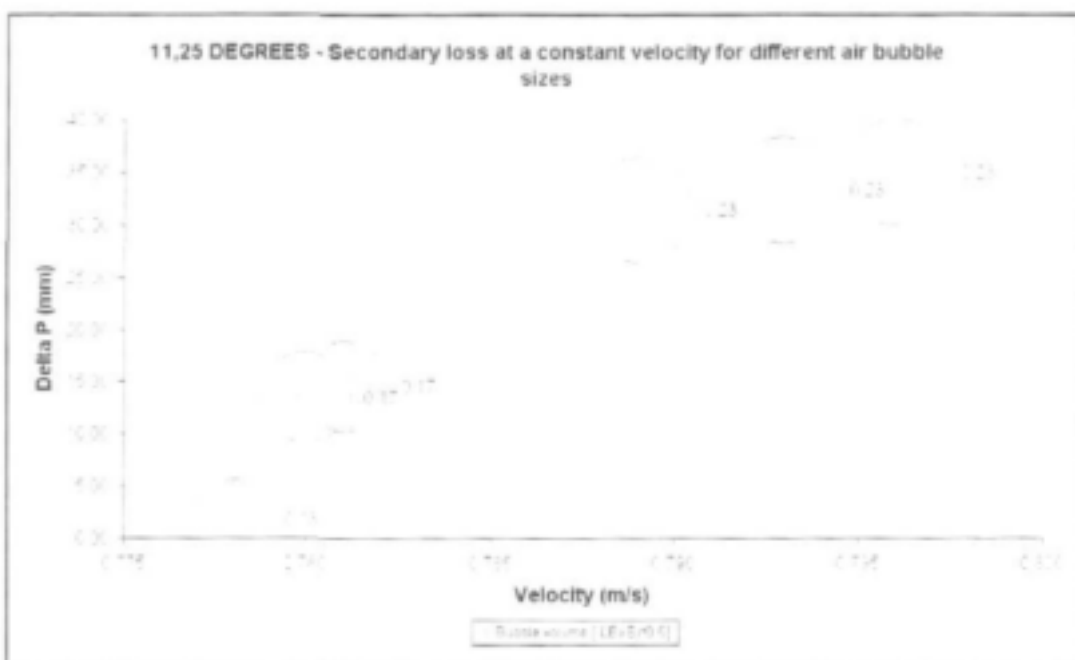


Figure 4.5.12: Graphical presentation of bubble shape and velocity measured against secondary energy loss

The data is also presented to obtain a relationship between ΔP and the air bubble size (volume).

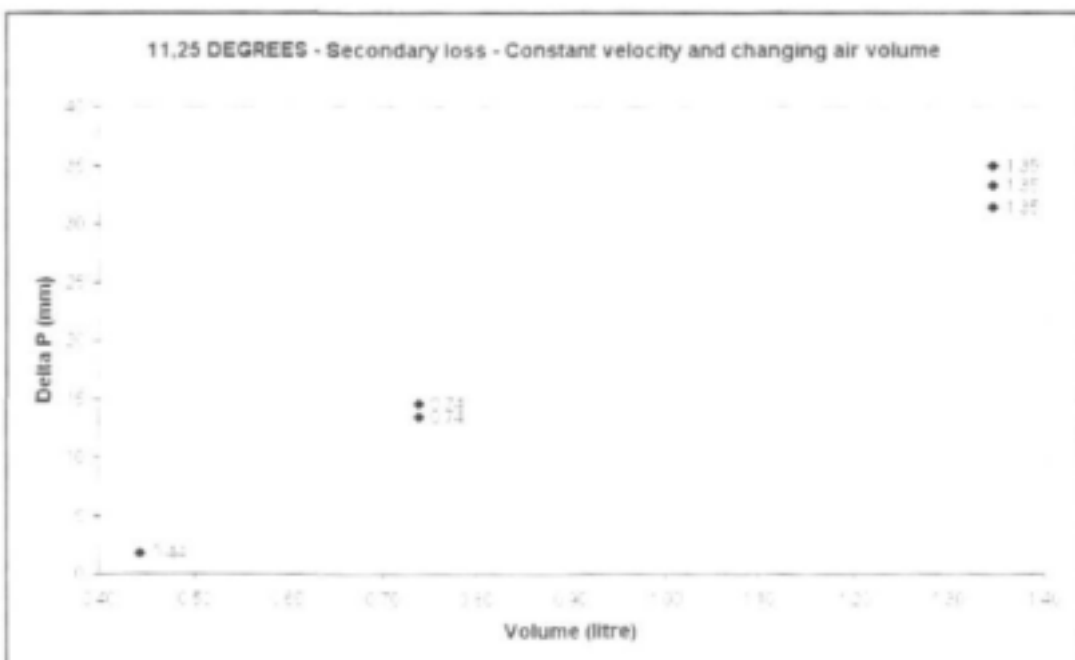


Figure 4.5.13: Graphical presentation of the secondary energy loss at a constant velocity for different air bubble volumes

The increase in bubble size resulted in a higher secondary loss as is illustrated in Figure 4.5.13.

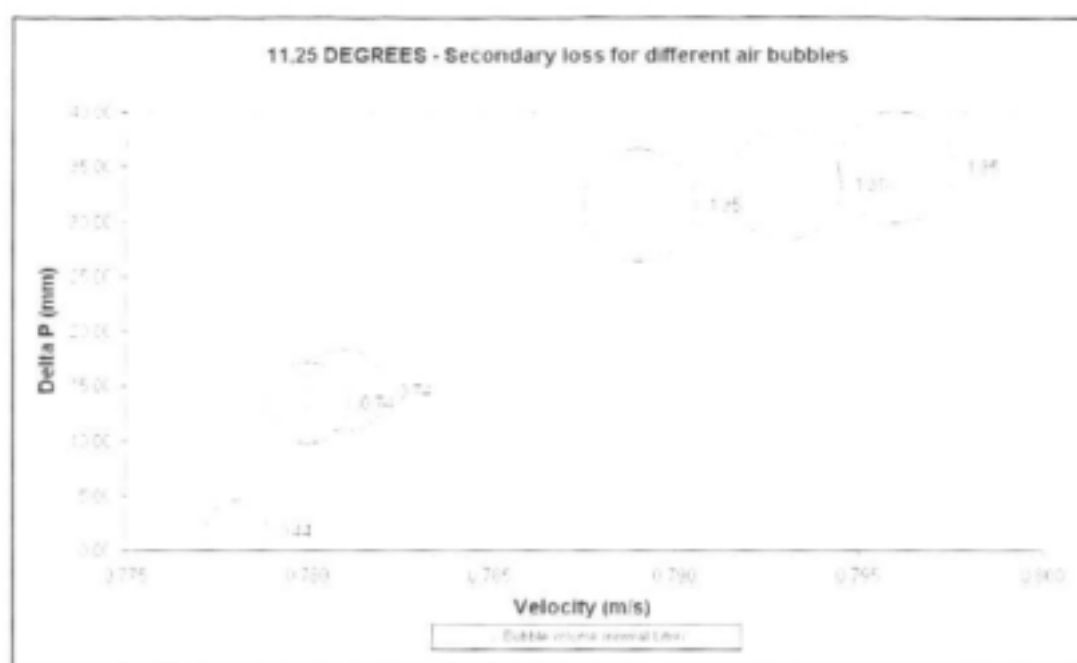


Figure 4.5.14: Graphical presentation of bubble shape measured against secondary energy loss

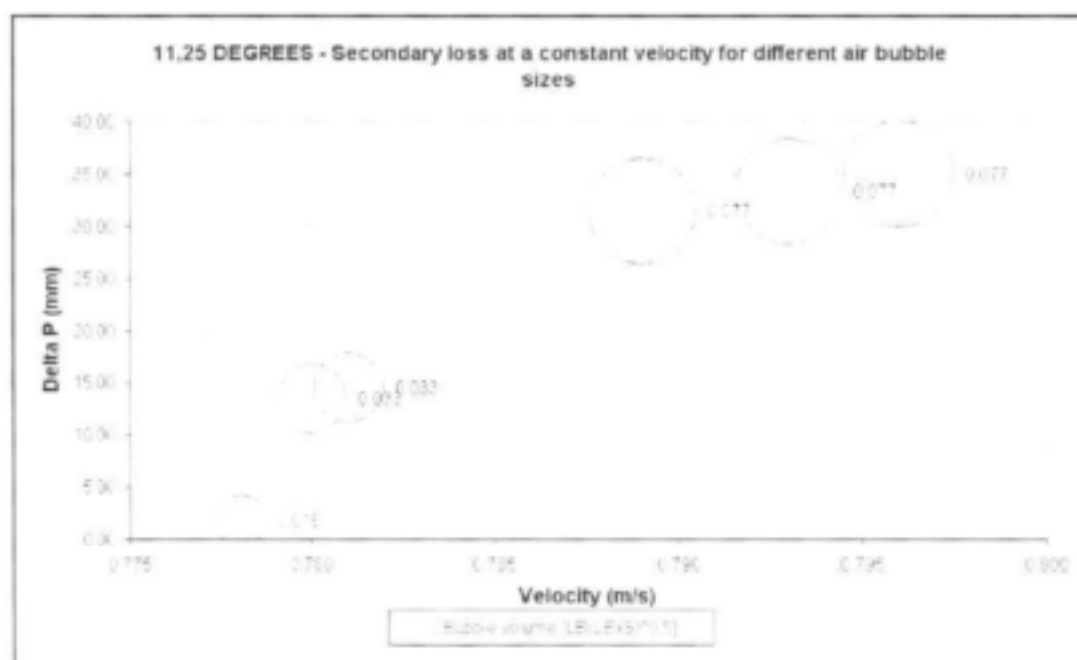


Figure 4.5.15: Graphical presentation of bubble shape measured against secondary energy loss

Figure 4.5.15 provides a clear relationship between the secondary loss and the bubble size as represented by the characteristic, $L_B \sqrt{L_B \times S}$

4.5.5.1.4 Secondary losses due to artificial air bubble – 11,25°

Due to the difficulty of generating a stable air bubble at high velocities, an artificial bubble was used to record the secondary losses. The artificial bubble was inserted through point C1 and kept in place with the use of a fish line that was secured at C1. The bubble was positioned at the apex point. Although the air volume was constant, the shape changed considerably. The results are summarised in Table 4.5.7.

Table 4.5.7: Results for the evaluation of an 11,25° angle at the APEX – With an artificial air bubble. Upstream leg at a positive 5° slope

Test	Velocity (m/s)	Secondary loss ΔP (mm)	Comments
1	0,491	96,555	Air bubble shape : long and stretched out Air volume 3.517litre
2	0,488	99,632	
3	0,495	102,974	
4	0,331	38,337	
5	0,330	41,876	
6	0,331	46,459	
8	0,838	119,505	Air bubble shape: long and stretched out Air volume 3.517litre
9	0,839	121,165	
10 to 12	Data rejected		
13	0,929	126,252	Air bubble shape long and stretched out Air volume 3.517litre
14	0,931	125,468	
15	0,931	118,436	
16	1,114	154,849	
17	1,109	150,045	
18	1,100	145,009	
19	1,360	177,755	
20	1,358	178,508	
21	1,358	178,290	
22	1,455	193,092	
23	1,455	195,288	
24	1,457	194,640	
25	1,552	195,094	Air bubble shape stretched out- 420mm Air volume 3.517litre
26	1,544	190,555	
27	1,541	196,152	
28	0,609	54,479	
29	0,606	56,416	
30	0,607	57,494	

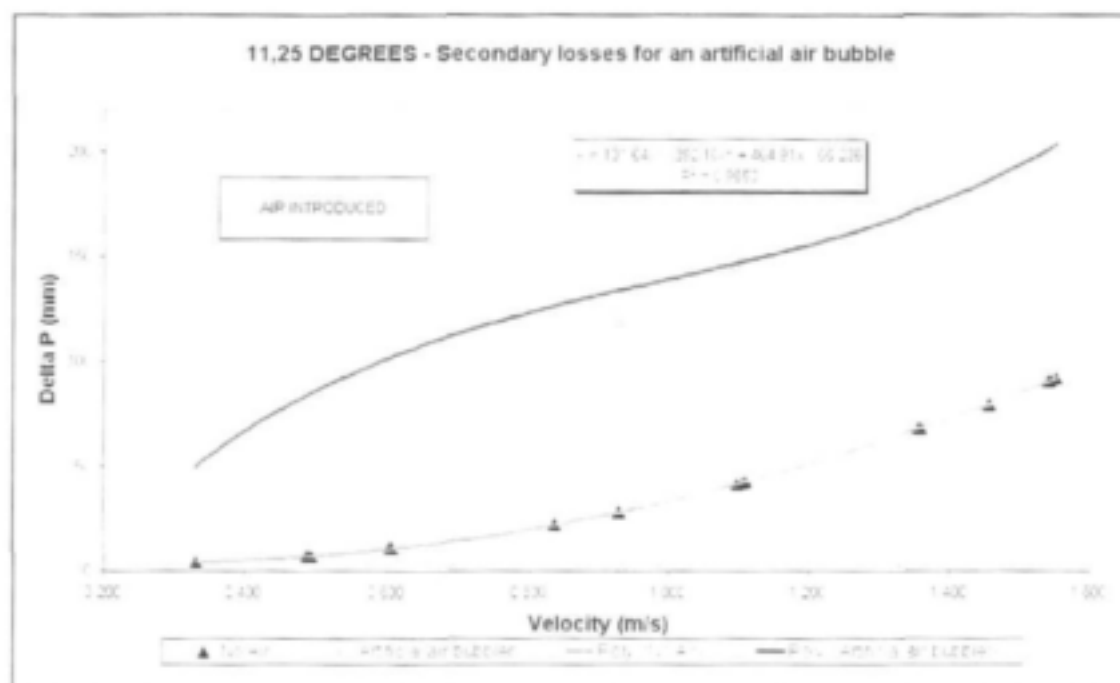


Figure 4.5.16: Secondary energy loss across the 11,25° bend with an artificial air bubble and without air

With the artificial air bubble, test velocities could be varied between 0.6 and 1.6 m/s. Secondary losses were measured and compared with the results for the set-up without the air bubble.

Table 4.5.8: Results for energy loss associated with an 11,25° angle at the APEX – With an artificial air bubble and the losses without an air bubble

Test	Velocity (m/s)	Secondary loss ΔP (mm)		$H_f = (B - A)$ (m)	k-Value
		No air (A)	With air (B)		
1	0.491	6.936	96.555	0.090	7.294
2	0.488	6.861	99.632	0.093	7.643
3	0.495	7.036	102.974	0.096	7.682
4	0.331	4.260	38.3365	0.034	6.102
5	0.330	4.252	41.876	0.038	6.779
6	0.331	4.260	46.459	0.042	7.557
8	0.838	21.854	119.505	0.098	2.728
9	0.839	21.915	121.165	0.099	2.766
10 to 12	Data rejected				
13	0.929	27.832	126.252	0.098	2.237
14	0.931	27.973	125.468	0.097	2.207
15	0.931	27.973	118.436	0.090	2.048
16	1.114	42.631	154.849	0.112	1.774
17	1.109	42.184	150.045	0.107	1.721
18	1.100	41.387	145.009	0.103	1.680
19	1.360	67.804	177.755	0.110	1.166
20	1.358	67.574	178.508	0.110	1.180
21	1.358	67.574	178.290	0.110	1.178
22	1.455	79.203	193.092	0.113	1.055
23	1.455	79.203	195.288	0.116	1.076
24	1.457	79.453	194.640	0.115	1.065
25	1.552	91.808	195.094	0.103	0.841
26	1.544	90.732	190.555	0.099	0.822
27	1.541	90.330	196.152	0.105	0.874
28	0.609	10.608	54.479	0.043	2.321
29	0.606	10.497	56.416	0.045	2.453
30	0.607	10.534	57.494	0.047	2.501

With the high velocities, the shape of the artificial bubble changed due to drag forces. The artificial bubble initially had a spherical shape, but as the velocity increased it changed, to a long flat shape with some air at the end of the bubble. The shape change caused a **decrease in the drag forces** and resulted in the decrease of the secondary loss coefficient, k (Table 4.5.8 and Figure 4.5.17).

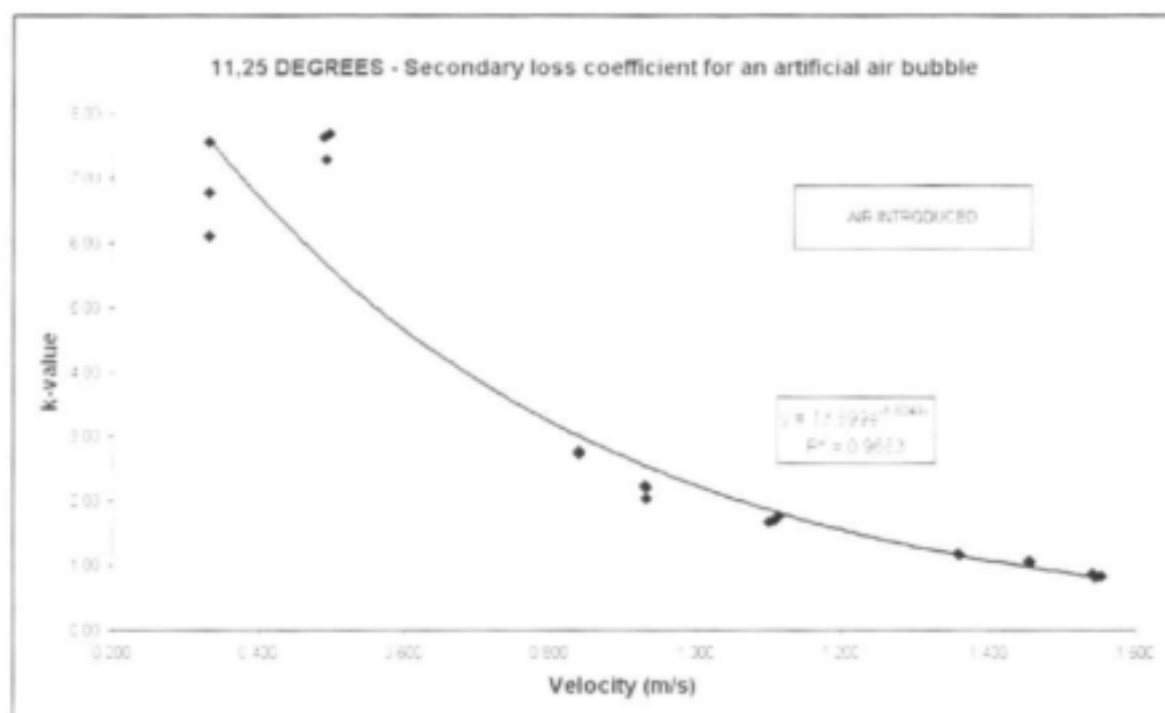


Figure 4.5.17: Graphical presentation of the calculated secondary loss coefficient, k for an angle of $11,25^\circ$ with an artificial air bubble

The increase in velocity and the application of a constant bubble volume resulted in a small increase in secondary energy loss. This was observed between flow velocities 0,6 and 1,0 m/s. and again between 1,2 and 1,6 m/s. The flow velocity influenced the shape of the air bubble. The shape changed from pear shaped to an elongated form, as the velocity increased.

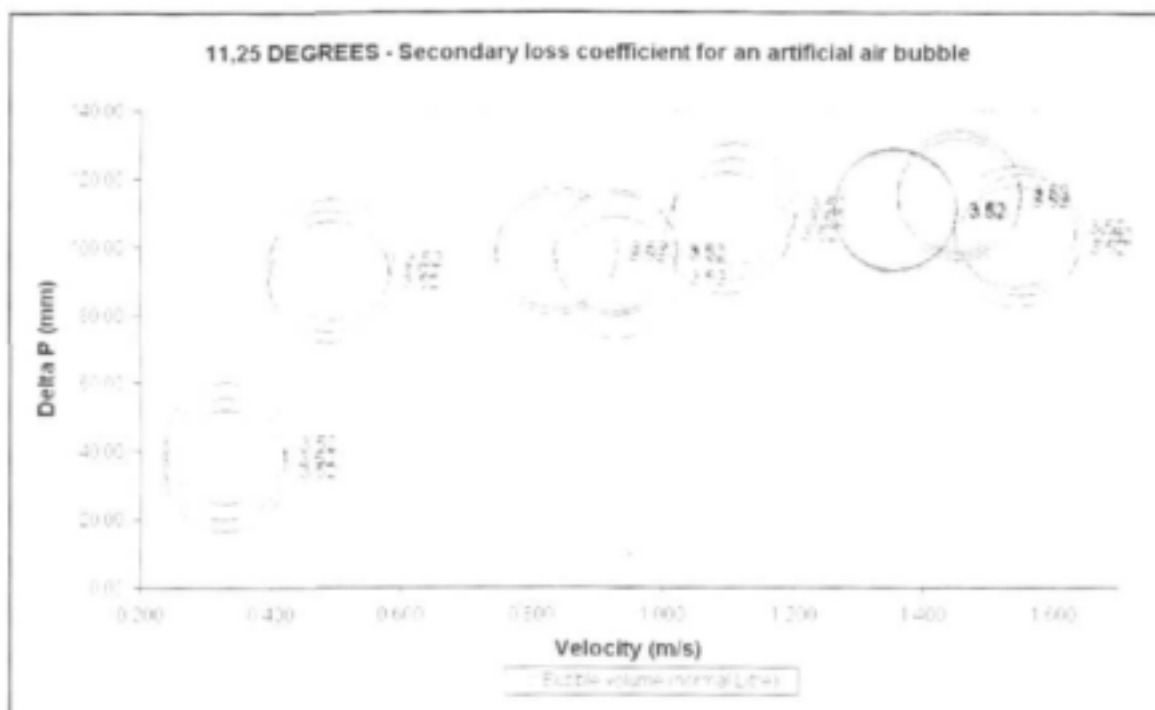


Figure 4.5.18: Graphical presentation of the secondary energy loss as function of the flow velocity for a given air bubble volume (artificial air bubble)

4.5.6 Conclusions and recommendations

The influence of trapped air at apex points on the energy loss is substantial and needs to be taken into consideration when designing pipelines. Based on the results, there are a number of factors that must be highlighted in the calculations of secondary pressure loss due to the presence of air:

- Residual pressure in the pipe will determine the compression of the air bubble, and will hence influence the energy loss.
- The energy loss is proportional to the bubble size for a specific flow velocity and the secondary loss coefficient, k is inversely proportional to the flow velocity. **This finding suggests that the influence of the cross sectional decrease in the flow area, contributes profoundly more to the loss than the length of a bubble (same volume).**
- The flow velocity is a crucial parameter for the stable size air bubble that will form at the apex.

- The angle at the apex and the angles of the up- and downstream sections influence the size of the air bubble that will form. The steeper the downstream pipe slope for a given angle the larger the air bubble will be for a specific flow velocity and hence the larger the secondary loss will be.

From these results it is impossible to provide an exact relationship for the calculation of secondary losses due to captured air bubbles at apex point. The experimental results however highlighted that it is possible to have a secondary loss coefficient as high as 7. The application of this in practice will require an assessment of the system for secondary loss coefficients of 7 and higher for steeper downward slopes and lower velocities than the clearing velocity.

5. NUMERICAL MODELING OF AIR RELEASE FROM PIPELINES (COMPUTATIONAL FLUID DYNAMICS)

5.1 Introduction

It was determined in the experimental work and backed by field investigations, that the relationship between the air valve off-take diameter and the pipe diameter influences the ability to de-aerate a pipeline. Parameters that influence the effectiveness of the air valve off-take include the pipe diameters of the off-take and the conveying pipe as well as flow velocity, air pocket size and the slope of the pipe (inclination).

The objective of numerical modelling was to determine the applicability of Computational Fluid Dynamics (CFD) in the optimisation of the design of the air off-take on a water pipeline.

The aim was to investigate the effectiveness of the CFD code used, in modelling the motion and the capturing of the air pocket in a pipeline for three different inclinations. Input from personnel at cfdlab, University of Pretoria, was obtained during this part of the research (Danie de Kock (MEng) and Thomas Kingsley (BEng)).

5.2 Problem description

Figure 5.1 depicts the physical parameters of a typical air valve installation on a pipeline. The variables indicated will be parameterized allowing (within certain limits) the ability to create answers for different operating parameters.

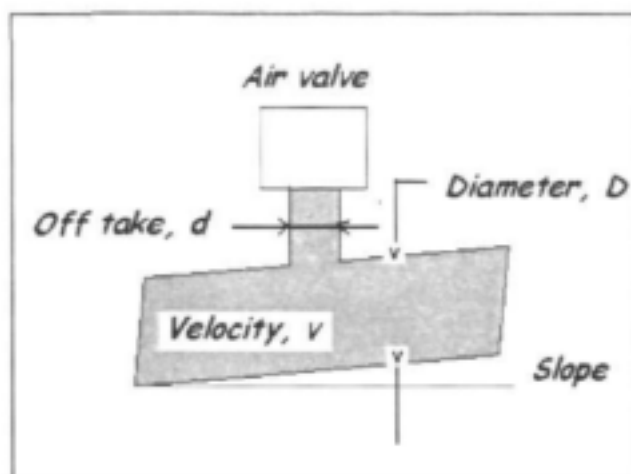


Figure 5.1: Typical air valve installation

5.3 Software selection

FLUENT provides comprehensive modelling capabilities for a wide range of incompressible and compressible, laminar and turbulent fluid flow problems. For all flows, FLUENT solves conservation equations for mass and momentum. For flows involving heat transfer or compressibility, an additional equation for energy conservation is solved. Discretized forms of the Navier-Stokes equations are solved over each grid cell.

In addition to the models available in FLUENT, further models are required for modelling the effects of turbulent fluctuations of velocity and other effects due to turbulence. In this case, the Reynolds Averaged Navier-Stokes equations (RANS) are solved with the $k-\epsilon$ model providing turbulent computational capacity. This turbulence model uses the eddy-viscosity approach in which the Reynolds stresses are assumed to be proportional to the mean velocity gradients, with the constant of proportionality being the turbulent viscosity.

Finally, since the study in question contains the added problem of multi-phase flow, a further multi-phase flow model must be included in the modelling. The specific model used is the Volume of Fluids model (VOF). The VOF formulation relies on the fact that two or more fluids (or phases) are not interpenetrating. For each additional phase added to the model, a variable is introduced: the volume fraction of the phase in the computational cell. In each control volume, the volume fractions of all phases add to unity. The fields for all variables and properties are shared by the phases and represent volume-averaged values, as long as the volume fraction of each of the phases is known at each location. Thus the variables and properties in any given cell are either purely representative of one of the phases, or representative of a mixture of the phases, depending upon the volume fraction values.

5.4 CFD Modeling

5.4.1 Geometry building

The geometry of a section of pipeline and the connected air valve was generated in FLUENT's GAMBIT (Gambit, 2001) pre-processor. The geometry can be seen in **Figure 5.2** below, where a smaller pipe was connected to the larger pipeline with a small box at the termination of the small pipe to contain any captured air pockets. It was assumed that the model has a symmetry boundary, and swirl effects were assumed to be negligible.

The geometry in **Figure 5.2** represents 1 000 mm of pipeline length with a pipeline diameter of 110 mm and a diameter ratio of 0.35 between the small pipe and the pipeline. The discontinuity was offset to the right of the entrance of the pipeline. GAMBIT uses a journal file, which consists of a string of command line inputs to define how the geometry of the model will be set up.

Input variables include the diameters of the pipeline and the small pipe, the offset from the centre of the discontinuity, the pipe lengths, and a bubble size parameter. This last parameter determines the area (as a fraction of the total pipeline diameter) at the inlet that will allow air into the system for a given period.

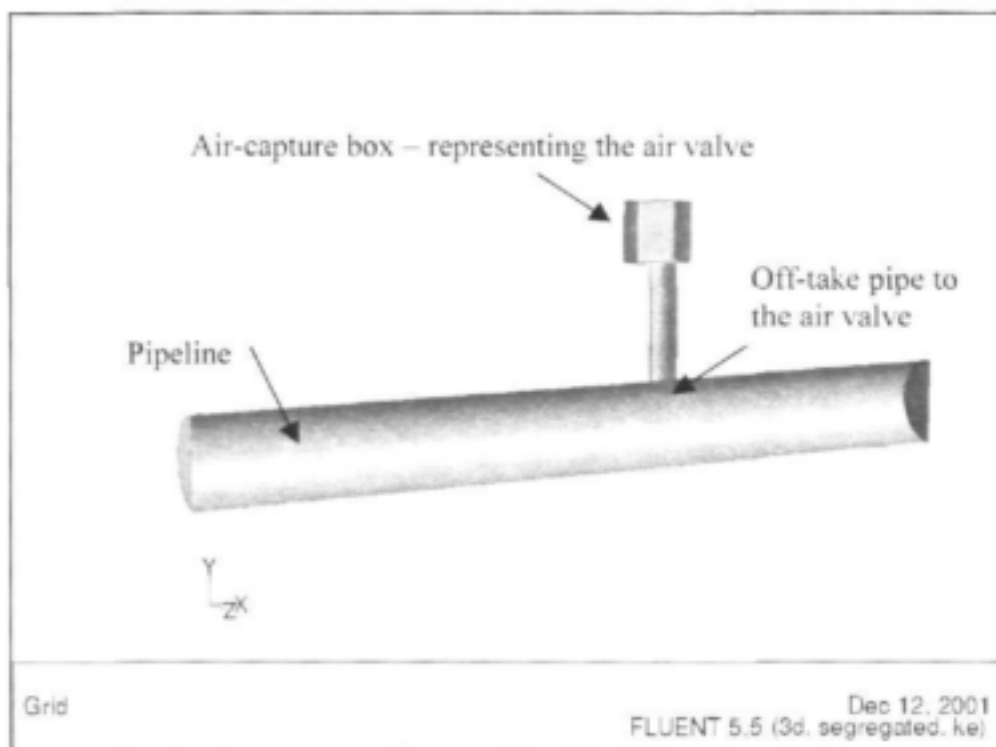


Figure 5.2: Model Geometry

5.4.2 Grid Generation

Sections of the journal file will prompt GAMBIT to divide the volume generated into a grid. **Figure 5.3** below shows the layout of the grid on the symmetry plane. The effectiveness of the numerical simulation is largely influenced by the structure of the grid. Irregularities will inhibit its ability to solve the solution accurately and quickly. Quadrilateral type grid cells are more desirable, but are difficult to implement at discontinuities like the off-take to the air valve (T-junction) in question. The grid was therefore divided into quadrilateral and tetrahedral type cells, where the tetrahedral cells will be more appropriately matched to the

geometry in the region of the discontinuity. Cell density is also higher in this region to accommodate the expected high gradients in the flow field. **Figure 5.4** below shows a closer view of the region of the discontinuity, reflecting a higher grid density nearer to the pipe walls. This boundary layer grid structure assists in the more accurate solution of the flow profile nearer the wall, where a zero velocity or no-slip exists at the boundary. The total number of grid cells are in the order of 100 000 cells, but this value will be influenced by the specific geometry chosen.

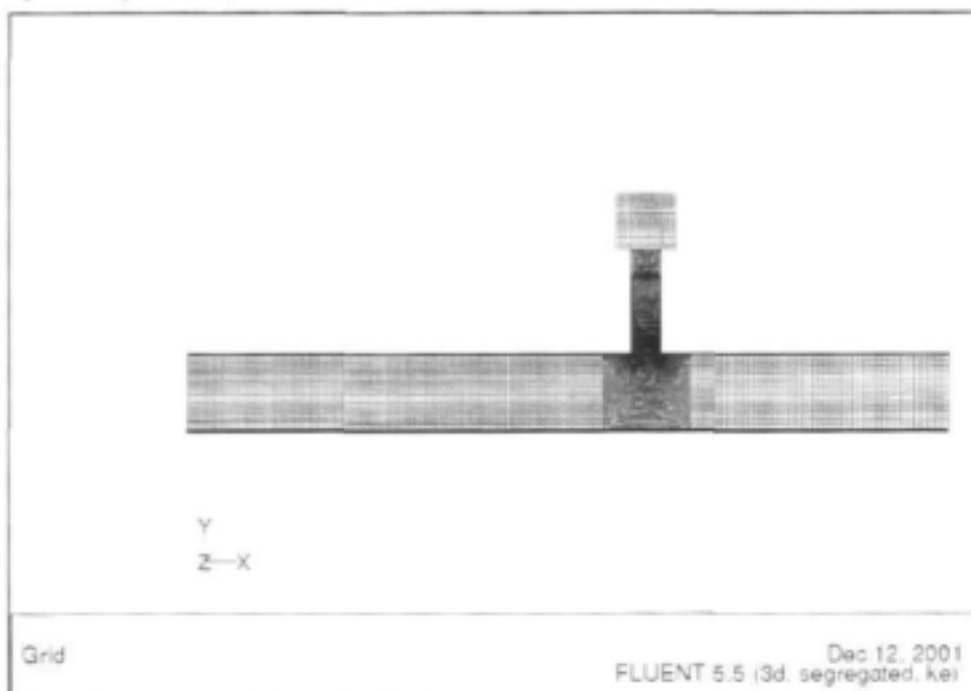


Figure 5.3: Grid structure

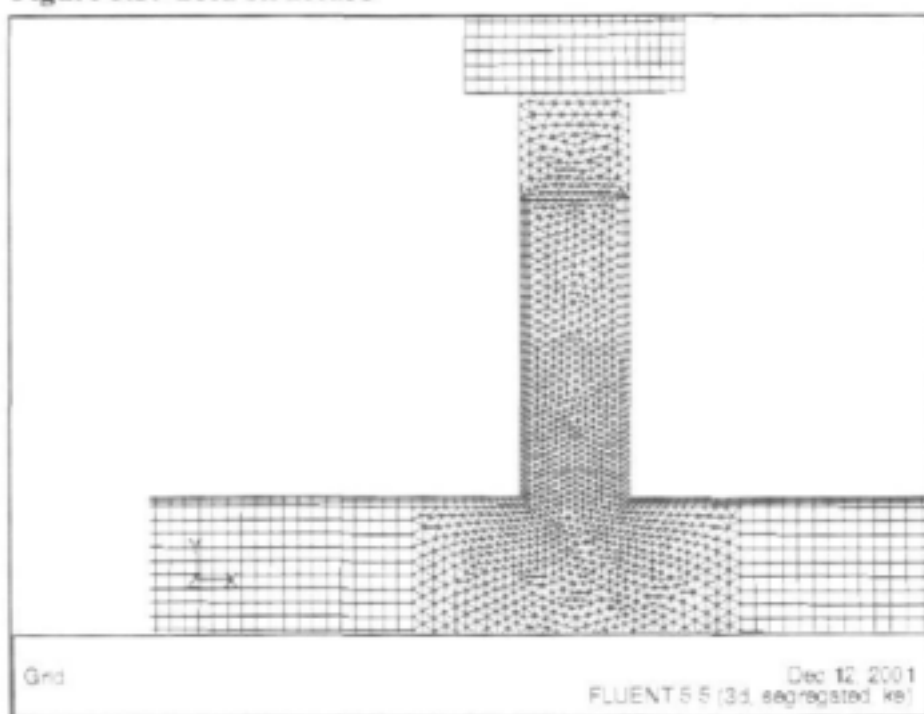


Figure 5.4: Tetrahedral cells and Boundary Layer

5.4.3 Boundary Conditions

Figure 5.5 illustrates the boundary conditions used in this model. As mentioned before, a symmetrical boundary was assumed and this helps to lower computational time, since fewer cells exist. The yellow face represents the symmetry boundary condition, while the black face represents a wall boundary condition. The blue and red faces show inlet and outlet boundaries respectively.

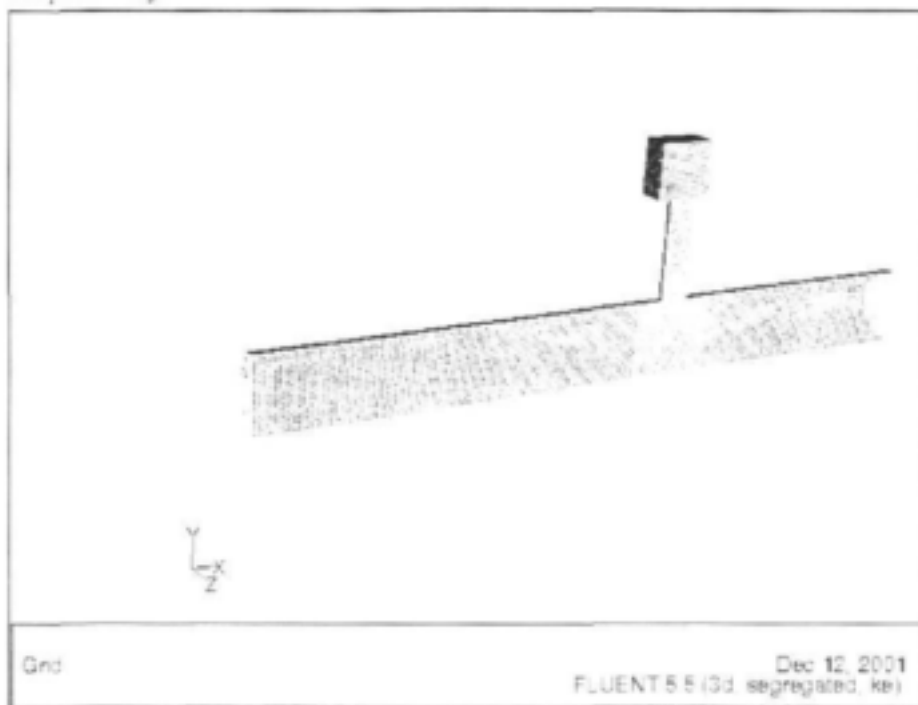


Figure 5.5: Boundary Conditions

The inlet boundary condition is provided to FLUENT in the form of a c-coded profile file. The variables of interest are those of velocity and bubble time. The velocity variable is the mean velocity of the flow in the pipeline. The profile-file then uses the logarithmic law (White, 1994) for turbulent pipe flow to determine the local mean velocity on each grid cell face on the inlet boundary. The file also determines the turbulence parameters used by the $k-\epsilon$ model on the same faces. The intention, when introducing this profile, is to assist the CFD code in reaching fully developed turbulent flow within the pipeline before reaching the discontinuity.

Figure 5.6 shows the variation in velocity profile over a 1 000 mm length of pipe (without the discontinuity), where the inlet velocity condition is shown by the red dots and the outlet velocity conditions are shown by the black dots. The plot shows little variation over the length of pipe and it was assumed to be an adequate inlet condition. Position (m) in **Figure 5.6** is measured from the center of the pipe.

The inlet was further divided into an air-inlet and a water inlet. **Figure 5.7** shows the divisions created from the instruction from the journal file. The blue face is the water-inlet boundary and the red face is the air-inlet boundary. For the period mentioned above (bubble time), the profile file will introduce air in the air-inlet section of the inlet boundary condition. The combination of the bubble time parameter in the profile file and the bubble size parameter in the journal file will determine the bubble volume.

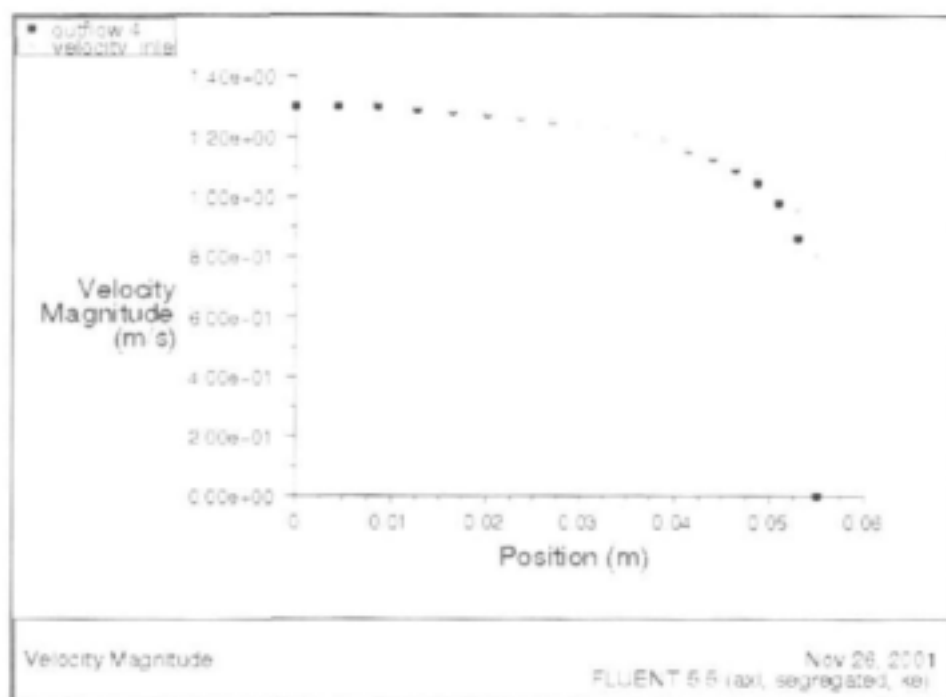


Figure 5.6: Inlet and Outlet velocity profiles

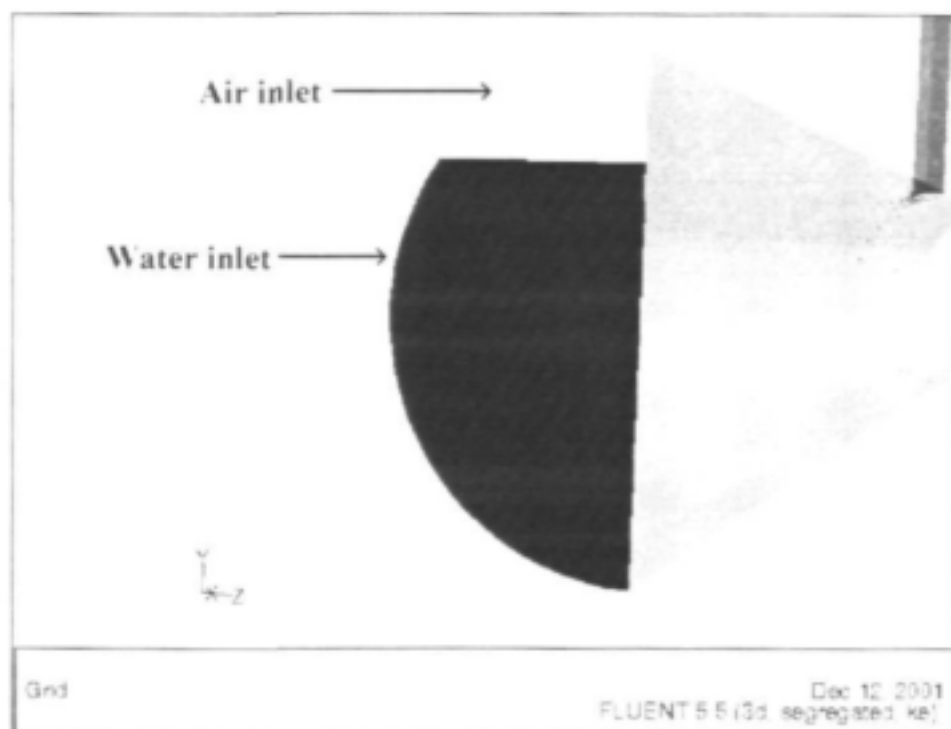


Figure 5.7: Air and Water inlet boundaries

For a given mean flow velocity V , pipe diameter, and bubble volume, the following may be calculated. The area on the air-inlet boundary A_a can be calculated using Equation 5.1 below:

$$A_a = 0.5R^2 \left(\frac{1}{\pi b} - \sin\left(\frac{1}{\pi b}\right) \right) \dots\dots\dots(5.1)$$

Where R is the radius of the pipeline, and b is the bubble size parameter mentioned as one of the inputs to the journal file used to build the geometry.

To establish the bubble time (Bt) input for the profile file, Equation 5.2 is utilised:

$$t = \frac{Vol}{A_a V} \dots\dots\dots(5.2)$$

Where Vol is the chosen bubble volume.

The final condition is that which manifests as the slope or inclination of the pipeline. This is controlled by an operational condition input into FLUENT, where a gravity vector was defined in Cartesian co-ordinates.

5.4.4 Fluid Properties

The fluid properties assumed for the two phases in question are as follows:

Table 5.1: Fluid properties

Fluid property	Value
Density (water) [kg/m ³]	1 000
Density (Air) [kg/m ³]	1.225
Viscosity (Water) [kg/m-s]	0.001003
Viscosity (Air) [kg/m-s]	1.7894 e-05
Surface Tension [N/m]	0.0728
Operating Pressure [kPa]	187

5.4.5 Solving Procedure

Since an unsteady multi-phase solution is desired in the study, a number of steps need to be followed. The end result will be the tracking of the boundary between the two phases. The steps followed are as follows:

- To initialise the flow in the pipeline, as it would be before the arrival of any air pockets, a steady state solution is computed. This allows the fluid to find an equilibrium state, and prepares it for the introduction of the air bubble.
- Secondly, a period of bubble introduction allows for the injection of the air pocket into the flow field. This section of the computation is un-steady, and is thus solved over defined time steps. These time step sizes are in the order of 0.0005 seconds and will continue for the time specified in the profile input file, (in the order of 0.076 seconds for larger bubbles).
- Once the bubble is introduced, the un-steady computation continues with similar time step sizes, until such time as the air pocket reaches the discontinuity/outlet. During this time, the bubble will assume an equilibrium state, influenced by mean flow velocity and buoyancy forces introduced due to the inclination of the gravity vector.
- Finally, during the period when the air pocket starts to rise into the smaller pipe as it is captured by the supposed de-aeration valve, smaller time step sizes are assumed. These smaller time step sizes ensure that the solution at each time step size converges correctly in accordance with certain criteria. Computation is slower in this step due to two major factors, namely the use of tetrahedral grid cells, and an increase in the number of free surfaces due to turbulence and air bubble break-up into smaller bubbles in the region of the discontinuity. Time step sizes for this step in the simulation can range between a high of 0.0002 seconds and as low as 0.00005 seconds.

Air pockets captured by the discontinuity will then continue to move into the small box at the termination of the small pipe, where it will be contained, while un-captured air pockets will continue past the discontinuity and out via the outlet boundary.

Run times for the complete solution of the simulation are in the order of 3-4 days, when run on Pentium 4 workstations with 1.5 GHz processing speed and 640 MB of RAM running on a Linux operating system.

5.5 Results

5.5.1 Steady State Single Phase

The first section of each simulation includes the completion of a steady state solution. **Figures 5.8 to 5.10** below show some of the flow characteristics of the steady state flow solutions. **Figure 5.8** illustrates contours of velocity magnitude in the region of the discontinuity. Non-zero velocity values can be seen in the first section of the small pipe. **Figure 5.9** illustrates velocity vectors in the region and indicates that the flow pattern in this region is one of recirculation. It was observed that smaller air pockets may be caught in this region, unable to rise due to buoyancy forces. The arrival of larger air pockets will, however, assist the movement of the smaller pockets, in that the smaller pockets will rejoin the larger ones and continue upward. **Figure 5.10** also illustrates the expected increase in turbulent kinetic energy in the region of the discontinuity. This in turn may have a negative influence on the ability of the discontinuity to effectively extract the air pocket from the pipeline. High regions of turbulence and circulation in the region of the entrance to the small pipe may provide a driving force, which may reintroduce smaller bubbles into the main flow field of the pipeline and may allow such bubbles to pass the discontinuity without extraction.

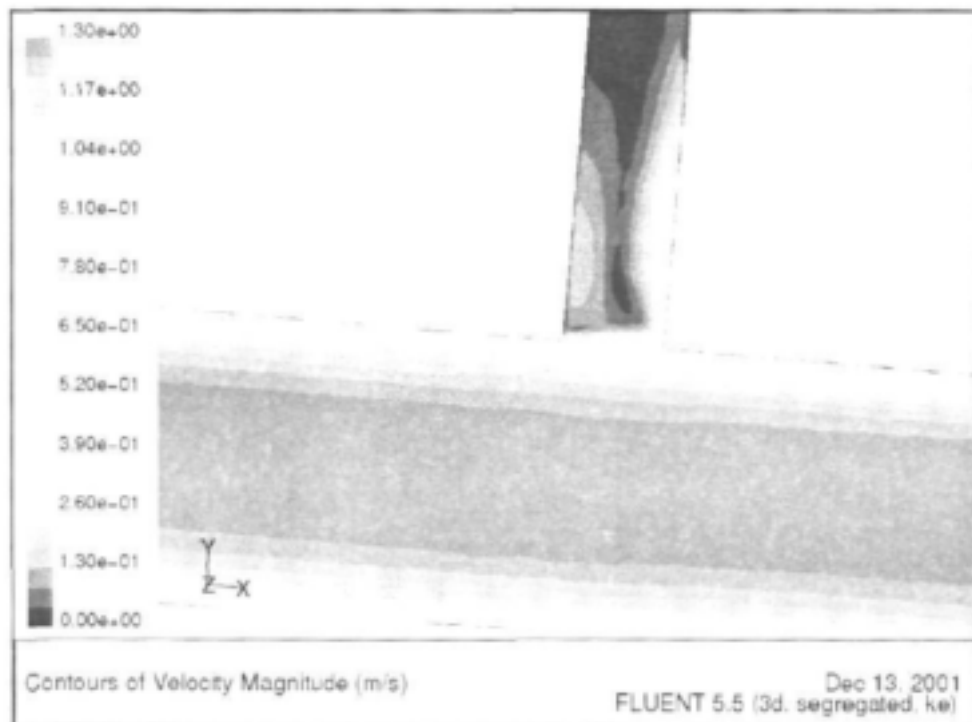


Figure 5.8: Contours of Velocity Magnitude

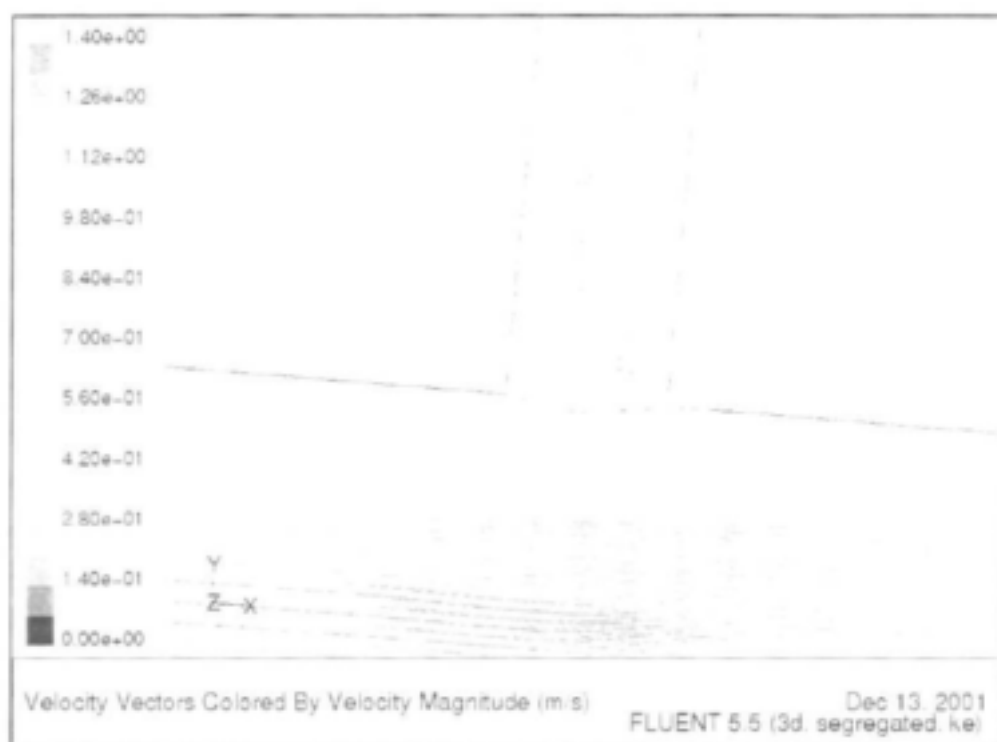


Figure 5.9: Velocity Vectors

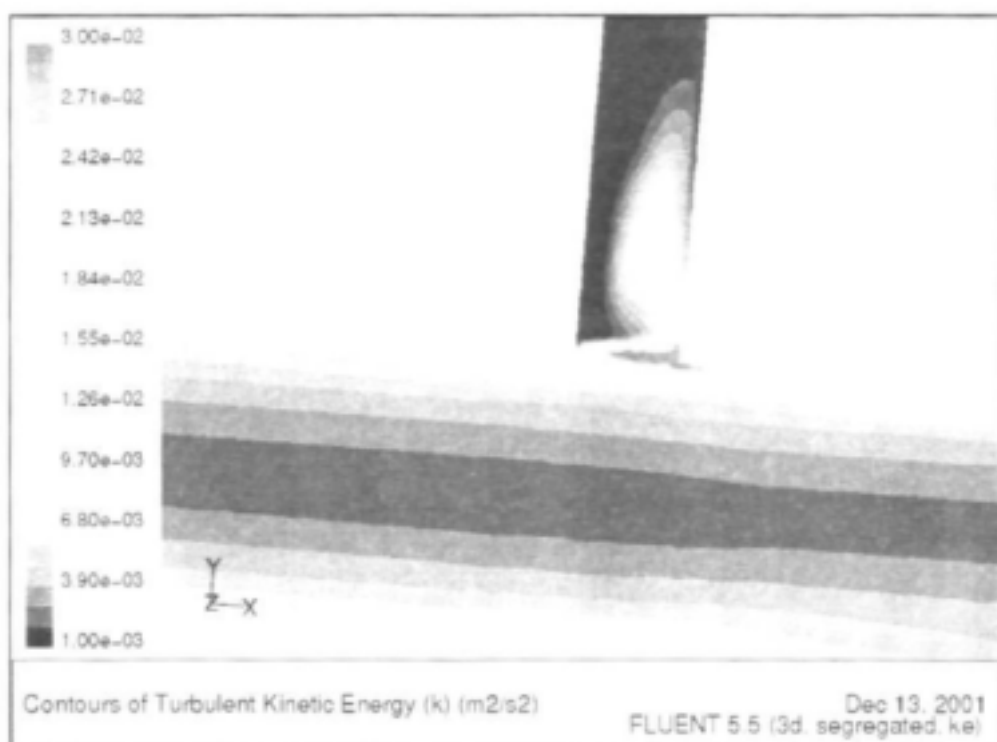


Figure 5.10: Contours of Kinetic Energy

5.5.2 Transient VOF solution

Transient results of this nature are best examined by visually tracking an iso-surface of constant volume fraction in the order of 0.5; it is possible to effectively track the motion of the air pockets in time. Video files were created that illustrate the motion of an air pocket, for a 5-degree pipeline inclination. These video clips can be obtained from the authors.

Similar results for variations in the other parameters considered have been obtained by varying the input values for these parameters in the input files.

The results were also analysed by considering the capture efficiency of the de-aeration valve. This result basically compares the ratio of the total volume of the air pocket captured by the discontinuity and settles in the box at the termination of the small pipe, and the total volume introduced into the pipeline upstream of the off-take.

These values can be directly computed by prompting FLUENT to provide results for average fluid density in the regions in question. **Table 5.2** below provides the results for the capture efficiency for a variation in inclination where mean flow velocity and air pocket volume were considered constant.

Table 5.2: Capture efficiencies for various slopes and diameter ratios

Air pocket Volume = 240 nml		
Mean Velocity = 1.2 m/s		
Diameter Ratio (d/D) = 0,35		
Pipeline Diameter = 110 mm		
Slope [degrees]	5	93,1%
	10	71,1%
	15	77,3%

During the unsteady simulation, the variation in the average volume fraction on a given face vs. time was monitored. This reflects when the large air pocket or smaller air pockets pass that face, and the speed at which the air pocket travels at for that given slope and flow velocity. **Figures 5.11 to 5.13** show the results of this surface monitoring for three variations

in slope. One should notice the decrease in air pocket speed with increasing pipeline inclination. This is due to the increase in the effect due to buoyancy, which also has an effect on air pocket shape. Also notice an increase in small air pocket breakaway from the front of the large air pocket. These smaller pockets are much faster and therefore reach the internal face sooner.

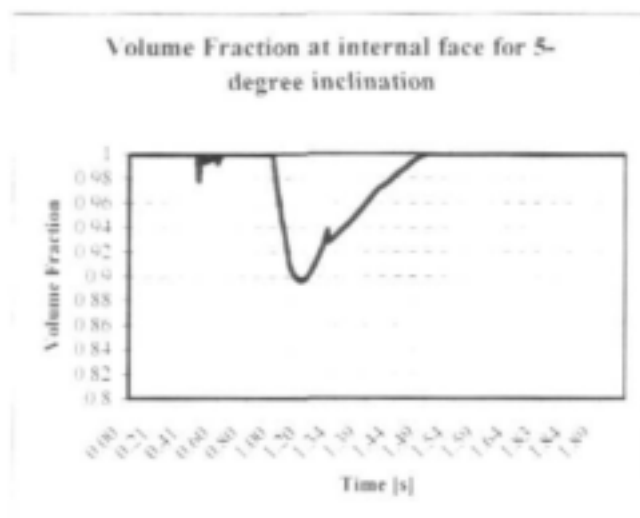


Figure 5.11: Volume Fraction at internal face for 5-degree inclination

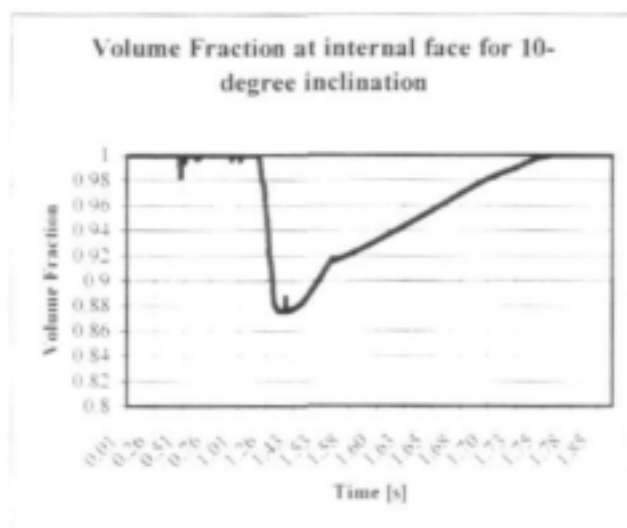


Figure 5.12: Volume Fraction at internal face for 10-degree inclination

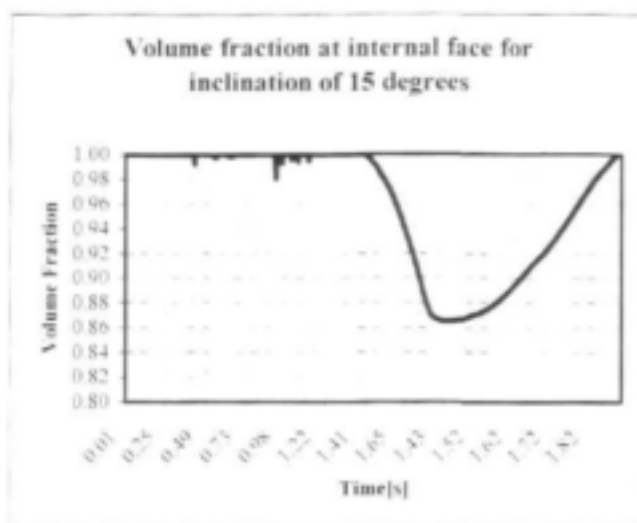


Figure 5.13: Volume Fraction at internal face for 15-degree inclination

5.5.3 Further analyses

Further analyses were performed in phase 2 of the CFD modelling. In these analyses, changes were made to the Geometry and the Grid generation. These aspects are discussed below.

5.5.3.1 Geometry building

The geometry used is identical to that used in previous studies, except that the length of the pipe is increased from 1 000 mm to 1 500 mm. This alteration is made in an attempt to improve the formation of the flow patterns before reaching the discontinuity. A further geometrical alteration was made for a single case, where instead of a small pipe being connected to the large pipe, a large pipe with similar diameter was used with a drilled hole linking the two pipes as shown in **Figure 5.14**. This geometry is investigated as it is hoped that it will match the experimental setup more accurately.

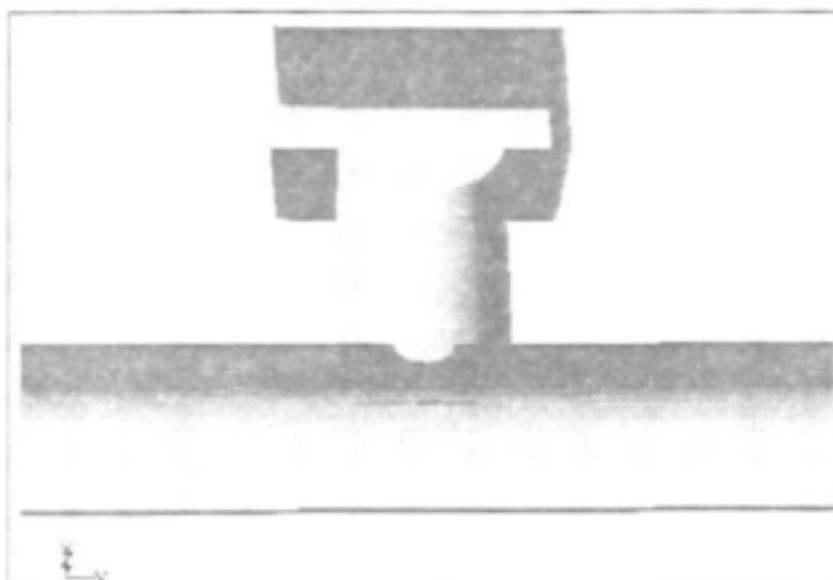


Figure 5.14: Geometry of drilled-hole configuration

5.5.3.2 Grid Generation

A major alteration, which has been made to the grid generating approach for this study, is that only hexagonal grid cells are used. Similar multiphase studies have shown that this grid cell type significantly improves both solution time and accuracy. **Figure 5.15** below shows the configuration of the grid on the symmetry plane. More effort was made to ensure cell volume uniformity, as this is considered important when performing free surface simulations. The cell count for the simulations ranges from 350 to 400 thousand grid cells for the entire volume.

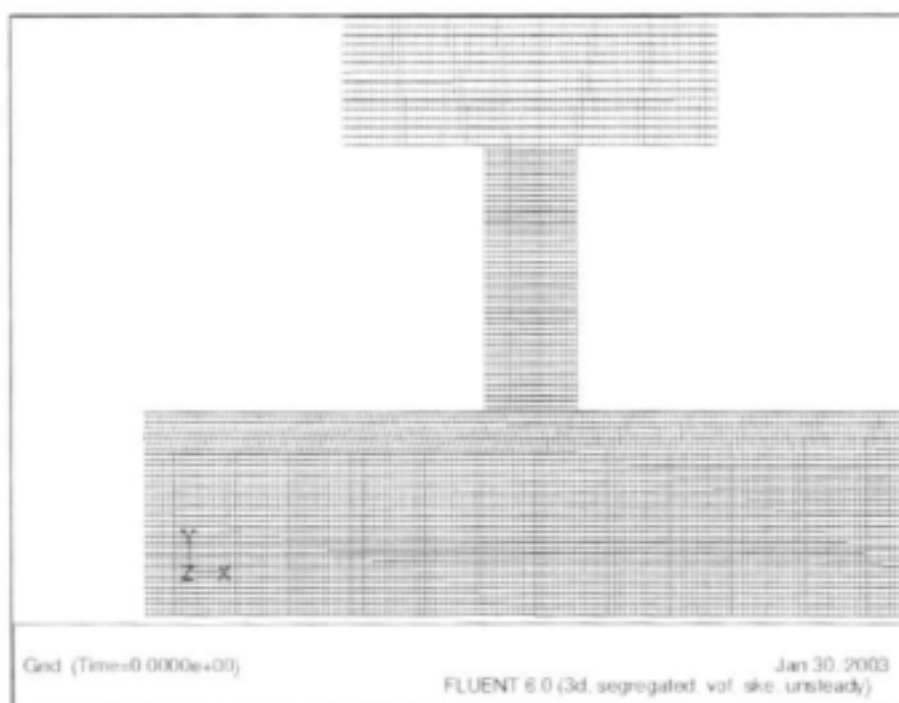


Figure 5.15: Grid in region of discontinuity

Table 5.3 reflects the details of the different analyses that were undertaken as part of this Phase of the project.

Table 5.3: Numerical analyses conducted as part of phase 2 of this investigation

Air pocket Volume = 240 nml Mean Velocity = 1.2 m/s Pipeline Diameter = 110 mm				
Slope [degrees]		35% drilled *	45% Pipe	55% Pipe
	5°	-	X	X
	10°	-	X	X
	15°	X	-	-

Note:

* The outlet from the pipe was modelled as an orifice that bleeds into an equal diameter pipe

Figures 5.16 to 5.20 graphically represent the results. These figures indicate the location of the air bubble that was introduced at the start of the analysis. With reference to these figures the "Entire Volume" represents the total volume of the numerical model while the "Valve" reflects the air volume that is trapped in the air valve.

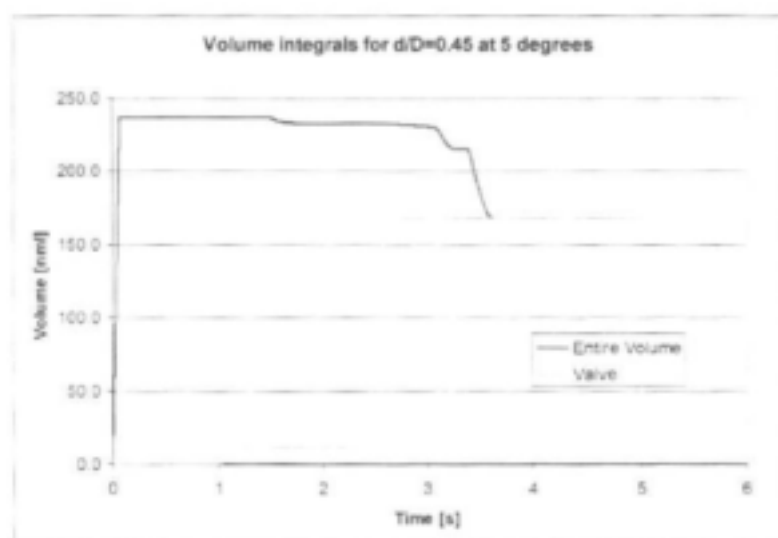


Figure 5.16: Volume integrals for $d/D=0.45$ at 5 degrees

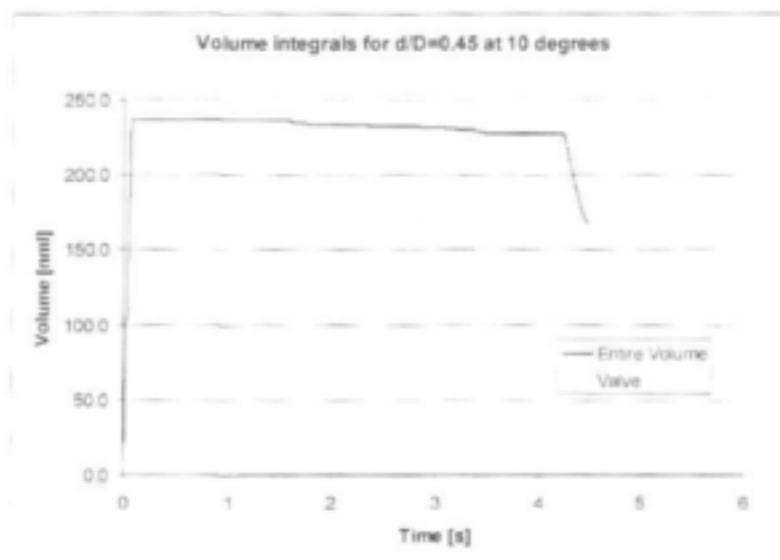


Figure 5.17: Volume integrals for $d/D=0.45$ at 10 degrees

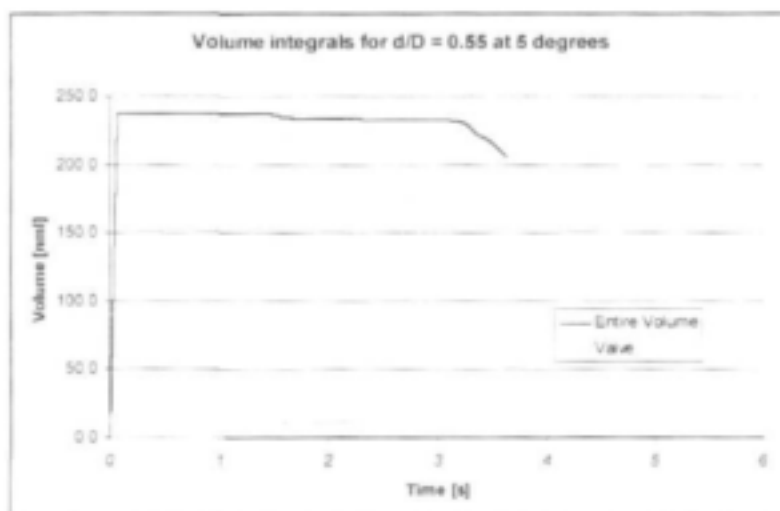


Figure 5.18: Volume integrals for $d/D=0.55$ at 5 degrees

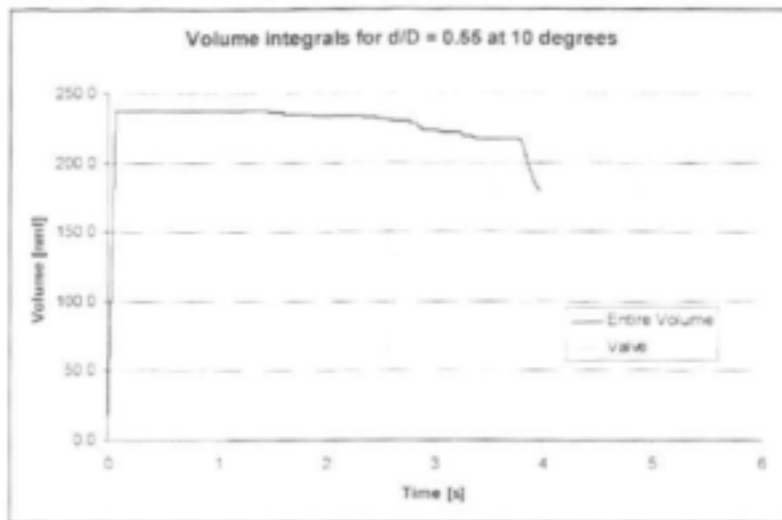


Figure 5.19: Volume integrals for $d/D=0,55$ at 10 degrees

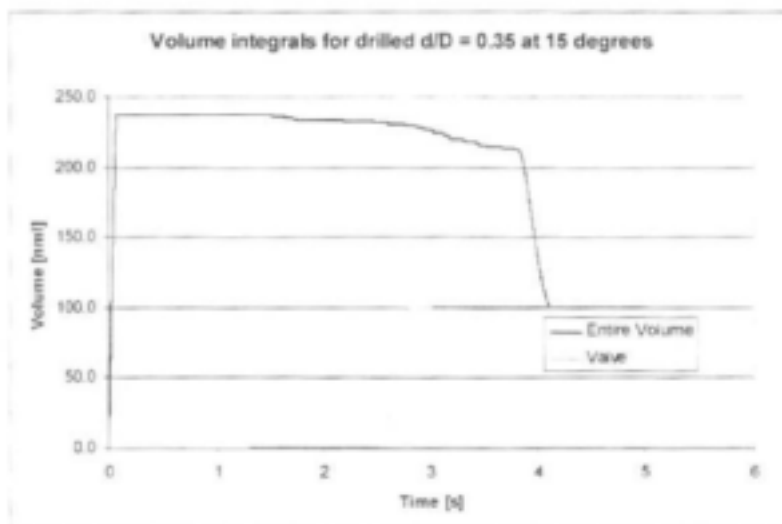


Figure 5.20: Volume integrals for drilled $d/D=0,55$ at 15 degrees

The general trend for these results is that the larger the off-take and the flatter the slope, the more air is released through the air valve. An explanation for the improved efficiencies for shallower angles may be that the bubbles are spread over a larger area; hence they have more time to be expelled and in this way introduce them slowly into the off-take (air-release) pipe. The AVI videos made for these analyses clearly reflect this.

This investigation showed the applicability of CFD in the modelling and optimisation of the air off-take on a water pipeline. The results showed that the complex two-phase flow could be successfully modelled using the Volume of Fluid Method. However, the physical computational time involved might be of concern. The results obtained from this study were compared to experimental results to validate the accuracy of the method. **Figure 5.21** reflect the comparison between the experimental and the numerical modelling results.

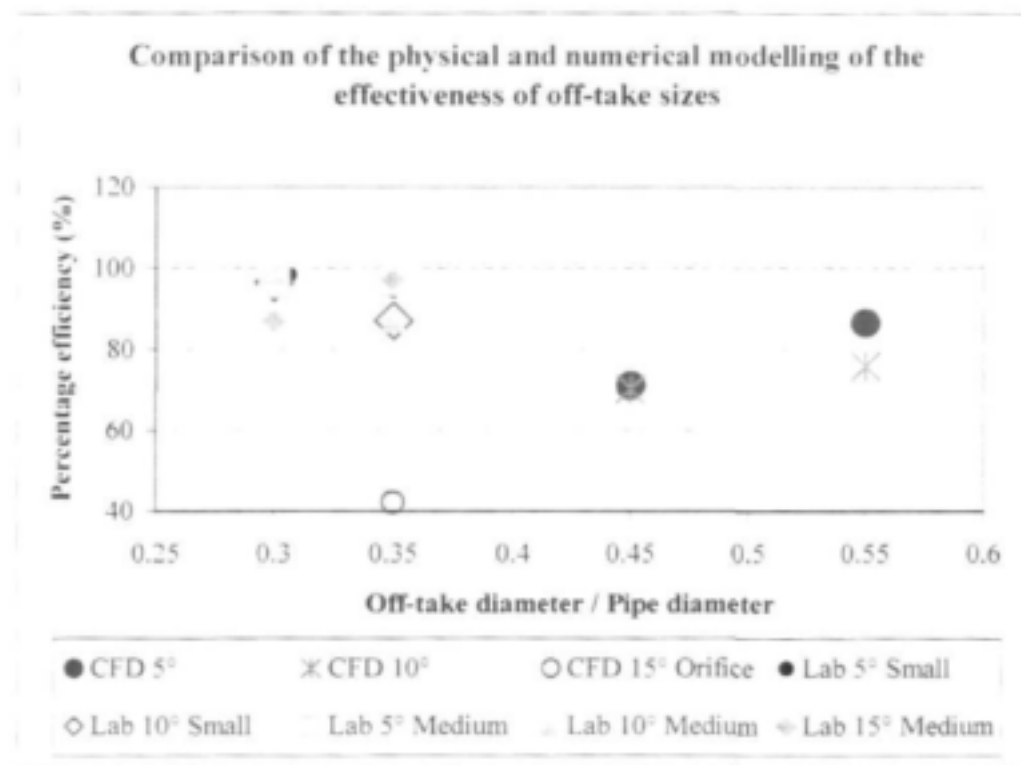


Figure 5.21: Comparison of the laboratory results with the results from the CFD modelling

In **Figure 5.21** it seems that the CFD model efficiencies are less than those obtained from the laboratory tests. The reason for this discrepancy is due to different air bubble sizes. **Figure 5.22** reflects the same data set but the bubble sizes have been included.

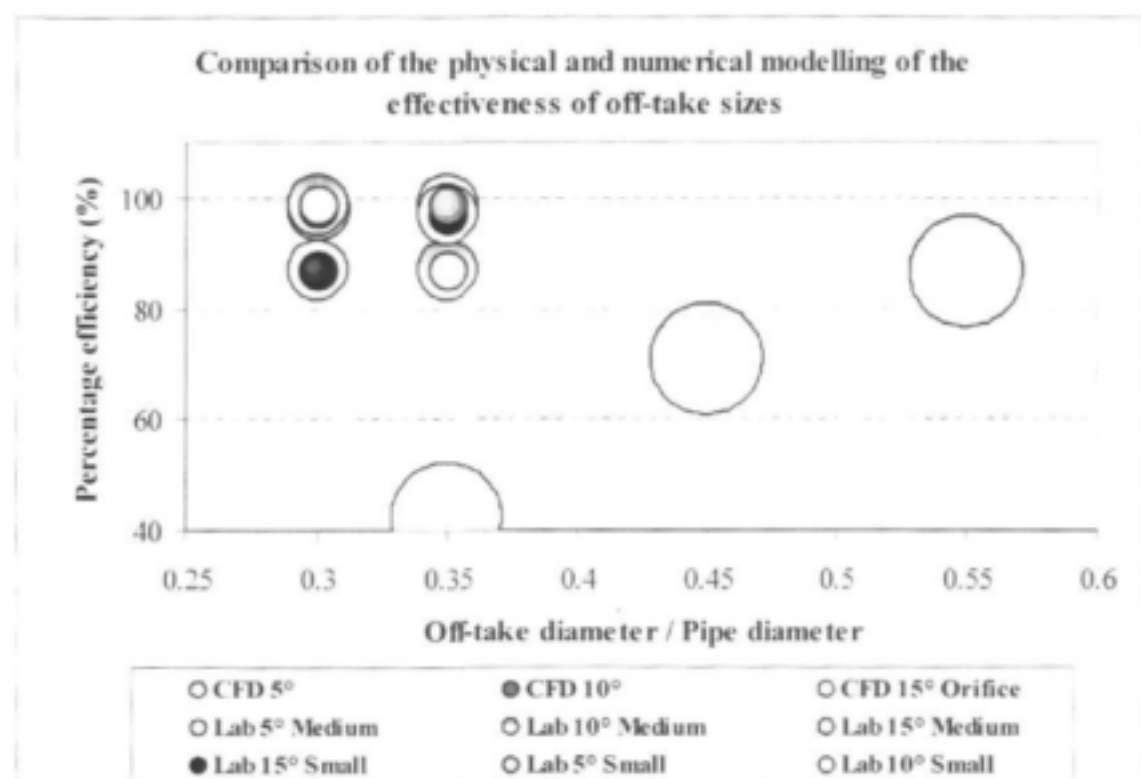


Figure 5.22: Comparison of the laboratory results with the results from the CFD modelling

The methodology developed in this study can be used to study the effect of the influence of the different design parameters of the air off-take.

6. EFFECTIVE DE-AERATION OF PIPELINES

6.1 Introduction

The objectives of this research was to define the parameters needed for effective de-aeration of large diameter pipeline and derive procedures that can be used to ensure an effective de-aerated pipeline. Two operating scenarios that needs to be evaluated for any pipeline, in so far as the influence and consequences of air in water pipelines is concerned are:

- First time filling and
- Air release during normal operating scenarios.

6.2 First time and subsequent filling of the pipeline

During the testing and commissioning of pipelines, the problems related to ineffective de-aeration is common. Ineffective filling can be contributed to a number of factors, the most important are probable the lack of planning and a structured way to fill the pipeline in a controlled manner, ensuring that all the air release valves are fully operational and that dynamic induced pressures are limited to within the structural capability of the pipeline.

Difficulties with regard to the definition of the save filling rate involves a number of parameters that have been discussed in the document. The conservative approach to fill the line as slow as possible can however be impractical and it should be compared with other options to allow water to be flushed through the system, ensuring the transfer of momentum to localized air pockets that can be release at selected positions along the line. Filling is a site-specific problem and should be handled as such. In Volume 2 – **“Provisional Guidelines for the effective de-aeration of large diameter water pipelines”** this aspect is discussed further.

6.3 De-aeration during operating conditions

The objective to effectively de-aerate pipelines requires a number of aspects that should be considered and reviewed. The most important aspects are:

- Accurate description of the system and its functionality.
- Detailed longitudinal profile and pipe details

- Operational conditions, especially for a phased implementation of the system, resulting in low demands at the offset of the project
- Positioning of valves, abstractions and controls along the pipeline
- Client's preferences with regard to materials and the implementation of acceptable design criteria and
- Operational controls of the system.

Using air valves are an integral part of the effective de-aeration of pipelines and it is therefore important to incorporate the following steps into the design of the pipeline:

- Examine appropriate ways to evaluate the performance of air valves
- Determine the positioning and size of air valves and
- Consider the alternative benefits of air induction as a way to decrease dynamic pressures during the operation.

6.4 A structured way to ensure a fully de-aerated system

Based on the findings of the work that was conducted under this project a guideline for the effective de-aeration of large diameter water pipelines has been compiled, including the following aspects:

- Description of air valve types
- Sizing and positioning of air valves
- Typical air valve installations and
- Software to assist the designer of pipelines to ensure effective de-aeration.

This information should assist pipeline designers to design a de-aerated system and be aware of all the possible pitfalls and possible options and solutions to problems.

7. FURTHER RESEARCH

The promotion by the Water Research Commission to commercialise and use research findings creates the opportunity that the Provisional Guideline (Volume 2) might have to be extended in the future.

The contribution of the research findings and the provisional guide will contribute to the evaluation of existing systems and the design of new pipelines.

With the aging water infrastructure in South Africa a crucial aspect remains the quantification of the contributing factors that reduces the hydraulic capacity. In the absence of reliable as-built details, it becomes extremely difficult to assess the potential influence of air on the hydraulic capacity. The high capital cost of renovating all the air valves on a pipeline, necessitates the development of procedures to locate air pockets in pipelines under operating conditions.

In light of this it is recommended that:

- The use of inverse transients be developed as a tool to evaluate the operational characteristics of the system and to identify factors that influence the capacity of the system. The procedure should be developed to such a level that it can be used to evaluate a pipeline, mainly with the objective of determining the effectiveness of de-aeration equipment as well as the identification of leakages.
- Due to the difficulty of field investigations with regard to the movement and capturing of air in large diameter pipelines, further numerical modelling should be conducted on the transportation and capturing of air at the discontinuities that are provided for air valves. This will indicate if any scale effects are present which have to be considered and incorporated for the application of normative guidelines on large diameter pipelines. The optimisation of the proposed layouts could result in considerable cost savings for a sustainable water supply system.

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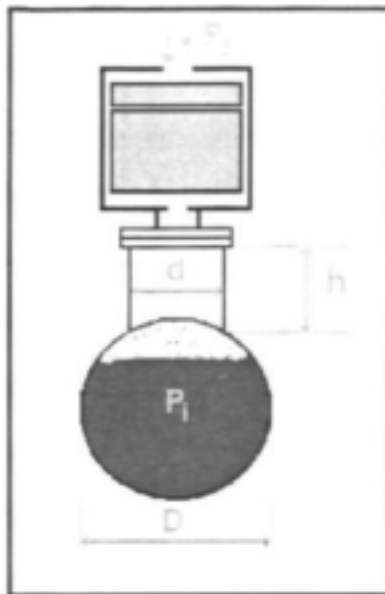
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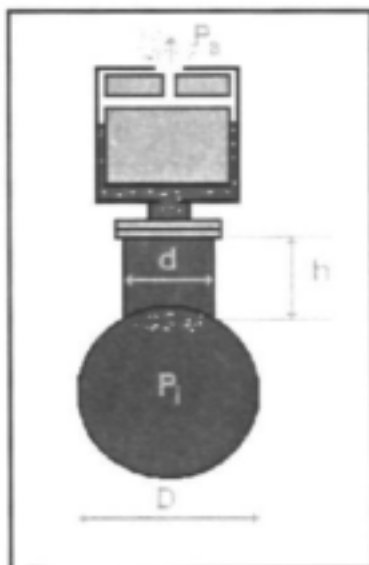
APPENDIX 2.1

The layout of air valve installations (Van Vuuren, 1993)



VARIABLE	LARGE ORIFICE FUNCTION:	
	VACUUM BREAK	AIR RELEASE
Pressure (orifice size)	$P_2 - P_1 \leq 40 \text{ (kPa)}$	$P_1 - P_2 \leq 5 \text{ (kPa)}^*$
Spacing	Between shut-off valves or to allow free draining	High points on an isolated pipeline
h	$h \geq 0.5D$; $h > 0.15 \text{ (m)}$	$h \geq 0.5D$; $h > 0.15 \text{ (m)}$
d	$d = D$; $d \leq 0.8 \text{ (m)}$	$d = D$; $d \leq 0.8 \text{ (m)}$

Note : * To prevent dynamic closure of the large orifice



VARIABLE	SMALL ORIFICE FUNCTION:
Spacing	As topography requires
h	$h \geq 0.5D$; $h \geq 0.15 \text{ (m)}$
d	$d \geq D$; $d \leq 0.8 \text{ (m)}$

APPENDIX 3.3.1

Determining the priority of changes to the system

		Ranked sections		
Pipe section nr	Slope (m/m)	Pipe section nr	Slope (m/m)	
2	-0.0292	67	-0.2339	Priority 1
7	-0.0320	22	-0.1711	
11	-0.0122	154	-0.0631	
13	-0.0052	51	-0.0521	
14	-0.0230	96	-0.0418	
16	-0.0120	7	-0.0320	
19	-0.0184	2	-0.0292	
21	-0.0186	103	-0.0276	
22	-0.1711	54	-0.0272	
25	-0.0074	104	-0.0265	
37	-0.0010	97	-0.0250	
38	-0.0091	95	-0.0231	
43	-0.0007	14	-0.0230	
44	-0.0115	122	-0.0211	
45	-0.0125	61	-0.0192	Priority 2
46	-0.0097	21	-0.0186	
49	-0.0042	19	-0.0184	
51	-0.0521	53	-0.0177	
52	-0.0069	62	-0.0174	
53	-0.0177	123	-0.0170	
54	-0.0272	94	-0.0170	
56	-0.0158	121	-0.0169	
57	-0.0068	130	-0.0163	
61	-0.0192	109	-0.0159	
62	-0.0174	56	-0.0158	
63	-0.0108	120	-0.0153	
67	-0.2339	124	-0.0132	
73	-0.0043	45	-0.0125	
75	-0.0049	11	-0.0122	
76	-0.0001	16	-0.0120	
77	-0.0016	153	-0.0120	
79	-0.0105	44	-0.0115	
80	-0.0062	108	-0.0112	
81	-0.0058	63	-0.0108	
82	-0.0003	79	-0.0105	
83	-0.0015	145	-0.0102	
92	-0.0048	46	-0.0097	Priority 3
93	-0.0020	38	-0.0091	
94	-0.0170	25	-0.0074	
95	-0.0231	52	-0.0069	
96	-0.0418	57	-0.0068	
97	-0.0250	80	-0.0062	
103	-0.0276	81	-0.0058	
104	-0.0265	152	-0.0056	
108	-0.0112	125	-0.0053	
109	-0.0159	13	-0.0052	
120	-0.0153	75	-0.0049	
121	-0.0169	92	-0.0048	
122	-0.0211	136	-0.0044	
123	-0.0170	73	-0.0043	
124	-0.0132	49	-0.0042	
125	-0.0053	150	-0.0033	
126	-0.0012	135	-0.0028	
130	-0.0163	138	-0.0023	
135	-0.0028	93	-0.0020	
136	-0.0044	77	-0.0016	
138	-0.0023	83	-0.0015	
141	-0.0000	126	-0.0012	
145	-0.0102	37	-0.0010	
150	-0.0033	151	-0.0008	
151	-0.0008	43	-0.0007	
152	-0.0056	82	-0.0003	
153	-0.0120	76	-0.0001	
154	-0.0631	141	-0.0000	

PIPE ANGLE =15 DEGREES AT A 70mm DISCONTINUITY

Test nr	Small flow meter		pump flow meter Q	Measurements at air inlet			Air regulator settings				Measurements at the discontinuity			
				Calibration bubble volume		Bubble speed	Bubble volume	Amount bubbles	Gap	Bubble pressure	Pressure at discontinuity		Electronic pressure meter	Mechanical pressure meter
	V	Q		Pressurized	Atmospheric						Under pressure	Atmospheric		
	m³/s	l/s	l/s	liters	liters	m/s	milliseconds		sec	bar	liters	liters	m	kPa or N/m²
2.1	1.00	1.8627	18.1			0.330	25	100	3	3	4.28	5.54	2.7	28.00
2.2	1.00	1.8627	18.1			0.364	25	100	3	3	4.27	5.45	2.7	28.00
2.3	1.00	1.8627	18.1			0.330	25	100	3	3	4.28	5.54	2.7	28.00
conclusion	1.00	1.8627	18.1				25	100	3	3			2.7	28.00
3.1	1.00	1.8627	18.1			0.136	75	30	5	6 ksc	4.01	5.26	2.7	28.00
3.2	1.00	1.8627	18.1			0.227	75	30	5	6 ksc	4.29	5.51	2.7	28.00
3.3	1.00	1.8627	18.1			0.230	75	30	5	6 ksc	-	5.60	2.7	28.00
3.4	1.00	1.8627	18.1				75	30	5	6 ksc	4.20	5.52	2.7	28.00
conclusion	1.00	1.8627	18.1				75	30	5	6 ksc			2.7	28.00
4.1	1.00	1.8627	18.1			0.154	80	12	9	6 ksc/s	7.10	9.27	2.7	28.00
4.2	1.00	1.8627	18.1			0.154	80	12	9	6 ksc/s	3.75	4.85	2.7	28.00
4.3	1.00	1.8627	18.1				80	12	9	6 ksc/s	4.32	5.44	2.7	28.00
conclusion	1.00	1.8627	18.1				80	12	9	6 ksc/s	4.00	5.40	2.7	28.00

APPENDIX 4.2.2

EXAMPLE OF DATA SHEET

Table A1: Example of a data sheet for recording the velocities of the bubbles

PIPE ANGLE = 0°

Water Velocity = 1 m/s				Water Velocity = 1,5 m/s		
Readings	Small	Medium	Large	Small	Medium	Large
1	6.40	5.89	3.71	4.00	4.29	2.67
2	6.57	6.07	3.90	4.10	4.07	2.54
3	6.35	6.65	3.84	4.07	4.41	2.63
4	5.86	5.79	3.69	4.21	4.49	2.55
5	6.48	6.03	3.70	3.96	4.27	2.57
6	6.53	5.74	3.92	4.16	4.02	2.62
7	6.50	5.89	3.91	4.02	4.12	2.53
8	6.41	6.47	4.09	4.37	4.18	2.60
9	6.73	6.40	3.84	4.15	4.10	2.46
10	6.65	6.39	3.76	4.16	3.65	2.56
11	6.34	6.24	4.06	4.22	3.79	2.68
12	6.22	6.59	3.67	4.09	4.07	2.63
13	6.40	5.92	4.00	4.36	3.92	2.64
14	6.05	5.28	3.98	4.00	4.05	2.65
15	6.39	6.22	4.01	3.95	4.39	2.67
16	6.30	5.80	3.90	4.38	4.21	2.66
17	6.41	6.47	3.95	4.19	3.76	2.67
18	6.70	6.25	3.97	3.83	4.09	2.54
19	6.10	5.60	4.00	4.30	4.17	2.57
20	6.08	6.46	3.90	4.12	4.02	2.58
Total	127.47	122.15	77.80	82.64	82.07	52.02
Average	6.37	6.11	3.89	4.13	4.10	2.60
Velocity	0,784	0,819	1,285	1,210	1,218	1,922

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The prediction for chlorine decay from potable water in pipeline systems

Viljoen OJ; Haarhoff J; Joubert JBH

Knowing the rate of chlorine level in a potable water distribution system is important for correct chlorine dosage. The research results from this project made a significant contribution to the improved design and management of local potable water disinfection technologies - especially in the case of long distribution lines typically found in South Africa. Overseas models could not be applied with confidence locally because of the:

- Higher level of treatment generally practised overseas
- Generally colder temperatures encountered there
- Higher levels of organic materials normally found in South African raw waters.

Overseas models generally use a first-order chlorine decay rate. However, the research team showed convincingly that the use of an n^{th} order model improves the accuracy of such a decay model. This reaction model, developed for both the decay of chlorine and chloramines, may now be used for almost all distribution systems found in South Africa. Other water utilities will be able to determine their own reaction rate constants for the model by using the test protocol described in the report.

Report Number: 704/1/97

ISBN: 1 86845 324 3

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Water Research Commission
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Tel: +27 12 330 0340, Fax: +27 12 331 2565
Web: <http://www.wrc.org.za>