Improving the Airlift Pump Prediction Model for Aquaculture Applications

by

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ABSTRACT

IMPROVING THE AIRLIFT PUMP PREDICTION MODEL FOR AQUACULTURE APPLICATIONS

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The rapid growth of the aquaculture industry has resulted in a demand for highly efficient methods to control water quality. This thesis is an investigation of airlift pumps and their applications to the aquaculture industry. Airlift pump performance data was collected experimentally in a laboratory setting and in a real-world setting at an aquaculture facility. A pump diameter of 3.175 cm was tested in a lab with an adjustable submergence level. Submergence ratios of 0.5, 0.7, and 0.9 were considered. Four pump size were tested at different aquaculture farms: 5.08 cm, 10.16 cm, 15.2 cm, and 20.32 cm. A theoretical airlift pump performance prediction model was developed using the slip two-phase model and was improved on by implementing two-phase flow pattern detection. Four general flow patterns were considered: Bubbly flow, Slug flow, Churn flow and Annular flow. The experimental results were compared to the improved flow pattern dependent prediction model. The model was implemented to design an airlift pump system for an aquaculture raceway. The operation of airlift pumps for circulating water in the raceway was validated by analyzing an existing system.
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# Table of Contents

Abstract ................................................................................................................................. ii
Acknowledgements ................................................................................................................ iii
List of Figures .......................................................................................................................... vi
List of Tables .......................................................................................................................... viii
Nomenclature ........................................................................................................................ ix
Chapter 1 - Introduction ....................................................................................................... 1
Chapter 2 - Literature Review .............................................................................................. 6
  2.1 Theory and Background of Airlift Pumps ....................................................................... 6
  2.2 Experimental Work .......................................................................................................... 8
  2.3 Modelling Airlift Pumps .................................................................................................. 13
  2.4 Use of Airlift Pumps in Aquaculture Applications ......................................................... 15
  2.5 Objective of the Study ..................................................................................................... 18
Chapter 3 - Theoretical Analysis Mathematical Modelling .................................................. 20
  3.1 Conservation of Mass ..................................................................................................... 20
  3.2 Conservation of Momentum ........................................................................................... 21
  3.3 Conservation of Energy .................................................................................................. 22
  3.4 Simplified Two-Phase Flow Approach .......................................................................... 24
  3.5 Flow Pattern Dependent Modelling: ............................................................................. 29
      The Bubbly-Slug Transition ............................................................................................. 29
      The Slug-Churn Transition ............................................................................................. 30
      Transition to Annular Flow ............................................................................................ 30
  3.6 Integrating the Slip Ratio .............................................................................................. 32
      Bubbly Flow Correlations .............................................................................................. 32
      Slug Flow Correlations .................................................................................................. 33
      Churn Flow Correlations ............................................................................................... 33
      Annular Flow Correlations ............................................................................................ 34
  3.7 Algorithm Solution: ....................................................................................................... 36
Chapter 4 - Experimental Setup ........................................................................................... 39
  4.1 Laboratory Setup ............................................................................................................ 39
List of Figures

Figure 2-1 A schematic of an airlift pump ................................................................. 6
Figure 2-2 Flow pattern illustration ........................................................................ 8
Figure 3-1 Solution algorithm logic expanded ......................................................... 37
Figure 3-2 Solution algorithm logic simplified ......................................................... 38
Figure 4-1 Capacitance sensor (Dimensions in mm) ................................................. 40
Figure 4-2 Lab setup schematic ............................................................................. 41
Figure 4-3 Field experimental setup (dimensions in meters) .................................... 44
Figure 4-4 Airlift pumps system installation and operation ....................................... 44
Figure 5-1 Experimental performance data of D=3.175 Sr=0.9 .................................. 47
Figure 5-2 Performance results D=3.175 cm ......................................................... 47
Figure 5-3 Efficiency results of 3.175 cm ............................................................... 48
Figure 5-4 sequence of slug flow pattern ............................................................... 49
Figure 5-5 Typical flow patterns observed in airlift pump ....................................... 49
Figure 5-6 Generated flow pattern map D=3.175 cm Taitel et al. (1980) .................. 50
Figure 5-7 Performance results of 0.5 Sr compared to a drift flux model with different slip ratio correlations ......................................................................................... 51
Figure 5-8 Efficiency results of 0.5 Sr compared to drift flux model with different slip ratio correlations ......................................................................................... 52
Figure 5-9 Performance results of 0.7 Sr compared to drift flux model with different slip ................................................................. 53
Figure 5-10 Efficiency results of 0.7 Sr compared to drift flux model with different slip ratio correlations ......................................................................................... 54
Figure 5-11 Performance results of 0.9 Sr compared to drift flux model with different slip ratio correlations ......................................................................................... 55
Figure 5-12 Efficiency results of 0.9 Sr compared to drift flux model with different slip ratio correlations ......................................................................................... 56
Figure 5-13 Measured vs predicted output water mass flow rate .............................. 57
Figure 5-14 Performance of large diameter airlift pumps ........................................ 59
Figure 5-15 Efficiency of large diameter airlift pumps ............................................ 60
Figure 5-16 Performance of 5.08 cm diameter airlift pump compared to model .................. 60
Figure 5-17 Performance of 10.16 cm diameter airlift pump compared to model ................ 61
Figure 5-18 Performance of 15.24 cm diameter airlift pump compared to model ................ 61
Figure 5-19 Performance of 20.32 cm diameter airlift pump compared to model ................ 62
Figure 5-20 Airlift pump performance prediction model for different D ............................. 63
Figure 5-21 Airlift pump efficiency prediction model for different D .................................. 64
Figure 5-22 Airlift pump performance prediction model for different L .............................. 65
Figure 5-23 Airlift pump efficiency prediction model for different L ................................... 65
Figure 5-24 Airlift pump efficiency prediction model for different Sr ................................. 67
Figure 5-25 Airlift pump efficiency prediction model for different Sr ................................. 67
Figure 5-26 Raceway velocity location .............................................................................. 70
Figure 5-27 Airlift pumps system installation and operation for raceway setup ..................... 71
List of Tables

Table 1 Summary of flow pattern transition lines developed by Taitel et al. (1980) .................. 31
Table 2 Void fraction and slip ratio correlations ............................................................................ 35
Table 3 Root mean square values of measured vs predicted water mass flow rate ...................... 57
Table 4 McMillian Pitts raceway dimensions ................................................................................. 68
Table 5 Baseline raceway velocity data .......................................................................................... 69
Table 6 Velocity reading after installing airlift pumps ................................................................. 70
## Nomenclature

<table>
<thead>
<tr>
<th>Symbol</th>
<th>Definition</th>
<th>Symbol</th>
<th>Definition</th>
</tr>
</thead>
<tbody>
<tr>
<td>( A )</td>
<td>Cross-sectional area of pipe</td>
<td>( v_{jg} )</td>
<td>Weighted mean drift velocity</td>
</tr>
<tr>
<td>( C_o )</td>
<td>Distribution parameter</td>
<td>( w )</td>
<td>Weight</td>
</tr>
<tr>
<td>( d_h )</td>
<td>Hydraulic diameter</td>
<td>( W )</td>
<td>Mass flow rate</td>
</tr>
<tr>
<td>( D )</td>
<td>Diameter</td>
<td>( \alpha )</td>
<td>Void Fraction</td>
</tr>
<tr>
<td>( f )</td>
<td>Friction</td>
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<td>( g )</td>
<td>Acceleration of gravity</td>
<td>( \tau )</td>
<td>Shear stress</td>
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<tr>
<td>( H_s )</td>
<td>Static head</td>
<td>( \epsilon )</td>
<td>Internal energy</td>
</tr>
<tr>
<td>( i )</td>
<td>Enthalpy</td>
<td>( \eta )</td>
<td>Efficiency</td>
</tr>
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<td>( J )</td>
<td>Superficial velocity</td>
<td>( \rho )</td>
<td>Density</td>
</tr>
<tr>
<td>( l )</td>
<td>Slug length</td>
<td>( \nu )</td>
<td>Kinematic viscosity</td>
</tr>
<tr>
<td>( L )</td>
<td>Pipe length</td>
<td>( \phi )</td>
<td>Heat flux</td>
</tr>
<tr>
<td>( p )</td>
<td>Pressure</td>
<td>( \Gamma )</td>
<td>Mass generation rate over unit length</td>
</tr>
<tr>
<td>( P )</td>
<td>Perimeter</td>
<td>( \sigma )</td>
<td>Pipe roughness</td>
</tr>
<tr>
<td>( Q )</td>
<td>Volumetric flow rate</td>
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<td>Slip ratio</td>
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<td>Submergence ratio</td>
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<td>( u )</td>
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### Subscript:

- \( a \) Atmospheric pressure
- \( g \) Gas phase
- \( l \) Liquid phase
- \( s \) Slug
- \( w \) Wall
- 1 Injection location
- 2 Outlet location
Chapter 1 - Introduction

The aquaculture industry has been growing rapidly and is continuing to grow. Half of the global fish supply was produced by aquaculture in 2008 and the industry continues to grow by 5.8% annually (Bostock et al., 2010). More efficient systems are in demand for aquaculture facilities to keep up with this rapid growth. Energy efficiency is an important aspect in making these facilities more effective, and airlift pumps could provide a solution for this issue. A reliable and scientific way to integrate airlift pumps into an aquaculture system has yet to be achieved. This study aims to provide a robust and accurate model to implement airlift pumps into any aquaculture system. As well, experimental results from both a laboratory and a real world setting will be obtained to validate this model.

An airlift pump is a pump that relies on the expansion of gas to push a liquid through a vertical pipe. An airlift pump does not depend on rotational energy or moving parts to push the liquid, but an air source is required to operate it. Pressurized gas is injected into the submerged level of a vertical pipe. Due to the change in density, the buoyancy forces will lift the liquid towards the exit of the pipe. The simplicity of airlift pumps gives the technology a great advantage over conventional centrifugal pumps. A gas-liquid mixture or gas-liquid-solid mixture can be pumped by airlift pumps. When compared to common centrifugal pumps, airlift pumps can mix gases and liquids and transport them. Energy costs of operating airlift pumps are significantly less than that of conventional pumps as well.

There are other advantages to using an airlift pump over conventional centrifugal pumps. Airlift pumps excel in situations in which rotating blades would be undesirable, such as the pumping of sensitive fluid mixtures. Another situation is when chemical reactions could occur from corrosive agents in the fluid and could cause damage to a centrifugal pump’s metal blades and components. These cases are ideal for airlift pumps and there could be other scenarios as well in which gas injection would be preferred instead of traditional pumps to move fluids in a system. Airlift pumps would not face these issues that centrifugal pumps encounter (Kassab et al., 2009). Airlift pumps have a lower installation and maintenance cost compared to other pumps. The pumps
are portable and can fit in small spaces making them simpler to integrate into systems (Kassab et al., 2009; Castro et al., 2009).

Due to the simplicity of their design, airlift pumps can be integrated into any industrial system that requires transportation of multi-phase fluids. Many systems benefit from procedures where gas and liquids are mixed and moved within the system. The mining industry has historically been utilizing airlift pumps and is where the concept originated in the 1700’s (Castro et al., 1975). Airlift pumps should be used in any system that requires the vertical transfer of multiphase fluids. In recent years, airlift pumps have been used successfully in a diverse number of systems, such as oil and gas, chemical, sea mining, water waste treatment, power generation industry, aquaculture, aquaponics, and more. The applications of airlift pumps are vast, and some examples from the literature are discussed below.

Airlift pumps have been used effectively in the oil industry from as early as 1847 (Parker and Suttle, 1987). A slurry of oil, natural gas, and sediment particles can be pumped simultaneously by airlift pumps. They are mostly used in shallow wells. Air or natural gas from the well could be used to pump the oil to the surface. Recently, new air injection methods and gas pulsation methods were introduced by Badr and Ahmed (2012). These well-engineered airlift pumps are designed to achieve higher standards of efficiency for pumping oil. High flow rates are achievable even for very deep wells. This can be done by having multiple airlift pumps located in a series on a vertical pipe. A computer is used to monitor and regulate the gas pulsation rate into the airlift. Airlift pumps excel at running at a low head and high pressures, as can be the case for shallow wells (Ahmed et al., 2016; Ahmed and Badr, 2012; Kassab et al., 2009; Mahrous, 2014).

Deep-sea mining is another field that has been using airlift pumps since the 1970s (Ma et al., 2017). Ocean mining was investigated and it was concluded that airlift pumps could be profitable for small particle excavation because of their ability to push these particles vertically through seawater by air injection. Pipe diameter plays a big role in the pumping of solid particles. Better results could be achieved with a larger diameter of pipe. However, when trying to pump from a depth lower than the maximum applicable sea level, the airlift quickly becomes inefficient. Also, moving smaller particles is more feasible than moving larger particles in terms of the energy
consumption of the system. Under the right conditions and constraints, airlift pumps are more economical and efficient than conventional mining technologies used in deep-sea mining (Fan et al., 2013; Stanulla et al., 2016). Airlift pumps have also been used in aquatic research to retrieve seawater samples. Sewage treatment facilities use airlift pumps for pumping, aeration and desertification processes. Sewage treatment facilities benefit from the simplicity of airlift pumps, which significantly reduce clogging and maintenance costs when compared to conventional pumps (Parker and Suttle, 1987; Yoshinaga and Sato, 1996).

Another use of airlift pumps is in fluidized bed reactors with multi-phase media. These fluidized beds are usually used in the chemical industry. Hybrid airlift pumps serve these delicate applications because they do not have moving parts that might damage any sensitive membranes. Prończuk et al. (2017) did both an experimental and a theoretical study on the importance of using airlift pumps for these processes. These hybrid airlift pumps have a unique design which does not allow the gas phase and the solid phase to come into contact. This is desirable for some fluidized bed applications. Airlift pumps play a role in direct carbon fuel cell power generation industries when molten carbon-based fuel needs to be transported (Kim et al., 2014). The high-temperature molten carbon fuel could damage conventional pumps; however, airlift pumps can withstand these high temperatures depending on what materials they are built from. This method uses carbon dioxide to push the molten fuel.

Oueslati et al. (2017) performed an experiment to investigate the potential of humidification and dehumidification for desalination processes. This technique utilizes airlift pumps to push the water through the system. The study looked at the effect of submergence ratio, air flow rate and water temperature, and how each of these factors could affect the operation of the system. Their findings noted that airlift pumps create better mass transfer conditions, and that the overall efficiency of the desalination process was improved with the implementation of airlift pumps. Temperature and air flow rate had a strong effect on the system's productivity, and submergence ratio had a small effect on the humidification and dehumidification.

The aquaculture industry can benefit from airlift pumps in many areas. Animal safety risks are reduced significantly, almost negating any potential threat, because there are no moving blades
involved in the system. Airlift pumps also have the advantage of transferring oxygen into the water that the fish require. When air and water are the mediums used in the pump, oxygen from the air would transfer into the water at the injection zone of the airlift. The main function of airlift pumps is to move water and recirculate it in a system. Animal life in aquaculture tanks requires certain oxygen levels and a constant current of water flow to grow in a healthy manner. By using airlift pumps, aquaculture farms could potentially eliminate costly systems, such as centrifugal pumps and air stones, which are currently used to achieve water quality standards. Centrifugal and impeller pumps are used to transport the water, and air stones or diffusers are used to oxygenate the water supply. Both systems could potentially be replaced by airlift pumps, which would reduce the costs and the energy consumption of the aquaculture systems. The current study intends to provide a reliable method to integrate airlift pumps into any aquaculture system by developing a theoretical model that can predict the performance of airlift pumps to a high degree of accuracy.

In this study, experimental data was collected for different cases and compared to theoretical outcomes to test the validity of the proposed airlift pump model. The flow pattern detection method was verified by the experimental data which was collected. This was done to confirm that the range of flow patterns were being chosen properly by the theoretical transition line equations. Two experimental setups were produced to collect data for different sized airlift pumps. An airlift pump with a small diameter was tested using a clear acrylic pipe with an inner diameter of 3.175 cm. The high-speed footage was documented to analyze the flow patterns being created. This confirmed the accuracy of the theoretical flow pattern maps that were generated by the equations. A larger setup was used to accommodate a range of airlift diameter sizes, starting at a diameter of 5.08 cm and continuing until a diameter of 20.32 cm. The pipes used for the larger diameters are made from PVC material. Performance data for each setup was collected using identical submergence ratios and pipe lengths. These large pipe diameter tests were conducted at an aquaculture facility. The tests validated the predicted water flow rates produced by the improved model depending on the amount of air injected.

The main goal of this work was to recognize the liquid flow rate potential of airlift pumps through theoretical and experimental studies with a focus on aquaculture applications. A proposed predictive model for airlift pumps was established and verified through lab experiments and field
trials that were sensitive to flow pattern changes. The end result was a powerful tool to use when designing recirculating systems or any system that requires the movement of any liquid while running airlift pumps at the most efficient settings.
Chapter 2 - Literature Review

This chapter will start by reviewing the theory and background of airlift pumps. Then, an extensive literature review will be presented to address recent experimental and modelling work, as well as how airlift pumps are used in aquaculture applications. Finally, the objectives of this study will be described.

2.1 Theory and Background of Airlift Pumps

The airlift pump is a simple mechanism with no moving parts and involves multi-phase flow physics to pump a fluid mixture. Figure 2-1 shows a schematic diagram of an airlift pump. The components of an airlift pump are a vertical pipe and a gas injector. The vertical pipe is partially submerged in a vessel or in a well that is filled with liquid. The gas is injected at the bottom of the pipe or at a submerged level to create air pockets that pump the gas-liquid mixture to the top of the pipe due to buoyancy forces. The airlift pump is also able to pump solid particles that could be present in the liquid, making the medium a three-phase (gas-liquid-solid) mixture. (Apazidis, 1985; Castro et al., 1975; Fujimoto et al., 2005; Hanafizadeh and Ghorbani, 2012; Khalil et al., 1999; Oh, 1992; Saito et al., 2004). The performance of airlift pumps is impacted directly by the concepts of two-phase and/or multi-phase flow, where interfacial surfaces and flow patterns play a big role in how well an airlift pump will function. Understanding how these parameters affect the performance of the airlift pump is the key to utilizing airlift pumps to their maximum potential.

![Figure 2-1 A schematic of an airlift pump.](image-url)
Different types of flow patterns can occur when two phases mix in a vertical pipe. These patterns have been divided into different categories by various researchers and the names of the patterns vary according to whom has categorized them. Each research group has its own terminology to differentiate types of flow patterns. The most recognized definitions of flow patterns were documented by Hewitt and Roberts (1969), who made a thorough study of two-phase flows in vertical pipes using x-rays and flash photography. Their five main flow patterns include: bubbly flow, slug flow, churn flow, wispy annular flow, and annular flow. This work will use the flow patterns classified by Hewitt and Roberts (1969).

Bubbly flow is defined as separate bubbles which flow through a column with minor coalescences, and flow through the medium semi-homogenously. In contrast, the definition for slug flow is as follows “bubbles which have nearly the same diameter as the tube and which have a characteristically rounded front, move along the tube separated by liquid slugs which may or may not contain a dispersion of smaller bubbles” (Hewitt and Roberts, 1969). In airlift pumps, the crucial flow pattern is the slug flow pattern, and a pump should be at its most efficient while operating in the slug flow pattern. When in slug flow, the air bubbles are almost the same size as the pipe diameter, making them ideal for pushing the liquid up the pipe as they act like pistons, pumping the water to the top of the pipe. Reverse flows in the liquid are also minimized in this flow pattern making it the most desirable flow pattern for airlift pumps.

Certain flow patterns would not be ideal for the operation of airlift pumps. For example, if the pump experienced annular flow, most of what would come out at the top of the pipe would be air because most of the liquid would be pushed aside to the pipe walls. The annular flow pattern is achieved when the air column becomes one continuous stream throughout the pipe riser. This occurs when the pipe's cross-section is identical or similar throughout the pipe with no obstructions. This would be the least ideal case in which to operate airlift pumps.

The other flow pattern that would be effective to some extent would be the churn flow pattern. The churn flow pattern occurs when flow velocities increase, causing chaotic motion in the fluids. The bubbles’ shape becomes more unpredictable, and both oscillation and unstable slugs
occur. The gas, the liquid, and the shapes can vary dramatically with a change in diameter and flow velocities (Cachard and Delhaye, 1996; Fernandes et al., 1983; Hanafizadeh et al., 2011; Hewitt and Roberts, 1969; Mahrous, 2013; Nakoryakov et al., 1981; Nakoryakov et al., 1986; Serizawa et al., 1975).

As an airlift pump experiences lower to higher air flow rates, it will pass through different flow rates, starting with bubbly flow, then passing through slug flow, churn flow and annular flow respectively. As the air flow rate increases, these patterns will occur and there will be transition phases between them. The submergences ratio, pipe diameter, phase densities, pressure, and superficial velocities will all play a role in when and how the pattern will change, which in turn will affect the performance of the airlift pump. To have a better understanding of these effects, experimental investigations must be conducted.

2.2 Experimental Work

Experimental studies are necessary to obtain a better understanding of the operating conditions of airlift pumps. Experimental investigations are currently the most reliable method to study airlift pump behavior and changes in a system. Several experimental studies have been conducted to investigate the effects of different parameters on airlift pumps. Airlift pump
performance can be affected by two main sets of parameters: geometrical parameters, and phase parameters. Geometrical parameters could be the airlift pump geometry itself, such as diameter, air injection hole size, number of holes, and placement of holes. The geometry of the system itself could also be considered, the submergence ratio which is defined as the submerged length of the pipe divided by the total pipe length. The equation of the submergence ratio is as follows:

$$Sr = \frac{H_s}{L}$$  \hspace{1cm} (1)

where $H_s$ length is static head of water or the length of the pipe submerged in the water. $L$ is the total length of the pipe.

Phase properties and injection methods affect the flow pattern and multi-phase flow properties. Slip ratio and void fraction, for example, will affect the pump's performance. Slip ratio is defined as the ratio between the velocity of the gas phase over the velocity of the liquid phase. Void fraction is defined as the fraction of the channel volume that is occupied by the gas phase. In the literature, researchers have been experimenting on how changing these parameters will affect the efficiency of the pump.

Kim et al. (2014) conducted an experimental study on how pipe diameter and submergence ratio affected the performance and flow patterns of airlift pumps in direct carbon fuel cells. These systems would convey molten carbon fuel using carbon dioxide gas. In the study, however, air and water were used instead to simulate the conditions. The diameters that were investigated ranged from 8 mm to 24 mm. Submergence ratios of 0.8 to 1.0 were observed in these experiments. These high submergence ratios and diameters were chosen in order to more closely simulate the operating conditions of a direct carbon fuel cell system. As the submergence ratio increased and the pipe diameter decreased, it was noted that water flow rates increased. It was also noted that the initial water discharge was similarly affected. They would occur at smaller air flow rates when the diameters were smaller, and would occur later if the diameters were increased. Air flow rates would always have a limit at which the water flow rates would plateau no matter how much more air was added (Kim et al., 2014). A recent study conducted by Wang et al. (2017) addressed the
effects of submergence ratio on the performance of airlift pumps. They tested submergence ratios ranging from 0.3 to 0.7 Sr. The results confirmed that with an increased submergence ratio, there was an increase in the water flow rates produced. This is in agreement with Tighzert et al. (2013) study on the effects of submergence ratio in an airlift pump. The higher the submergence ratio the better the airlift pump will perform.

Hanafizadeh et al. (2014) performed experiments which investigated a non-dimensional parameter that would give a good representation of airlift pump performance characteristics. Experimental work was conducted on a 50 mm diameter airlift pump with a height of 6 m. The results showed that pump curves could be produced for airlift pumps that were independent of the submergence ratio. The study concluded that the slip ratio between the phases was closely related, and changed depending on the flow pattern. It was also found that when the slip ratio increased, the efficiency decreased dramatically.

A study conducted by Castro et al. (1975) looked at small diameter and short length airlift pumps and documented the findings. The study tested airlift pumps with diameters ranging from 1.27 cm to 7.62 cm, and lengths from 30 cm to 3.7 m. It was noted that the maximum water flow rate happens at a specific air flow rate and occurs after the maximum efficiency point. It was also noted that when the total length was increased, the flow rates achieved by lower lift airlift pumps were closer to the flow rates achieved by high lift airlift pumps. The results were compared with previous findings and found to be in good agreement with those studies.

Kouremenos and Staïcos (1985) studied airlift pumps with smaller diameters. The diameters that were observed were 12 mm, 14.50 mm, 16 mm and 19.23 mm. The experiments were performed in a perfect slug flow pattern. An equation was developed that represented the airlift pump performance and included reversal effects in the flow pattern. Most experimental studies conducted in laboratories use small diameter airlift pumps rather than large airlifts. The current study aims to fill the gap regarding larger airlift pump investigations. (De Cachard and Delhaye, 1998; Nakoryakov et al., 1986; Reinemann et al., 1990; White, 2001)
Experiments using vertical tubes with a mixture of water and air can also be useful in understanding the characteristics of flow patterns in airlifts. An experimental study, which analyzed an upward glass tube of 27.6 mm containing an air-water mixture, was conducted to test the fluctuations in the void fraction when in slug flow. A capacitance sensor was used to document the void fraction data. This experiment gave a better understanding of how slug flow behaves and validated slug flow correlations that could be used for modelling airlift pumps (Akagawa and Sakaguchi, 1966).

Researchers have looked at trying to influence the flow patterns created in airlift pumps by using a few different methods. One successful method was performed by Ahmed et al. (2016). In this method, the air injected at the bottom of the pump was manipulated by different injectors. The four injectors that were tested were radial injection, axial injection, swirl injection and dual injection. The results of the study showed that dual injection, which is a combination of radial and axial injection, performed better than the rest. The study also briefly mentioned the pulsation of air, which it found to be beneficial (Ahmed et al., 2016). These air injection technologies are patented by Badr and Ahmed (2012) and have shown a promising future for aquaculture applications.

A study conducted by Alasadi and Habeeb (2017) looked at how different air injection angles could affect the performance of an airlift pump. The airlift that was tested had a diameter of 42 m and a length of 2.2 m. The angles of air injection that were tested ranged from 90° to 22.5°. It was found that an angle of 22.5° was the optimum operational setting. That angle increased the performance by approximately 11% and there was no noticeable change in flow pattern structure when the injection angle was altered. In addition, it was found that the airlift operated best while it was in the slug and slug-churn flow patterns. Another study using injection methods to improve airlift pump performance was conducted by Cho et al. (2009). This study focused on improving airlift pumps with air jet nozzles that had an angle of 60° and using booster pumps to increase the pressure of the gas. Their results showed that airlift pumps have a good potential even for high head applications if used this way.
An experimental study on upwelling in deep-sea mining from an airlift pump perspective was conducted by Fan et al. (2013). Different air injector nozzle designs were tested at a submergence ratio of 1.0, meaning that the pipe was fully submerged. The injector was at a depth of 20 m and the pipe had an added suction length of 30 m. Four different designs were used: two cross designs and two circular designs. The difference between each of the circular and each of the cross designs is the number and density of holes on each. It was found that the upwelling capabilities were increased when the pipe diameter was enlarged. This was due to less friction loss. The study concluded that the efficiency and the performance of the airlift pump was highly related to the geometrical parameters, and that more work was needed to design the optimum airlift for an artificial upwelling application.

An experiment was conducted by Abou Taleb and Al-Jarrah (2017). In this study, the injector hole size and the number of holes were changed to study the effects of hole geometry on performance. The study found that the maximum water flow rate does not occur at the maximum efficiency of the pump, which agrees with the findings in the literature. It was also noted that both the efficiency and the water flow rates increased when the submergence ratio increased. Submergences ratios of 0.18, 0.258, 0.336, 0.414 and 0.492 were tested. The most efficient setup with the highest flow rate was a specific geometry of holes in which the hole size was 4 mm in diameter and the injector had 6 by 4 holes on it. A 2 mm and a 6 mm version of this setup was tested as well.

Experimental data for aquaculture-focused airlift pumps are prevalent in the literature as well. The data collected by Wurts et al. (1994) looked at PVC airlift pumps with diameters of 7.6, 10.2, and 15.2 cm. Air injection at three different depths was studied. A volumetric flow rate range of water from 66 to 225 liters per minute was recorded in all the cases. The pumps were tested in different aquaculture settings. Good levels of aeration and destratification of particulates were achieved in floating raceways, open cages, and recirculating systems.

Other aquaculture experiments on airlift pumps have been done (Barrut et al., 2013; Castro et al., 1975; Froehlich et al., 2016; Gephardt et al., 2009; Loyless and Malone, 1998; Mcgee and Cichra, 2000; Parker and Suttle, 1987; Reinemann, 1987; Wurts et al., 1994). Each research group
stated that airlift pumps have great potential for the aquaculture industry, and that improved airlift pumps would be desirable to increase the overall efficiency of the current systems.

2.3 Modelling Airlift Pumps

Predicting the performance of airlift pumps is a challenge. Several approaches to modelling airlift pumps under different conditions are available in the current literature. Researchers use methodologies of modelling which contain variations on the two-phase separated flow model, or simulation and CFD could be used to predict airlift pump performance. However, each researcher has adapted the model for a particular application. What these models have in common is that they rely on principle equations: mass, energy, and momentum are usually the preliminary steps in all of the airlift models. The most prevalent method of modelling found in the literature uses a variation on the separated flow/drift flux method of analysis. The mechanism of the airlift pumps is dramatically influenced by two-phase or multi-phase flow concepts. Interfacial surfaces and flow patterns play a big role in how well an airlift pump will function. Understanding how these parameters affect the performance of the airlift pump is the key to utilizing a pump to its maximum potential. (Kassab et al., 2001; Nenes et al., 1996; Saito et al., 2004; Taha and Cui, 2006). The progress of airlift pump modelling has been largely assisted by developments in vertical two-phase modelling has been executed by multiple researchers in the literature (Akagawa and Sakaguchi, 1966; Hatta et al., 1998; Shipley, 1984).

Clark and Dabolt (1986) developed an airlift pump model that was experimentally validated using a small diameter airlift. The approach taken was a deferential momentum on a closed pipe system in which the full column of the two-phase mixture was occupied. It was assumed that the flow was one-dimensional and that the flow pattern was constantly in slug flow throughout the operating range of the airlift. The model that was developed could not solve for the water mass flow rate directly and needed to be approximated iteratively to predict the correct solution.

The Clark and Dabolt (1986) model was improved on by Kassab et al. (2009). This model could predict water flow rates produced by airlift pumps due to the amount of air injected into the airlift pump. The model is built on momentum, mass, and energy equations and relies on two-
phase parameters, such as two-phase friction factors and slip ratio. This model has proven to be adequate when approximating performance. Its flaw, however, is that it only considers the airlift pump to be running in one flow pattern, which is slug flow, and the slip ratio correlations of the model are set to that particular flow pattern. When in the slug flow pattern, which is the most ideal case for operating an airlift pump, the buoyant forces are maximized. In reality, airlift pumps run in multiple varying flow patterns depending on what airflow rate is injected. The pattern could be bubbly flow, slug flow, churn flow, annular flow or any flow pattern in between. Implementing flow pattern detection methods into the model would be advantageous as it would produce more accurate predictions of flow rates in airlift pumps. These other flow patterns and their respective correlations should be considered in the model as well for higher predictive accuracy.

Reinemann et al. (1990) developed what is now a well-established model for small diameter airlift pumps. Using the separated flow model, Reinemann established a method to predict the water flow rates produced by an air flow rate input. However, the model was limited as it was only viable with a diameter in the range of 3 mm to 25 mm. It also only considered slug flow pattern correlations, making it inaccurate for different flow patterns.

A numerical simulation was successfully performed by Kajishima and Saito (1996) to study the unsteady flow of airlift pumps for deep-sea mining applications. A discretized finite difference method of the drift-flux model was applied to an airlift pump system with a depth of 200 m. A good model was achieved that agreed with some experimental results. Taha and Cui (2006) attempted to create a computational fluid dynamics approach for Taylor bubbles in vertical pipes. As mentioned before, Taylor bubbles in the slug flow pattern is the flow pattern in which airlift pumps are most efficient.

Models of different flow patterns in vertical pipes are useful for molding airlift pumps since due to the nature of airlifts, the models of vertical pipes can be applied onto them. Guet et al. (2004) proposed a model of vertical pipe bubbly flows, and how this flow pattern influences drift flux parameters. This model could be a good tool for predicting performance of low flow airlift pumps in which bubbly flow conditions are dominant. A study conducted by Brauner and Barnea (1986) investigated predicting the transition phase from slug to churn flow in a vertical pipe. A
simple model was proposed which denoted that the flow pattern will transition from slug to churn when a void fraction of 0.52 is reached.

Theoretical and empirical flow pattern maps of two-phase flow in vertical pipes are valuable information to have when modeling airlift pumps. Many researchers (Taitel et al., 1980; Hatta et al., 1998; Hewitt and Roberts, 1969) have done studies which have been valuable in understanding the transition of flow patterns. The work conducted by Taitel et al. (1980) on theoretical transition lines in vertical tubes is considered one of the most trusted methods of predicting when flow patterns change. This model will be used to detect flow patterns in this study.

A theoretical flow pattern detection method will be implemented in the predictive model to improve it. After detection, the appropriate flow pattern correlation will be used for its respective domain of air injection. After identifying each flow pattern region at which the airlift pump will be operating, the model will be modified to better fit these flow patterns. The motivation of doing this research is to have a more reliable and robust theoretical airlift pump model that can predict pump performance under any conditions, permitting airlift pumps to be applied in any aquaculture systems. Depending on the water flow rate requirements, the size limitations and geometry of a system, the model can be adjusted for the required outcome. Therefore, an appropriate size airlift pump can be integrated into the system based on the predicted flow rates from the proposed model. The model will allow for different pipe diameter size, pipe length, submergence ratio, and different fluid properties so that any case scenarios could be recreated by the Flow pattern identification will be done through theoretical flow pattern maps that will be generated by the transition line equations developed by Taitel et al. (1980). While that study developed theoretical flow pattern maps for vertical pipes, the theoretical equations in this study will depend on pipe geometries as well as bubble size and shape.

2.4 Use of Airlift Pumps in Aquaculture Applications

Airlift pumps could have a substantial impact on the aquaculture industry. Initial installation, energy consumption, maintenance, aeration, and other aspects of the aquaculture facility would all
be affected positively when introducing airlift pumps into the system. The use of airlift pumps in aquaculture is mentioned by many sources throughout the literature (Barrut, Blancheton, Callier, Champagne, and Grasmick, 2013; Castro et al., 1975; Froehlich, Gentry, and Halpern, 2016; Gephardt, D’aloia, and Ndonga, 2009; Loyless and Malone, 1998; Mcgee and Cichra, 2000; Parker and Suttle, 1987; Dougal Joseph Reinemann, 1987; Wurts et al., 1994). The findings of these studies will be discussed in this section.

The efficiency of the aquaculture industry is much greater than that of the meat production industry, mainly because the feed-to-produce ratios are superior when dealing with fish. The energy consumption of aquaculture systems is more to the scale of producing rice or wheat than other animal-based production. This is one of the reasons why in recent years the demand for more fish farms and other aquaculture facilities has increased globally (Reinemann, 1987). Exponential growth has been noticed in aquaculture over the last five decades and it now contributes to 50% of all global seafood production. With all of this recent growth, it still does not seem to slow down the demand for seafood worldwide (Froehlich et al., 2016). The productivity of aquaculture facilities relies heavily on circulation and aeration of the fish tanks. The movement of the water and the addition of air is necessary to decrease the stratification of harmful particles, improve the solubility of nutrients, and lessen the accumulation of organic material at the lower levels of the tank. These factors all improve the production and quality of the livestock (Parker and Suttle, 1987).

A study was conducted on an endangered species of fish named the Rio Grande silvery minnow. In this study, airlift pumps were implemented to circulate and aerate a circular tank that contained the fish. Oxygen levels in these tanks must be kept at 5 mg/L for the fish to be healthy; the airlift pumps provided enough dissolved oxygen that the levels never dipped under 6.30 mg/L. The survival rate of the fish exceeded expectations and the use of airlift pumps was proven to be a safe and inexpensive solution for the required application (Tave et al., 2012).

The application of airlift pumps in different types of fish tanks is common: aquarium tanks, artificial fish ponds, open cage floating raceways, closed recirculating systems and other fish enclosures all use airlift pumps to eliminate stratification of particles and circulate water. Wurts
et al. (1994) conducted a study to better understand the airlift pumps that are used in industry by observing airlift pumps with diameters of 7.6 cm, 10.2 cm and 15.2 cm and a total length of 185 cm. The range of flow rates achieved were from 71 L/min to 324 L/min. In the findings, all applications of airlift pumps in these fields were positive and desirable for the industry.

As mentioned before, aeration is crucial in aquaculture for the oxygenation of the tanks. Sources from the Department of Fisheries and Aquatic Sciences in Florida state that dissolved oxygen levels should be over 60% in most cases for fish farms. Most airlift pumps meet these criteria (Mcgee and Cichra, 2000). Another study conducted by Barrut et al., (2013) showed that vacuum airlift pumps were quite effective at evacuating undesirable particles, such as uneaten food and feces, from the water in aquaculture tanks.

Fish farms are not required to pump a high head of water from the surface of the tanks. In aquaculture it is more important that the water is in constant motion and that a sufficient current is achieved. As stated earlier, the mass transfer of oxygen into the water is crucial for aquaculture facilities. In recirculating aquaculture systems, multiple parameter criteria must be satisfied for any designated species to grow in a healthy and safe manner. Some parameters are more crucial than others depending on how long the livestock can handle insufficiencies in the system. Oxygen level is the first and most important parameter. It must be kept over a certain level or it could lead to disease, stress or even loss of stock within a few minutes. Other important water quality parameters in recirculating aquaculture systems include carbon dioxide, ammonia, alkalinity and pH levels. These factors can affect the aquatic life’s health depending on the species, and can affect some species more than others. Loyless and Malone (1998) discussed water quality parameters, and made an in-depth study regarding how effectively airlift pumps could be used to maintain all the required levels for the aquaculture system. Experiments were conducted with a particular setup; the airlift had either an air-stone as an injector or no air-stone at the inlet of the pump. The goal was to achieve and report on how satisfactory gas transfer levels could be when using only the airlift. The results showed that airlift pumps with air-stones do transfer more oxygen and other gases but require more power to perform in an open tank setting. Airlift pumps with no air-stones have lower gas transfer rates. They are, however, much more effective at moving the water in the tank, which is also an important parameter to have in recirculating aquaculture
systems. These kinds of pumps are designed to compensate with higher flow rates and can still achieve a reasonably promising amount of gas transfer. The amount will depend on how many are installed in the tank and what air flow rates are injected into them. A study done by Loyless and Malone (1998) achieved a dissolved oxygen level of more than 6mg/L and a dissolved carbon-dioxide level of less than 5mg/L by running ten airlift pumps with flow rates of 113L/min. These results showed that airlift pumps are more than capable of moving water and ensuring that gas levels are adequate for the fish to thrive within the aquaculture system. Airlift pumps that are optimized in design and operation could replace other methods of achieving these water qualities. Open tank aerators or centrifugal pumps could all be replaced by one system, reducing power costs and improving the overall economic revenue. This result could be achieved for these systems if airlift pumps alone are adopted by industries to achieve multiple goals at once.

2.5 Objective of the Study

As concluded from previous studies, flow pattern has a strong effect on an airlift pump's performance. Therefore, a main goal of this of this work is to develop a model that considers the effect of flow pattern and determines the airlift pump's performance accordingly. The motivation for doing so is to have a more reliable and robust theoretical airlift pump model that can predict performance so that airlift pumps could be utilized in any sort of system, including aquaculture systems. Depending on the water flow rate requirements, the size limitations and the geometry of the system, the model can be adjusted for the required outcome. An appropriately sized airlift pump can therefore be integrated into the system based on the predicted flow rates that can be achieved by the proposed model. This will permit a method to predict airlift pump performance that is flow pattern dependent. The following are the specific objectives that this work intends to achieve. The main objective of this study is to set out and evaluate the performance of an airlift pump, which has any pipe length, diameter and submergence ratio, at all stages of air injection. Submergence ratio is the ratio between the level of the static head in the pipe over the overall length. As the air injection changes, so does the two-phase flow pattern, and this must be taken into account while modelling.
Airlift pumps are considered one of the simplest pumps to manufacture; however their flow characteristics are complex to study. One factor that arises when modelling airlift pumps is slip ratio, which is the interaction at the water-air interface where slippage occurs. This slip ratio will constantly change as more air is added to the system due to changes in the flow pattern. The slip ratio correlation should therefore change for each flow pattern type. Because of this complex behavior, most models assume the flow pattern of the system to be a slug flow pattern for simplicity. To achieve higher accuracies in flow rate predictions, all slip ratio correlations of different flow patterns will be considered in this work. The purpose of this study is as follows:

1. To develop a method to predict an airlift pump's performance and associated parameters using different correlations that take into account the flow pattern of the gas and liquid.
2. To evaluate the airlift pump's performance in various aquaculture systems, such as raceways, open cages, and recirculating systems.
Chapter 3 - Theoretical Analysis Mathematical Modelling

Analysis of a two-phase flow in the pump riser is done by solving the basic governing equations, including the conservation of mass, energy and momentum. The equations are usually solved in a one-dimensional manner by using various assumptions or models. Common models used to solve the governing equations are the homogenous model, the separated flow model and the drift flux model, which is a type of separated flow model. In this study, two-phase flow general principles will be covered first, then the study will move into the simplified separated flow model of airlift pumps. A flow pattern dependent model will be developed to integrate various slip ratio correlations depending on the flow pattern in the pump riser. In this study, a drift flux model will be used to determine the two-phase flow characteristics of the pipe riser of the airlift pump.

3.1 Conservation of Mass

The general equation for mass conservation with a system of coordinates consisting of the z axis and θ axis along a flow channel is as follows:

\[
\frac{\partial}{\partial t}(A_k \rho_k) + \frac{\partial}{\partial z}(A_k \rho_k u_k) = \Gamma_k
\]  

(2)

\(\alpha_k\) represents the average void fraction of phase \(k\) over time \(t\), and \(A_k\) represents the cross sectional area of phase \(k\). Void fraction is defined as the fraction of the channel volume that is occupied by the gas phase. The density and the mean velocity of phase \(k\) are represented respectively by \(\rho_k\) and \(u_k\). \(\Gamma_k\) is the generation rate over a unit length of mass for phase \(k\) between the interface of the two phases. In steady state situations, in which gas and liquid are in a constant area flow channel, the equations become.

\[
\frac{d}{dz} (A_g \rho_g u_g) = \Gamma_g
\]  

(3)

\[
\frac{d}{dz} (A_l \rho_l u_l) = \Gamma_f
\]  

(4)

\[
\Gamma_g = -\Gamma_f = \frac{dW_g}{dz} = -\frac{dW_l}{dz}
\]  

(5)
where $W_g$ and $W_l$ represent the mass flow rate of the gas phase and the liquid phase respectively.

### 3.2 Conservation of Momentum

Momentum generation along the flow channel of phase $k$ in a controlled volume added to the rate of inflow momentum will counteract the sum of the forces on phase $k$ in the control volume. The momentum conservation can be written as follows:

\[
\frac{\partial}{\partial t}(W_k \delta z) + \left( W_k u_k + \delta z \frac{\partial}{\partial z} (W_k u_k) \right) - W_k u_k = 0 \tag{6}
\]

\[
\frac{\partial}{\partial t}(W_k \delta z) + \delta z \frac{\partial}{\partial z} (W_k u_k) \tag{7}
\]

where

\[ W_k = A \alpha_k \rho_k u_k \tag{8} \]

when balancing the summation of forces in the $z$ direction on the phase $k$, the momentum generated by the mass transfer is as follows:

\[
\left[ A \alpha_k p_s - \left( A \alpha_k p_s + \delta z \frac{\partial}{\partial z} (A \alpha_k p_s) \right) - \left\{ p_s \left( -\delta z \frac{\partial}{\partial z} (A \alpha_k) \right) \right\} - A \alpha_k \rho_k \delta z g \sin \theta - \tau_{kw} P_{kw} \delta z \right]
+ \sum_{i=1}^{n} \tau_{knz} P_{kn} \delta z + u_k \Gamma_k \tag{9}
\]

where $p_s$ is the static pressure, $\tau_{kw}$ is the shear stress between the wall of the flow channel and phase $k$, and $P_{kw}$ is the perimeter of the wall that is in contact with phase $k$. When relating the forces to the generation of momentum the outcome is as follows:

\[
-A \alpha_k \frac{\partial p_z}{\partial z} \delta z - \tau_{kw} P_{kw} \delta z + \sum_{i=1}^{n} \tau_{knz} P_{kn} \delta z - A \alpha_k \rho_k \delta z g \sin \theta + u_k \Gamma_k = \frac{\partial}{\partial t} (W_k \delta z) + \delta z (W_k u_k) \tag{10}
\]
When considering steady state conditions for the phases of gas and liquid respectively:

\[-A_g dp - \tau_{gw} P_{gw} dz + \tau_{gl} P_{gl} dz - A_g \rho_g dz g \sin \theta + u_g \Gamma_g = W_g du_g \quad (11)\]

\[-A_g dp - \tau_{gl} P_{gl} dz + \tau_{ig} P_{ig} dz - A_l \rho_l dz g \sin \theta + u_l \Gamma_l = W_l du_l \quad (12)\]

When momentum is conserved, Equation (12) must be true.

\[\tau_{gl} P_{gl} dz + u_g \Gamma_g = \tau_{ig} P_{ig} dz + u_l \Gamma_l \quad (13)\]

Therefore, considering Equation (12) and adding equations (10) and (11), the result is as follows:

\[-Adp - \tau_{gw} P_{gw} dz - \tau_{lw} P_{lw} dz - g \sin \theta [A_l \rho_l + A_g \rho_g] = d(W_l u_l + W_g u_g) \quad (14)\]

Equation (13) is the general format for the basic differential momentum equation that utilizes the simplified one-dimensional method.

### 3.3 Conservation of Energy

The starting point to derive the general differential energy equation is by relating the total energy increase of phase \( k \). This includes internal and kinetic energy in addition to external energies such as heat and work, while also adding the energy that crosses the interface, all within a controlled volume. The resulting relation is as follows:

\[
\frac{\partial}{\partial t} \left[ \alpha_k \rho_k \left( \varepsilon_k + \frac{u_k^2}{2} \right) A \delta z \right] + W_k \left( \varepsilon_k + \frac{u_k^2}{2} \right) \delta z - \left[ W_k \left( \varepsilon_k + \frac{u_k^2}{2} \right) - \delta z \frac{\partial}{\partial z} W_k \left( \varepsilon_k + \frac{u_k^2}{2} \right) \right] \quad (15)
\]

where \( \varepsilon_k \) is the internal energy of phase \( k \) per unit mass. The addition of heat into phase \( k \) in the control volume is given by the following:

\[
\phi_{kw} P_{kw} \delta z + \sum_{i=1}^{n} \phi_{kn} P_{kn} \delta z + \phi_k \alpha_k \delta z \quad (16)
\]
where $\phi_{kw}$ is the heat transferred from the walls of the flow channel to phase $k$. Then $\phi_k$ is the heat generated internally for phase $k$. The work done on phase $k$ in the control volume is represented by the following:

$$\left[ \frac{W_k p_s}{\rho_k} - \left( \frac{W_k p_s}{\rho_k} + \delta z \frac{\partial}{\partial z} \left( \frac{W_k p_s}{\rho_k} \right) \right) \right] - W_k g \sin \theta \delta z - p_s A \delta z \frac{\partial \alpha_k}{\partial t} + \Gamma_k \frac{\delta z p}{\rho_k} + u_k \sum_{1}^{n} \tau_{kn} p_{kn} \delta z \quad (17)$$

In Equation (16) the first term is the work done by the pressure force. The work done by the body forces is represented by the second term, while the work done by shear forces and pressure at the interface of the phases are the remainder of terms. The last term, however, is related to the heat added by friction on the flow channel walls. Usually, this can only be neglected if the velocities are substantial. The final energy term added to phase $k$ is done by mass transfer across the interface and is represented by the following:

$$\Gamma_k \delta z \left( \varepsilon_k + \frac{u_k^2}{2} \right) \quad (18)$$

Adding Equations (15), (16) and (17) and equating the summation to Equation (14), the following is achieved:

$$\frac{\partial}{\partial t} A \alpha_k \rho_k \left( \varepsilon_k + \frac{u_k^2}{2} \right) + \frac{\partial}{\partial z} W_k \left( i_k + \frac{u_k^2}{2} \right) \quad (19)$$

$$= -W_k g \sin (\theta) + \phi_{wk} p_{wk} + \sum_{1}^{n} \phi_{kn} p_{kn} + \phi_k A \alpha_k - p_s A \frac{\partial \alpha_k}{\partial t}$$

$$+ \Gamma_k \left( i_k + \frac{u_k^2}{2} \right) + u_k \sum_{1}^{n} \tau_{kn} p_{kn}$$

where $i_k$ represents the enthalpy per unit mass of phase $k$ and is equated to be as follows:

$$i_k = u_k + \frac{p_s}{\rho_k} \quad (20)$$
When considering the steady state for a two-phase flow of gas and liquid with no internal heat generation, \((\phi_k = 0)\), and in a constant area flow channel or pipe, the following is achieved with these assumptions:

\[
\begin{align*}
\frac{d}{dz} \left( W_g \left( i_g + \frac{u_i^2}{2} \right) \right) + W_g g \sin \theta \delta z &= \phi_{wg} P_{wg} \delta z + \phi_{gl} P_{gl} \delta z + u_g \tau_{gl} P_{gl} \delta z + \Gamma_g \delta z \left( i_g + \frac{u_i^2}{2} \right) \quad (21) \\
\frac{d}{dz} \left( W_l \left( i_l + \frac{u_i^2}{2} \right) \right) + W_l g \sin \theta \delta z &= \phi_{wl} P_{wl} \delta z + \phi_{lg} P_{lg} \delta z + u_l \tau_{lg} P_{lg} \delta z + \Gamma_l \delta z \left( i_l + \frac{u_i^2}{2} \right) \quad (22)
\end{align*}
\]

Summing Equations (20) and (21), the energy across the interface of the two phases is achieved.

\[
\Gamma_g \left( i_g + \frac{u_i^2}{2} \right) + \phi_{gl} P_{gl} + u_g \tau_{gl} P_{gl} = \Gamma_l \left( i_l + \frac{u_i^2}{2} \right) + \phi_{lg} P_{lg} + u_l \tau_{lg} P_{lg} \quad (23)
\]

The concluded deferential equation is as follows:

\[
\frac{d}{dz} \left[ W_g i_g + W_l i_l \right] + \frac{d}{dz} \left[ \frac{W_g u_i^2}{2} + \frac{W_l u_i^2}{2} \right] + (W_g + W_l) g \sin \theta = \phi_{wl} P_{wl} + \phi_{wg} P_{wg} \quad (24)
\]

These derivations of the general principle equations were established by Collier and Thome (1994) in the book “Convective Boiling and Condensation” in Chapter 2.

### 3.4 Simplified Two-Phase Flow Approach

To solve general two-phase equations, an appropriate method must be used. Several assumptions should be made. First, one must assume that the system is steady state, meaning that there are no changes over time, which makes the problem simpler. In this study, one dimensional flow analysis will be considered with the line running through the center of the pipe. Secondly one must assume that the phase densities are constant and do not change. The drift flux model will be used in this study to solve for the momentum, and the continuity equation will be used to model the airlift pump. The drift flux model has been worked on by multiple researchers (Zuber and Findlay, 1965; Wallis, 1969; Ishii 1977).
To model the airlift pump, the momentum equation and the continuity equation are applied to the system in one dimension. The validity of this assumption is confirmed by Clark and Dabolt (1986) for practical uses regarding the airlift pump. The airlift model was develop by Kassab et al. (2009) and Stenning and Martin (1968). The following equations is a derivation of the airlift model.

When analyzing a vertical pipe submerged in water, the pressure at the bottom of the pipe, \( p_o \), is given by Bernoulli’s equation as follows:

\[
p_o = p_a + \rho_l g H_s - \frac{1}{2} \rho_l u_1^2 \tag{25}\]

where \( H_s \) represents the static head level, \( u_1 \) is the mean velocity of the liquid phase at the injection point of the airlift, and \( p_a \) represents the atmospheric pressure. The continuity equation is as follows if the density of the air and the water are assumed to be constant and non-compressible:

\[
A u_2 = Q_g + Q_l = Q_g + Au_1 \tag{26}\]

A rearranged form of Equation (25) is as follows:

\[
u_2 = u_1 \left( 1 + \frac{Q_g}{Q_l} \right) \tag{27}\]

where \( u_2 \) is the velocity of the gas/liquid mixture exiting the injector. Assuming that the air mass flow rate is negligible compared to the liquid mass flow rate, the equation of continuity is as follows:

\[
\rho_2 A u_2 = \rho_l A u_1 \tag{28}\]

This can be written as:

\[
\rho_2 = \rho_l \left( \frac{u_1}{u_2} \right) \tag{29}\]

Substituting Equation (26) into Equation (28):
\[ p_2 = \frac{\rho_L}{\left(1 + \frac{Q_g}{Q_L}\right)} \]  

(30)

Then applying the momentum equation to the injection site at a controlled volume, while neglecting the wall friction results in:

\[ p_2 = p_o - \rho_t u_1 (u_2 - u_1) \]  

(31)

Substituting Equation (26) into Equation (30):

\[ p_2 = p_o - \frac{\rho_t u_1 Q_g}{A} \]  

(32)

If Equations (24) and (31) are then joined, the resulting equation is as follows:

\[ p_2 = p_a + \rho_t g H_s - \frac{1}{2} \rho_t u_1^2 - \frac{\rho_t u_1 Q_g}{A} \]  

(33)

As mentioned by Stenning and Martin (1968), the momentum equation of the upper section of the airlift pump can be written as follows:

\[ p_2 - p_a = \tau \frac{L P}{A} + \frac{w}{A} \]  

(34)

where \( L \) is the total length of the pipe, and \( \tau \) is the averaged shear stress for both phases on the walls of the pipe. This average shear stress is represented by Griffith and Wallis (1961) with the following:

\[ \tau = f \rho_t \left( \frac{Q_l}{A} \right) ^2 \left( 1 + \frac{Q_g}{Q_l} \right) \]  

(35)

The friction factor given by \( f \) assumes that water is the only substance flowing in the pipe. \( w \) is given as the sum of the weight of both the gas and the liquid phases in the system, which is as follows:

\[ w = L (\rho_t A_l + \rho_g A_g) \]  

(36)
\( A_l \) and \( A_g \) are the cross sectional area filled by the liquid and the gas phase respectively.

\[
A = A_l + A_g \tag{37}
\]

\[
Q_g = A_g u_g \tag{38}
\]

\[
Q_l = A_l u_l = A u_1 \tag{39}
\]

\( Q_l \) and \( Q_g \) represent the volumetric flow rate of liquid and gas respectively in the system. \( u_1 \) represents the velocity at the entrance of the injection zone.

Placing Equations (36) through (38) into (35), the result is as follows:

\[
w = L \frac{\rho_l A}{(1 + \frac{Q_g}{Q_l})} \tag{40}
\]

The slip ratio \( s \) is the ratio between the gas velocity over the liquid velocity:

\[
s = \frac{u_g}{u_l} \tag{41}
\]

Placing Equations (34) and (39) into Equation (33) results in the following:

\[
p_2 = p_a + \frac{4fL}{D} \rho_l u_1^2 \left(1 + \frac{Q_g}{Q_l}\right) + \rho_l \frac{L}{1 + \frac{Q_g}{sQ_l}} \tag{42}
\]

The main relationship of the model is shown in Equation (42). This is the model that was developed by Kassab et al. (2009). The geometric parameters that will be constant for each run of the model are the diameter, \( D \), the length of the pipe, \( L \), the static head, \( H_s \), and the lift above the head, \( L_s \). These four parameters will be set before each run of the model, and can be changed depending on the conditions of the airlift pump's geometry. The main equation of the model is then as follows:

\[
\frac{H_s}{L} - \frac{1}{\left(1 + \frac{Q_g}{sQ_l}\right)} = \frac{V_1^2}{2gL} \left[\left(\frac{4fL}{D} + 1\right) + \left(\frac{4fL}{D} + 2\right) \frac{Q_g}{Q_l}\right] \tag{43}
\]
In the current study, the friction factor $f$ developed by Xu et al. (2012) is used. The study focused on the frictional pressure drop for two-phase flow in pipes and developed an equation for finding the friction factor explicitly. The equation for the friction is as follows:

$$f = 0.25 \left[ \log \left( \frac{150.39}{Re^{0.98865}} - \frac{152.66}{Re} \right) \right]^{-2}$$  \hspace{1cm} (44)

The slip ratio, $s$, of the airlift changes as the air injection rate changes. Note that this is the slip ratio correlation. This correlation is only valid when the flow pattern is considered to be slug flow. This is the correlation most commonly chosen to represent the slip ratio in airlift pumps, and was given by Griffith and Wallis (1968). It will be represented as follows:

$$s = 1.2 + 0.2 \frac{Q_g}{Q_l} + \frac{0.35 \sqrt{gD}}{V_t}$$  \hspace{1cm} (45)

For other flow patterns, this term must be augmented to sufficiently predict the slip ratio for the required flow pattern. In the next section of modelling, other slip ratio correlations will be observed after the flow pattern has been identified.

The efficiency of airlift pumps has been well-established over time, and has been considered reliable by most researchers. The efficiency is defined by the work taken to move the liquid to the top of the pipe divided by the isentropic expansion work done by the gas. This efficiency equation was developed by Nicklin (1963) and is as follows:

$$\eta = \frac{\rho g Q_l (L - H_s)}{P_a Q_g \ln \left( \frac{P_1}{P_a} \right)}$$  \hspace{1cm} (46)
3.5 Flow Pattern Dependent Modelling:

Flow pattern detection will be implemented into the current model to better predict the flow rates. This will involve generating a flow pattern map of the airlift while it is running under certain conditions. This map will be able to predict the type of flow pattern within the airlift by depending on the superficial velocity of the gas $J_g$ and the liquid $J_l$. Superficial velocity is defined as the calculated velocity of each phase separately as if that phase was only occupying the channel space instead of both. The flow pattern map will be generated theoretically with equations developed by Taitel et al. (1980).

A scan through the $J_g$ gas superficial velocity axis will be initiated to identify the region in which the airlift pump is operating, which will in turn reveal the flow pattern. From there, depending on which flow pattern is currently apparent in the airlift, the correct corresponding slip ratio will be inserted instead of the slug flow pattern correlation used in the original model by Kassab et al. (2009). This will improve the prediction of the flow rates achieved by the pump.

A theoretical method was developed by Taitel et al. (1980) to predict the flow pattern in a vertical tube instead of using conventional flow pattern maps that are based on experimental results. Two-phase flow pattern maps have conventionally been developed through experimental correlations. A mathematical approach to predicting the flow pattern of the gas-liquid mixture was the outcome of the work done by Taitel et al. (1980). One of the goals of their research is to predict flow patterns without having to scale up or down from maps related to certain pipe sizes. This theoretical method is able to predict flow patterns for any pipe diameter. This information will be useful to detect the flow pattern an airlift pump is operating in.

The Bubbly-Slug Transition

As the void fraction increases and smaller gas bubbles coalesce into bigger bubbles, the flow pattern starts to transition from the bubbly flow pattern into the slug flow pattern. To predict when the transition occurs depends on the void fraction, superficial velocity of both phases, diameter of the pipe and densities. The point at which the bubble size reaches the same size as the diameter of the pipe and the coalescence between the gas bubbles increases is when the bubble-slug transition occurs. The overall relation developed for this transition line is as follows:
\[ J_l + J_g = 4 \left\{ D^{0.429} \left( \frac{\sigma}{\rho_l} \right)^{0.089} \frac{\rho_l}{\nu_l^{0.72}} \left[ g \left( \rho_l - \rho_g \right) \right]^{0.446} \right\} \]

(47)

For bubbly flow to occur, the diameter of the flow channel must meet certain criteria. If the relationship shown in Equation (47) is true, it means that the pipe is too small for the bubbly flow pattern to occur. There will therefore be no transition line, leaving only the possibility of slug flow to occur.

\[
\left[ \frac{\rho_l^2 g D^2}{(\rho_l - \rho_g) \sigma} \right]^{\frac{1}{4}} \leq 4.36
\]

(48)

**The Slug-Churn Transition**

The “Taylor Bubbles” or slugs will start to change into a churn flow pattern once the packing between the slugs gets too compact and the bubbles start to collide with one another and coalesce. This flow pattern is chaotic and random, therefore different researchers have found it hard to categorize to their satisfaction. Defining this transition line is more difficult as there are different intensities of churn pattern in the flow or slug.

\[
l_1 = \frac{l_s U_g}{0.35 \beta \sqrt{gD}} \sum_{n=2}^{\infty} \frac{\beta}{e^{\frac{2n-1}{\beta}}} - 1
\]

(49)

\[
\frac{l_1}{D} = 40.6 \left( \frac{J_l + J_g}{\sqrt{gD}} + 0.22 \right)
\]

(50)

**Transition to Annular Flow**

When the vapor flow rates reach higher levels, the bubbles become a single mass running through the center of the pipe and pushing the liquid to the walls of the pipe. The liquid creeps up
the pipe due to the shear stress and drag force of the vapor phase and becomes a wavy film on the wall of the pipe.

\[
J_g \rho_g^{\frac{1}{2}} \left[ \sigma g (\rho_l - \rho_g) \right]^{\frac{1}{4}} = 3.1
\]

(51)

\[
J_g = \frac{3.1 [\sigma g (\rho_l - \rho_g)]^{\frac{1}{4}}}{\rho_g^{\frac{1}{2}}}
\]

(52)

Table 1 Summary of flow pattern transition lines developed by Taitel et al. (1980)

<table>
<thead>
<tr>
<th>Flow pattern</th>
<th>Transition line</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bubbly-Slug transition line</td>
<td>( J_l + J_g = 4 \left{ D^{0.429} \left( \frac{\sigma}{\rho_l} \right)^{0.089} \frac{g (\rho_l - \rho_g)}{\rho_l} \right} )</td>
</tr>
<tr>
<td></td>
<td>If ( \left[ \frac{\rho_l^2 g D^2}{(\rho_l - \rho_g) \sigma} \right]^{\frac{1}{4}} \leq 4.36 ) is true, no transition</td>
</tr>
<tr>
<td>Slug-Churn Transition line</td>
<td>( \frac{l_1}{D} = 35.5 \left( \frac{U_g}{\sqrt{gD}} \right) )</td>
</tr>
<tr>
<td></td>
<td>( \frac{l_1}{D} = 40.6 \left( \frac{J_l + J_g}{\sqrt{gD}} + 0.22 \right) )</td>
</tr>
<tr>
<td></td>
<td>Assuming ( \alpha = 0.25 ) and ( \frac{l_s}{D} = 16 )</td>
</tr>
<tr>
<td>Transition to Annular flow</td>
<td>( J_g = \frac{3.1 [\sigma g (\rho_l - \rho_g)]^{\frac{1}{4}}}{\rho_g^{\frac{1}{2}}} )</td>
</tr>
</tbody>
</table>
3.6 Integrating the Slip Ratio

Once the flow pattern region has been determined in the flow channel of the airlift pump, the appropriate slip ratio and void fraction correlations can be chosen depending on the flow pattern. General equations are employed to find the void fraction for different flow patterns using two parameters given in the literature: the weighted mean drift velocity \( v_{gj} \) and the distribution coefficient \( C_0 \). The general equations are as follows:

\[
\alpha = \frac{\beta}{C_0 + \frac{v_{gj}}{J_{tot}}}
\]  

(53)

where \( J_{tot} \) is the summation of the gas superficial velocity and the liquid superficial velocity. \( \beta \) is the volumetric quality which is the fraction of the volume flow rates of each phase and is given as:

\[
\beta = \frac{Q_g}{Q_g + Q_f}
\]  

(54)

Because the model that is to be used involves slip ratio correlation and not void fraction, there is a relationship between the two which is:

\[
s = \frac{Q_g - \alpha Q_g}{\alpha Q_l}
\]  

(55)

A study of void fraction correlations used in the drift-flux model was conducted by Coddington and Macian (2002). There are many correlations that were used in this paper, a few of which are mentioned and used in this model. Some of these correlations contain viable criteria; however due to the flow pattern map and the ranges in which they will be used, all criteria are met. The correlations are shown in Table 2. (Hibiki and Ishii, 2003)

**Bubbly Flow Correlations**

Bubbly flow is when small bubbles are randomly and homogenously distributed along the flow channel. The bubbles are too small to coalesce when in this flow pattern. Two void fraction correlations were investigated as follows:
 Slug Flow Correlations

Slug flow occurs when gas bubbles are the same size as the cross section of the tube. The slug flow correlation used in this model is the same correlation used by Kassab et al. (2009), which was a correlation originally developed by Griffith and Wallis (1961).

\[ C_o = 1 \]  \hspace{1cm} (56)

\[ v_{gj} = 1.53(1 - \alpha)^2 \left( \frac{g\sigma \Delta \rho^2}{\rho_l} \right)^{1/4} \]  \hspace{1cm} (57)

\[ C_o = 1.2 - 0.2 \frac{\rho_g}{\rho_l} \]  \hspace{1cm} (58)

\[ v_{gj} = \sqrt{2} \left( \frac{g\sigma \Delta \rho^2}{\rho_l} \right)^{1/4} (1 - \alpha) \]  \hspace{1cm} (59)

(Ishii, 1977)

 Churn Flow Correlations

Churn flow is a chaotic, turbulent and high velocity flow pattern. In this pattern large and small bubbles with unorganized shapes are flowing in the channel. Two void fraction correlations were investigated as follows:

\[ C_o = 1 \]  \hspace{1cm} (61)

\[ v_{gj} = 1.53(1 - \alpha)^2 \left( \frac{g\sigma \Delta \rho^2}{\rho_l} \right)^{1/4} \]  \hspace{1cm} (62)

(Liao et al., 1985)
\( C_o = 1.2 - 0.2 \sqrt{\frac{\rho_g}{\rho_l}} \) \hspace{2cm} (63)

\[ v_{gj} = \sqrt{2} \left( \frac{g \sigma \Delta \rho^2}{\rho} \right)^{1/4} (1 - \alpha) \] \hspace{2cm} (64)

(Ishii, 1977)

**Annular Flow Correlations**

Annular flow occurs when the velocity of the injected air is much higher than the velocity of the water. A continuous column of air is flowing though the center of the channel and the water is pushed towards the wall of the pipe. The water is only pushed up the pipe by the shear forces from the air. Two void fraction correlations were investigated as follows:

\[ C_o = 1 + \frac{1 + \alpha}{\alpha + 4 \sqrt{\frac{\rho_g}{\rho_l}}} \] \hspace{2cm} (65)

\[ v_{gj} = (C_o - 1) \sqrt{\frac{g d_h \Delta \rho (1 - \alpha)}{0.015 \rho_l}} \] \hspace{2cm} (66)

(Liao et al., 1985)

\[ C_o = 1 + \frac{1 - \alpha}{\alpha + \sqrt{[1 + 75(1 - \alpha)] \sqrt{\alpha}} \frac{\rho_g}{\rho_l}} \] \hspace{2cm} (67)

\[ v_{gj} = (C_o - 1) \left[ J + \sqrt{\frac{g d_h \Delta \rho (1 - \alpha)}{0.015 \rho_l}} \right] \] \hspace{2cm} (68)

(Ishii, 1977)
Table 2 Void fraction and slip ratio correlations

<table>
<thead>
<tr>
<th>Flow Pattern</th>
<th>Reference</th>
<th>Void Fraction or Slip Ratio Correlations</th>
</tr>
</thead>
</table>
| Bubbly       | (Liao et al., 1985) | $C_o = 1$  
\[ v_{gj} = 1.53(1 - \alpha)^2 \left( \frac{g\sigma \rho^2}{\rho} \right)^{1/4} \]  
(Ishii, 1977) | $C_o = 1.2 - 0.2 \sqrt{\frac{\rho_g}{\rho_l}}$  
\[ v_{gj} = \sqrt{2} \left( \frac{g\sigma \rho^2}{\rho} \right)^{1/4} (1 - \alpha) \] |
| Slug         | (Griffith and Wallis, 1961) | $s = 1.2 + 0.2 \frac{Q_g}{Q_l} + 0.35 \sqrt{gD} \frac{u_1}{u_1}$ |
| Churn        | (Liao et al., 1985) | $C_o = 1.2 - 0.2 \sqrt{\frac{\rho_g}{\rho_l}} (1 - \exp(-18\alpha))$  
\[ v_{gj} = 0.33 \left( \frac{g\sigma \rho^2}{\rho} \right)^{1/4} \]  
(Ishii, 1977) | $C_o = 1.2 - 0.2 \sqrt{\frac{\rho_g}{\rho_l}}$  
\[ v_{gj} = \sqrt{2} \left( \frac{g\sigma \rho^2}{\rho} \right)^{1/4} \] |
| Annular      | (Liao et al., 1985) | $C_o = 1 + \frac{1 + \alpha}{\alpha + 4 \sqrt{\frac{\rho_g}{\rho_l}}}$  
\[ v_{gj} = (C_o - 1) \sqrt{\frac{gd_h\Delta \rho (1 - \alpha)}{0.015 \rho_l}} \]  
(Ishii, 1977) | $C_o = 1 + \frac{1 - \alpha}{\alpha + \sqrt{[1 + 75(1 - \alpha)] \sqrt{\frac{\rho_g}{\rho_l}}}}$  
\[ v_{gj} = (C_o - 1) \left[ J + \sqrt{\frac{gd_h\Delta \rho (1 - \alpha)}{0.015 \rho_l}} \right] \] |
3.7 Algorithm Solution:

Solving the algorithm for the model is an iterative process.

The steps of the solution are as follows:

1. Geometric parameters and phase parameters are decided and set for the airlift system. These include the total pipe length ($L$), the pipe diameter ($D$), the static head ($H_s$), the submergence ratio ($Sr$), and the densities ($\rho$).

2. The flow pattern map is developed based on the equations by Taitel et al., (1980) for the geometries and the fluid properties.

3. The airflow rate matrix is initialized using a range that covers the entire range of operation.

4. An initial water flow rate is assumed for the element of the airflow rate.

5. The correct correlation in the model is used depending on the flow pattern region from developed map.

6. The right hand side and the left hand side are solved if the difference is more than 0.01. Steps 4 to 5 are repeated if the difference is less than 0.01. The value is saved. The next element in the airflow rate matrix is then considered.

7. Steps 4 to 6 are repeated until all elements in operational range of the air flow rate are complete.
Figure 3-1 Solution algorithm logic expanded
Figure 3-2  Solution algorithm logic simplified
Chapter 4 - Experimental Setup

A laboratory experiment was conducted with high precision equipment to investigate the performance of airlift pumps and to analyze flow patterns that occur while in operation. These tests were conducted to validate the proposed theoretical prediction model for airlift pump performance with flow pattern dependency. Parallel field trials were conducted at an aquaculture facility, “McMillian Pitts,” to test large diameter airlift pumps; however, there were no methods to detect flow patterns in the field tests. These trials were performed to further validate the accuracy of the prediction method.

4.1 Laboratory Setup

An experimental setup was built to test and collect data for the airlift pump at different submergence ratios. This data will be compared to the developed model. An illustration of the experimental setup is shown in Figure 4-2. A constant head of water was provided to a supply tank (8) from the reservoir tank by a sump pump (11). In order to maintain the head, any excess water was dumped into the overflow tank (9) and retired to the reservoir. The turbulence of the water entering the supply tank was broken down by rocks to keep the head steady, simulating the static head in a large tank or well. The water then entered the bottom of the airlift pipe and filled it, achieving the same static head level ($H_s$) as in the supply tank. The injector (1) was placed at a submerged level under the supply tank (8). The injector was a dual type airlift injector patented by Badr and Ahmed, (2012). The air was supplied by an air compressor (5) and fed via a flowmeter controller (2) to control the air flow rate that was injected. The water was pumped up through a 1.5673 m length ($L$) of transparent acrylic pipe with an inner diameter ($D$) of 3.175 cm to the top of the pipe where a collection tank (4) sent the water to a measuring tank (10). The volume of water collected was measured over time in the measuring tank which had 1-liter increment markers labelled onto it. Finally, the water was then recirculated back into the reservoir tank and the system repeated itself.

A capacitance sensor (3) was connected to the airlift pipe and electrically connected to an interface, a DAQ, and a computer interface (6) where the void fraction could be measured and
studied. This sensor was developed in-house utilizing capacitance technology to measure the capacitance of the phase occupying the space. Depending on what phase (air or water) the readings indicated, the void fraction was determined. A high-speed camera (7) was aimed at a section of the pipe right above the injector to both analyze and better visualize the various aspects of the air-water interactions and the flow-patterns that occurred in the pipe due to different submergences or different injected air flow rates.

This setup could be configured in multiple ways by changing the position of the tank and the injector. This allowed for different submergence ratios to be achieved. Multiple samples of data were collected at different submergence ratios and airflow rates.

**Capacitance Sensor**

Void fraction was measured using a capacitance sensor. Two concave copper plates were wrapped on either side of the acrylic pipe and a current was sent between them. Depending on what phase (liquid or gas) was occupying the space, the amount of current transferred from one plate to another differed. This method has been used by many two-phase flow researchers and is known to be a reliable method of capturing the void fraction (Costigan and Whalley, 1997). An illustration of the capacitance sensor is shown in Figure 4-1.

![Capacitance Sensor (Dimensions in mm)](image-url)
1) Air injector  2) Flowmeter and controller  3) Capacitance sensor  4) Outlet collection tank  
5) Air compressor  6) DAQ and computer interface  7) High speed camera  8) Supply tank  
9) Overflow tank  10) Measuring tank  11) Reservoir pump

Figure 4-2 Lab setup schematic
4.2 Field Trial Setup

Real world testing of the predictive model was necessary to further validate its functionality. Trials on airlift pumps with a larger pipe diameter were performed in the field due to size constraints in the laboratory. The pipe length and submergence ratio were kept constant throughout these field trials to focus entirely on the effect of diameter on the performance of the pumps. These tests were performed at an aquaculture facility (McMillian Pitts). The available air lines at the site were used with minimal interruptions to the operation of the facility.

A survey of the site's equipment and potential space for the setup was done before the testing to ensure all necessary actions were taken. A frame was built for performance testing to accommodate all the different airlift pump diameter sizes. For performance, the 5.08 cm, 10.16 cm, 15.24 cm, and 20.32 cm airlift pumps were all tested at the same submergence ratio of 0.85 and the same total length of 2.08 m. these pipe sizes are in standard PVC sizes in inches therefore they were chosen for these tests and they would be the same sized used in industry. The purpose of the performance data collection was to compare the effect of diameter on various airlift pumps under the exact same conditions. Water was collected and weighed, and the time of collection was recorded. A minimum to maximum possible range of air flow rates were injected into the axial and radial injectors and data was collected multiple times for each test run. The limiting factors for the maximum airflow rate were the blowers and the pressure drop from the lines. A maximum rate of close to 100 kg/hr of air was possible. The blowers at the site provided the source of air for all the experiments at McMillian Pitts. There were a total of four air blowers: three smaller blowers and one large blower.

The airlift pumps were manufactured in the lab to strict dimensions. Injector hole size and spacing for the radial and axial injectors had to be set depending on the diameter size. Each airlift pump diameter size had different dimensions and a number of holes according to pre-existing calculations. The pumps were connected to open-ended PCV pipes with the same diameter as the pump to which they were connected. The most important consideration was that the total lengths of the setups were the same to achieve the same submergence ratio. The frame was constructed to accommodate all the different diameter sizes. Each diameter trial was done separately when installed on the frame. The static head depth is 1.78 m that is 0.85 submergence ratio is consistent
between all the diameters. The frame was secured with two heavy duty bolts to the floating dock. Two airlines were connected with two separate valves to control the flow rate to each injector separately. After the frame was secured to the dock, the submergence levels were checked and it was ensured that the pipe had no inclinations. A schematic of the frame and setup is shown in Figure 4-3.

Once the required airflow rate was set using the valves, the experimental run could begin. The process had to be done by at least by two people: one person collected the water and one person would note the time and write down the results. To time the run, a stopwatch was used. To collect and measure the water, a large container was held under the airlift outlet. The container was then weighed with a calibrated weight scale. For each airflow rate selected, at least three runs of the experiment were performed. Once all the possible flow ranges were complete from the frame was pulled out of the water. The range of air flow rate possible provided by the blower at the facility was 20 LPM to 1300 LPM. The next airlift pump was then installed onto the frame and the experiment was repeated. The installation and operation process of the trials are shown in Figure 4-4.
Figure 4-3 Field experimental setup (dimensions in meters)

Figure 4-4 Airlift pumps system installation and operation
4.3 Calibration and Uncertainty

For the lab experiments, two mass flow controller meters were used. Both had a 0.8% error of reading and ±0.2 % full scale with a range of 0-500LPM. Pressure measurements were done using the mass flow controllers which had a built in pressure measuring system with the same accuracy. Due to the nature of the airlift pump system the pressure at the injection point is constantly changing therefore the pressure measurements taken were inaccurate because they are taken before the injection point. For better pressure measuring results a local pressure measurement system with high capture rate should have been used. A stopwatch with 1 millisecond increments was used. The measuring tank that was used to collect the water output had 1 L increments. The following equation is the uncertainty calculated for the lab experiments: 

\[ Q_l \ (LPM) = 10\%, \ Q_a \ (LPM) = 1.2\% \]

To calibrate the capacitance sensor, the area where the sensor was located was emptied and dried entirely so that a capacitance reading could be taken for air only. A reading was then taken when the area was filled with water. The difference in capacitance should be in the order of 10 pico-farads.

Measuring the air flow rate during the field trials was done through a combination of digital flow meters and rotameters. For the lower flow rates, the digital flowmeters were used. The digital flow meters were only used in the lower ranges of the experiment for two reasons. First, the pressure drop that was caused in the lines due to the contractions was too high. Second, one of the digital meters had a cap of 100 LPM, which was designated as lowest range of the tests. The air rotameters purchased for these experiments had a higher range, which varied between 200 LPM to 3600 LPM.

The scale used in the field experiments to measure the water collected was provided by the aquaculture facility. The scale was used to weigh fish feed and had an accuracy of 0.001 kg. The scale was calibrated using a graduated cylinder. The calibration results had a of value \( R^2 = 1 \).
Chapter 5 - Results and Discussion

The data accumulated over the two sets of experiments will be discussed and compared to the prediction model developed for the validation of the method. From there, different case scenarios for airlift pump operations will be predicted from the model and the effects of geometrical parameters on the pump performance will be analyzed.

5.1 Laboratory Results

There were three sets of tests performed in the laboratory, each for a different submergence ratio: 0.5, 0.7 and 0.9. All tests showed that the highest water flow rate achieved does not occur at the highest efficiency. This has been confirmed by other researchers as well. The highest efficiency is usually in the slug flow range, and as the flow starts to transition into the churn flow range, the efficiency starts to drop but the water flow rate keeps increasing. With an increase in the air injection rate, the flow develops into churn flow. The water flow rate then starts to plateau no matter how much more air is introduced. Even if more air is added, the water flow rate will start to drop slightly. It was also noted that as the submergence ratio increases, the performance of the pumps increases as well, achieving higher flow rates and efficiencies. The relationship between the submergence ratio and performance is crucial for applications that require a low lift from the surface of the fluid. Aquaculture applications of airlift pumps mostly require a low head and a high submergence making this outcome ideal for those cases. Figure 5-1 shows the performance of Sr=0.9. Figure 5-2 and 5-3 shows the experimental data taken for all three submersences.
Figure 5-1 Experimental performance data of D=3.175 Sr=0.9

Figure 5-2 Performance results D=3.175 cm
The high-speed footage permitted a visual inspection of the flow patterns that occurred during the air injection. Figure 5-4 shows the typical flow operations observed as an airlift pump is operating in a section close to the injection point of the pipe.

Flow pattern maps were developed according to the pipe's diameter. The slug flow pattern correlations did give a good prediction when compared to the experimental data. The data was then compared to the improved model with all flow pattern correlations considered. Looking at the results, it was seen that bubbly flow does not occur in the 3.175 cm diameter pipe when considering the theoretical transition lines. Therefore, only slug, churn and annular flows were taken into consideration. When looking at the data, it was also noted that the available air flow rate in the laboratory could not achieve the required levels to reach an annular flow pattern. The slug and the churn flow patterns were the only two flow patterns that could be compared against the experimental data. The flow pattern could be inspected visually through the high-speed footage.
collected. The observed flow pattern closely matched the theoretical flow pattern map developed by Taitel et al., (1980).

Figure 5-4 sequence of slug flow pattern

Figure 5-5 Typical flow patterns observed in airlift pump
5.3 New Prediction Method Validation with Lab Results

The prediction method and the experimental data were directly compared to the model using the different correlations. Using Figures 5-6 through 5-11 the performance and the efficiency were looked at for each submergence ratio tested in the lab. From these graphs, only slug and churn flow patterns could be analyzed as there was no experimental data available for annular flow. The transition line from slug to churn is shown by the grey dotted line. For a submergence ratio of 0.5, the results are very close to what is predicted by the model. As the submergence ratio increases, the results start to stray slightly from what is predicted, as seen with a submergence ratio of 0.7. For a submergence ratio of 0.9, the results are quite far from the predicted results, meaning that either the model has difficulty predicting higher submergence ratios or that the experimental readings were more difficult to capture for the higher submergence ratio.
Figure 5-7 Performance results of 0.5 Sr compared to a drift flux model with different slip ratio correlations
Figure 5-8 Efficiency results of 0.5 Sr compared to drift flux model with different slip ratio correlations
Figure 5-9 Performance results of 0.7 Sr compared to drift flux model with different slip ratio correlations.
Figure 5-10 Efficiency results of 0.7 Sr compared to drift flux model with different slip ratio correlations
Figure 5-11 Performance results of 0.9 Sr compared to drift flux model with different slip ratio correlations
The validation of the model was done by comparing the predicted water flow rate to the measured flow rate from the experiments. The correlations used in the predictive model for the slug flow pattern and the churn flow pattern were compared. Using the graph in Figure 5-12 and calculating the root mean squared error in Table 3, the best correlations for the corresponding flow patterns were used in the model. When comparing the different correlations in the slug flow pattern, it was noted that the correlation developed by Griffith and Wallis (1961) is the best fit between the correlations. Most of the data points are within 20% of the predicted range obtained for the 0.5 and 0.7 submergence ratios. For a submergence ratio of 0.9, the data points range within 50% of those predicted. When comparing the churn flow correlations, the correlation closest to the measured results is Ishii, (1977), making it the better correlation for airlift applications. When comparing the measured data to the predicted results, the graph below represents the deviation between them.
Figure 5-13 Measured vs predicted output water mass flow rate

Table 3 Root mean square values of measured vs predicted water mass flow rate

<table>
<thead>
<tr>
<th>RMS (%)</th>
<th>RMS (%)</th>
<th>RMS (%)</th>
</tr>
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<td>±28.5%</td>
<td>±31%</td>
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</table>
The results collected in the field trials are presented below in Figures 5-13 and 5-14. Diameters of 5.08 cm, 10.16 cm, 15.24 cm, and 20.32 cm were tested. All tests were done at a 0.85 Sr with a total pipe length of 2.08 m. From the performance data it was observed that as the diameter increased, higher water flow rates and maximum efficiencies could be achieved. What was noted from the data is that there were more points in the lower air flow rate range than in the higher air flow rate range for all diameters. This could be the result of the methods used to collect the data. At lower flow rates, for example, the analog rotameter was used. As well, it was easier to collect water for longer periods of time at a lower flow rate. At higher flow rates it was more difficult to determine the water output due to the high amounts that were produced over a short period of time. Another reason is that the control value for the airlines was also more sensitive at a higher range.

The data collected for the 5.08 cm pump was only done in the lower range. This is due to a technical issue: the airline hose was smaller for the 5.08 cm pump than that for the other diameter pumps. This caused high-pressure drops in the lines when the digital mass flow meter was introduced to measure the higher range flow rates. The higher pressure caused the blower to supply insufficient air to the pump, making it impossible to measure these points. A future solution to this problem would be to have a larger airline inlet to this airlift pump.

The experimental data was close to the models for the different diameters. The chosen correlations were Griffith and Wallis (1961) for the slug flow correlation and Ishii (1977) for the churn flow correlation. The experimental results collected from the field showed that the model was able to predict airlift pump performance with a good agreement even for larger diameters. The performance of the large airlift pumps are shown from Figure 5-13 to Figure 5-18. In the 10.16 cm diameter airlift pump, the results were very promising, as there was only one outlier point, most likely due to human error. In the 20.32 cm airlift pump, the results were very close to the predicted model. The results were, however, only available in the slug flow region due to airflow limitations. In all cases the submergence ratio was set to 0.85, which is relatively high. The results appeared to be in accordance to those predicted by the model, meaning that it would be possible to predict
higher submergence ratios using the model. A list of the performance data for will be in the Appendix.

Figure 5-14 Performance of large diameter airlift pumps
Figure 5-15 Efficiency of large diameter airlift pumps

Figure 5-16 Performance of 5.08 cm diameter airlift pump compared to model
Figure 5-17 Performance of 10.16 cm diameter airlift pump compared to model

Figure 5-18 Performance of 15.24 cm diameter airlift pump compared to model
5.4 Geometrical Effects of Airlift Pump Performance

The model was validated using two sets of experiments, both in the lab and in the field. The model could now be reliably used to predict any case scenarios for airlift pump operation. A set of different cases were performed and water flow rates could be predicted depending on the airflow rate injected. In particular, the effects of the geometrical parameters $D$, $L$, and $Sr$ on airlift pump performance were of interest. The effects of these geometrical parameters could be identified with the model by changing only one of the parameters and keeping the other two constant. The following sections will deal with the results of the model.

Effect of Diameter Change

A series of model predictions were conducted for diameters of 5.08 cm, 10.16 cm, 15.24 cm, and 20.32 cm. The length and the submergence ratio remained constant at 2 m and 0.85 Sr
respectively. The different model solutions are shown in Figures 5-19 and 5-20. With an increasing diameter, higher water flow rates and lower maximum efficiencies were achieved. It was also noted that as the diameter increased, water starts to flow at a higher air flow rates. As the diameter increased, the point at which slug flow transitions to churn flow also occurred later. The efficiency of the airlift pump was also checked against changes in diameter and it was noted that the efficiency decreased slightly as the diameter increased. The efficiency peaked earlier when the diameter was smaller. The maximum efficiency in all cases always occurred in the slug flow pattern.

Figure 5-20 Airlift pump performance prediction model for different D
Effect of Length

The length of the pipe has an effect on the distance that the flow mixture travels before exiting the pipe. The longer the length, the deeper the air injection zone, which in turn would affect the water flow output. A series of runs were conducted using the model to study the effect of different pipe lengths. Lengths of 1 m, 2 m, 3 m, and 4 m were used, as shown in Figures 21 and 22. During these runs, the diameter and the submergence ratio remained constant; only the total length was changed. It was noted that as the length increased, the water flow rates produced for the same airflow rates were increased. The slug-churn transition line did not change with a change in pipe length. The equations that determine the transition point depend on the diameter, not the length. When analyzing the efficiency of the airlift pump in this case, it was noted that the highest efficiency occurred during the slug flow pattern. As the total length increased, the maximum efficiency increased as well.

Figure 5-21 Airlift pump efficiency prediction model for different D
Figure 5-22 Airlift pump performance prediction model for different $L$

Figure 5-23 Airlift pump efficiency prediction model for different $L$
Effect of Submergence

The ratio of the submerged level of pipe to the total length is defined as the submergence ratio. The performance of the airlift increases as more of the pipe is submerged. A series of models were solved for varying submergence ratios, as shown in Figures 23 and 24. The model was run for different submergence ratios while keeping the diameter and the length constant in order to analyze the effect of the submergence ratio on the pump's performance and efficiency. The submergence ratios tested were 0.25, 0.5, 0.7 and 0.9. As expected, when the submergence ratio increased the water flow rates also increased. It was also noted that more air flow was needed to start the water flow when the submergence ratio decreased. For a very low submergence ratio of 0.25, the water outflow only started after the transition line between the slug and churn flows, meaning there was only churn flow at a very low submergence ratio. The water flow rates and efficiency increased as the submergence ratio increased, which made the pumps perform better at lower submergence ratios. As in the other cases, the efficiency of the pump peaked in the slug flow region.
Figure 5-24 Airlift pump efficiency prediction model for different Sr

Figure 5-25 Airlift pump efficiency prediction model for different Sr
5.5 Integrating Airlift Pumps into an Aquaculture Facility

Now that a strong predictive model has been established for airlift pump performance, the required water flow rate can be designed for an application. One example in which the concept was tested was in aquaculture raceways, where specific water velocities must be achieved for good fish health. Baseline data was collected from the current airlift system installed at McMillian Pitts and was replicated numerically according to the amount of water flow needed. The appropriate airlift was selected, a system was installed, and new data was collected.

The facility has six larger raceways and two of the six are longer than the rest. There are also two smaller raceways that are identical. The raceways are constructed with metal framing and a 5.08 cm sheet of black plastic that wraps around the frame. The larger raceways can accommodate 3000 rainbow trout at once. These raceways are built on floating docks that are secured to the shore. The dimensions are listed in the table below with illustrations and pictures of the raceways.

<table>
<thead>
<tr>
<th>Table 4 McMillian Pitts raceway dimensions</th>
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<td>X-section area</td>
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<tr>
<td>Volume</td>
</tr>
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</table>

The velocities were measured at standard conditions for raceway operation. To measure the velocity, an ultrasonic flow tracking instrument was used. The current systems in place for the raceway operation are air-stone type airlift pumps that have a high-pressure drop and low flow rates for higher amounts of air input. The air-stone airlift pumps consist of a pipe inside a larger pipe with a large outlet at the surface of the water. Air is injected through the smaller pipe into the air-stones and the water and air are pushed to the top. This was designed and built by the aquaculture facility with very little consideration of two-phase flow properties. The velocity is measured 1 m downstream from the water flow outlet. Four points at the same distance away from the flow outlet were measured in a line crossing the raceway. Three readings were taken at each...
point and an average was recorded. The airflow rate in this measurement was 700 LPM which was the highest flow rate possible from the airline to the air-stone airlift pumps. The results of the velocity readings are in Table 5.

### Table 5 Baseline raceway velocity data

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<th>2</th>
<th>3</th>
<th>4</th>
</tr>
</thead>
<tbody>
<tr>
<td>Average Velocity (cm/s)</td>
<td>2.30</td>
<td>10.13</td>
<td>4.33</td>
<td>1.43</td>
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</table>

From this information, the amount of water flow rate required to achieve these velocities was calculated. Because the outlet area is known for the raceway, it is possible to determine the appropriate airlift to be used based on the conditions. Based on some simple calculations, four airlifts, each with a diameter of 10.16 cm, were chosen to operate in the raceways.

A setup was constructed with four 10.16 cm diameter airlift pumps connected in a parallel network. Dedicated valves were placed in the network to allow axial and radial injection for each pump, making a total of eight valves in the system. This allowed for greater control over the required air injection. The network was placed onto a metal frame. The velocity measurements were taken at 11 points along and across the raceway. The resulting data is displayed in table format and on a gradient on the raceway in Figure 5-25 and Table 6 respectively. The air flow rate that was used in the velocity readings shown was 1600 LPM, which is the highest airflow rate possible in the system. Higher air flow rates were achieved as the newer airlift pumps had less pressure drop. Data was collected for different air injection ratios for the dual injection airlift pumps: 75a/25r, 25a/75r, 50a/50r (a=axial air injection percentage; r=radial air injection percentage). Velocity data was collected using a Sontek Flowtracker that emits ultrasonic waves to measure water velocity. The system's installation and the operation of the raceway setup is shown in Figure 5-26. When comparing the newer system to the older system high air flow rates were possible because there was less pressure drop. Water flow rates were significantly improved as well. Less power can be used to operate the blowers at a lower flow rate to achieve required velocities making the new airlift system superior to the previous system.
Figure 5-26 Raceway velocity location

Table 6 Velocity reading after installing airlift pumps

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<th>4</th>
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<th>11</th>
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<td>3.16</td>
<td>8.63</td>
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</table>
Figure 5-27 Airlift pumps system installation and operation for raceway setup
Chapter 6 - Conclusion

A predictive model was developed for airlift pumps. It is sensitive to flow pattern changes and has been validated through lab experiments and field tests. The best correlations to be integrated into the model were Griffith and Wallis, (1961) for slug flow patterns, and Ishii, (1997) for churn flow patterns. The model can be applied to airlift pumps of any diameter; however, confirmation of flow pattern detection was only done to a 3.175 cm pipe due to experimental limitations. The field tests further proved that the predictive model does well even with larger diameter pumps. The predictive model was used to design an aquaculture airlift pump for a raceway system based on the velocity baseline data found in the existing system. The model can be adapted to any geometrical changes, such as changes in diameter, pipe length or submergence. This model will prove to be a powerful tool when designing any recirculating system that incorporates airlift pumps, and has been tested in the aquaculture industry. Future work could be done to further validate the flow pattern detection method in larger diameter pipes. Other future works could include the effect of air injection methods and how flow rates change depending on the method of injection. More case studies would further validate the model as well. The presented work could be used in any airlift pump system but it has only been tested for aquaculture applications. Other possible applications could be hydroponics, aquaponics, and vertical farming. More testing could be done to further validate and refine the model.

In conclusion, this study presented the following points:

- Experimental data was collected for a small diameter airlift pump, and performance and flow pattern were observed.
- Experimental data for larger airlift pumps was collected to study performance.
- A new method to predict airlift pump performance was validated by two sets of experiments.
- Geometrical parameter effects were analyzed using the prediction method. It was noted that:
  - The highest efficiencies were always achieved before reaching the maximum water flow rates.
• Water flow rates increased as the diameter of the pipe increased; the efficiencies, however, decreased.

• Water flow rates increased as the length of the pipe increased; the efficiencies also increased.

• Water flow rate increased as submergence ratio increased; the efficiencies also increased.

• Design of a raceway system was done by using the appropriate airlift pump system predicted by the model.

In conclusion, this work will be useful for the aquaculture industry, providing a reliable way to design and modify existing recirculation systems. Substituting current centrifugal or impeller pump systems with airlift pumps will be easier. Determining the correct diameter, length and submergence ratio of an airlift pump for a required flowrate will be more accurate than before using this new improved theoretical model.

For future work on improving the airlift pumps research on pulsating the air injected can should be investigated. Air pulsation can improve flow pattern control and more efficient performance could be achieved when doing so. Pulsating the air would also decrease the amount of air used to operate the system, further improving effectiveness of the airlift pump. Another aspect that would be beneficial examine would be the mass transfer capabilities of the airlift pump. Oxygen transfer would be the most important parameter to look at for aquaculture applications. Including this aspect into the model would be very important work as well.
Chapter 7 - Bibliography


https://doi.org/10.1016/0301-9322(75)90012-9


Appendix A - Laboratory experiments
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<th>Qg (m³/s)</th>
<th>Ql (m³/s)</th>
<th>Qg (kg/hr)</th>
<th>Ql (kg/hr)</th>
<th>Water Mass Flow (kg/s)</th>
<th>Air Mass Flow (kg/s)</th>
<th>Slip ratio</th>
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Table A-2: Experimental data of collected for submergence ratio of 0.7

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## Appendix B - Field Trials

Table B-1: Experimental data collected for Airlift diameter of 5.08 cm

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Table B-2: Experimental data collected for Airlift diameter of 10.16 cm (Part 1)

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Table B-2: Experimental data collected for Airlift diameter of 10.16 cm (Part 2)

Pipe Diameter = 10.16 cm
Pipe length = 2.08 m
0.85 Submergence ratio
Dual 50/50
Table B-3: Experimental data collected for Airlift diameter of 15.24 m (Part 1)

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Pipe Diameter = 15.24 cm
Pipe length = 2.08 m
0.85 Submergence ratio
Dual 50/50
Table B-3: Experimental data collected for Airlift diameter of 15.24 cm (Part 2)

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Table B-4: Experimental data collected for Airlift diameter of 20.32 cm

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</table>
The code below is a representation of the solution process.

% Airlift pump model
% Parameter setting
% Assume 25 C

clear all

g=9.81; % m/s^2 Gravity
L=2.08; % m Length
D=0.0508; % m Inner diameter
A=pi*D^2/4; % m^2 Area of inner pipe
rhol=997.05; % kg/m^3 Density of water
e=0.0015/1000; % m Pipe Roughness
v=0.0008901; % Ns/m^2 Dynamic viscosity
Sr=0.85; % Submergence ratio
Hs=Sr*L; % m Static head
Pa=101325; % Pa Atmospheric pressure
Po=rhol*g*Hs+(101325); % Pa Pressure at the inlet
rhoa=(Po)/(287*293.15); % kg/m^3 Density of air at inlet
y=0.072; % N/m Surface Tension

% Define Air flowrate matrix
Qa=linspace(0.5/(rhoa*3600),100/(rhoa*3600),10000);

% Define zero placeholder matrix for water flow rate
QLF=zeros(1, length(Qa));

% Initialize Iterative loop
j=1;
DIFF=1;
% Initial guesses for water flow rate
Ql=[.1/(rhool*3600):1/(rhool*3600):200000/(rhool*3600)];

% Define zero placeholder matrix for slip ratio and void fraction
s=zeros(1, length(Qa));
a=zeros(1, length(Qa));

for j=1:length(Qa)
i=1;
while DIFF > 0.01
    V=Ql(i)/A; % Velocity of liquid at entrance
    Re=((rhool*V*D)/v); % Reynolds
    Rz=(Re^0.98865); % Simplifier
    C=((150.39/Rz)-(152.66/Re)); % Simplifier
    f=(0.25*((log10(C))^(-2))); % friction factor
    K=((4*f*L)/D); % Simplifier
    Jtot=(Qa(j)+Ql(i))/A; % Total superficial velocity

    % Update DIFF
    % Update Ql
    % Update QL

end

end
The Distribution parameter and Drift velocity change depending on correlation used.

(Bubbly Laio et al., 1985)

\[
\text{%Distribution parameter} \\
C_0 = 1; \\
\text{%Drift velocity} \\
v_{jg} = 1.53 \times (1 - (a(j)^2)) \times ((g \times y \times (\rho_l - \rho_a)) / (\rho_l^2))^{0.25};
\]

(Bubbly Ishii, 1977)

\[
\text{%Distribution parameter} \\
C_0 = (1.2 - 0.2 \times ((\rho_a / \rho_l)^{0.5})); \\
\text{%Drift velocity} \\
v_{jg} = (2^{0.05}) \times ((g \times y \times (\rho_l - \rho_a)) / (\rho_l^2))^{0.25} \times (1 - a(j));
\]

(Churn Laio et al., 1985)

\[
\text{%Distribution parameter} \\
C_0 = (1.2 - 0.2 \times ((\rho_a / \rho_l) \times (1 - \exp(-a(j)))^0.5)); \\
\text{%Drift velocity} \\
v_{jg} = 0.33 \times ((g \times y \times (\rho_l - \rho_a)) / (\rho_l^2))^{0.25};
\]

(Churn Ishii, 1977)

\[
\text{%Distribution parameter} \\
C_0 = (1.2 - 0.2 \times ((\rho_a / \rho_l)^{0.5})); \\
\text{%Drift velocity} \\
v_{jg} = (2^{0.05}) \times ((g \times y \times (\rho_l - \rho_a)) / (\rho_l^2))^{0.25};
\]

(Annular Laio et al., 1985)

\[
\text{%Distribution parameter} \\
C_0 = 1 + ((1 - a(i)) / (a(i) + 4 \times (\rho_a / \rho_l))); \\
\text{%Drift velocity} \\
v_{jg} = (C_0 - 1) \times ((g \times y \times (\rho_l - \rho_a) \times (1 - a(i)) / (0.150 \times \rho_l))^{0.25});
\]

(Annular Ishii, 1977)

\[
\text{%Distribution parameter} \\
C_0 = 1 + ((1 - a(i)) / (a(i) + ((1 + 75 \times (1 - a(i))) \times ((a(i))^0.5) \times (\rho_a / \rho_l))^{0.5})); \\
\text{%Drift velocity} \\
v_{jg} = (C_0 - 1) \times (J_{tot}) + (2^{0.05}) \times ((g \times y \times (\rho_l - \rho_a)) / (\rho_l^2))^{0.25};
\]

Continue code

\[
B = Q_a(j) / (Q_a(j) + Q_l(i)); \quad \text{%Volumetric Quality} \\
a(j) = B / (C_0 + (v_{jg} / J_{tot})); \quad \text{%Void Fraction} \\
s(j) = ((Q_a(j) / a(j)) - Q_a(j)) / (Q_l(i)); \quad \text{%Slip ratio} \\
LHS = (H_s / L) - (1 / (1 + (Q_a(j) / (s(j) \times Q_l(i)))));
\]
RHS = \((\frac{V^2}{2gL}) \ast ((K+1)\ast((K+2)\ast(Qa(j)/Ql(i))))\);

DIFF=abs(abs(RHS)-abs(LHS));

\text{i=i+1; end}

\text{DIFF=1; QLF(j)=Ql(i); end}

\text{. . . . . .}

\text{\textbf{Special Case}}

\text{(Slug Griffith and Wallis)}

\text{%Slip ratio directly}
\begin{align*}
\text{s(j)} &= (1.2 + (0.2*(Qa(j)/Ql(i))) + (0.35*sqrt(g*D))/V)); \\
\text{x(j)} &= (Qa(j) \ast (rhoa \ast 3600)) / (Qa(j) \ast (rhoa \ast 3600) + (Ql(i) \ast (rhol \ast 3600))); \\
\text{a(j)} &= 1/((1-x(j)) / x(j)) \ast (rhoa/rhol) \ast s(j)); \\
\text{rhom(j)} &= (a(j) \ast rhoa) + ((1-a(j)) \ast rhol); \\
\text{LHS} &= (Hs/L) - (1/(1+(Qa(j)/(s(j) \ast Ql(i))))); \\
\text{RHS} &= ((V^2/(2gL)) \ast ((K+1)\ast((K+2)\ast(Qa(j)/Ql(i)))));
\end{align*}

DIFF=abs(abs(RHS)-abs(LHS));

\text{i=i+1; end}

\text{DIFF=1; QLF(j)=Ql(i);}
\text{%a(j)=(Qa(j)) / ((s(j) \ast (QLF(j)) + Qa(j))};
\text{end}

91