Hydraulic Air Compressor
Abstract

This report revolves around the hydraulic air compressor and increasing the knowledge concerning the workings of this system. A hydraulic air compressor uses hydraulic pressure generated by a descending water flow over a certain height. Air is entrained in this descending water flow in the form of bubbles and compressed due to the increasing hydraulic pressure. This system is a method of passively compressing air using a height difference. It can be a solution to situations where a constant supply of compressed air is needed and a natural height difference is present. With a hydraulic air compressor a maximum of 1 bar of pressurized air can be generated with a height difference of 10 m.

There are multiple different methods occurring in the literature of predicting the behavior and pressurization of this descending two phase flow of water and air. It was found that in the literature few experimental results are present, furthermore assumptions were based on these few experimental results. In order to determine whether these models contain unnecessary complexity or incorrect assumptions, a simpler method of determining this pressurization is suggested. These different models assumptions led to the design of an experimental set-up that would be able to determine the accuracy of these models.

The experimental set-up is designed as a closed active system, thus having the ability to control both the air and water inlet. The experimental set-up is able to measure pressure development of the descending two phase flow as well as the velocity of both the air and water inlet flow. High speed cameras film the two phase flow, giving inside in the flow dynamics.

The results generated with this experimental set-up are compared to models with different levels of complexity. This is done to determine which assumptions can be made while still accurately predict the behavior of the descending two phase flow in this experimental set-up. The friction occurring in the system is modeled with a liquid flow and assumed to be independent of the flow regime. It was also found that an approximation for the void fraction was needed and that the assumption of water and air velocity being equal is inaccurate. The modeled friction combined with the approximation for the void fraction give a relatively accurate representation of the experimental results. It was found that a changing air density had an small effect on the calculation and an approximation for the correct relation for the drag coefficient is made.
## Contents

1 List of Symbols .......................... 7

2 Introduction ............................ 9

3 Hydraulic air compressor .......................... 11
   3.1 The basic principle .......................... 11
      3.1.1 Comparison of models ......................... 15
   3.2 Presumptions ............................. 17
      3.2.1 Void fraction .............................. 17
      3.2.2 Flow regime .............................. 19
      3.2.3 Drag coefficient .......................... 20

4 Conceptual process design ......................... 23
   4.1 Downcomer pipe ........................... 24
   4.2 Separation tank ............................ 24
   4.3 Inlet of water and air .......................... 25
   4.4 Measurements .............................. 25
   4.5 Previous experimental set-up ...................... 27

5 Experimental set-up .......................... 29
   5.1 Water control ............................. 29
   5.2 Air control ............................... 30
   5.3 Pressure measurements ......................... 31
   5.4 Separation tank ............................ 32
   5.5 Bubble section ............................. 33
   5.6 Flow regime map ............................ 34

6 Experimental results .......................... 37
   6.1 Friction determination .......................... 37
   6.2 Two phase flow measurements ................. 39
      6.2.1 Calculation of two phase flow .............. 41
   6.3 HAC calculation ............................. 47
      6.3.1 Drag coefficient test ....................... 49
   6.4 Fluctuation dependent on environment ............ 50

7 Conclusion ................................ 53

8 Recommendation and Discussion ............... 55

A Appendix ................................ 57
# 1 List of Symbols

<table>
<thead>
<tr>
<th>Abbreviation</th>
<th>Description</th>
<th>Unit</th>
</tr>
</thead>
<tbody>
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2 Introduction

The earth experiences currently a lot of stress which has a negative impact on the environment and does not appear to lessen. This stress is caused by over consumption and the excessive use of fossil fuels. There are many different initiatives for decreasing this stress but there is still a long way to go. Research is focused on replacing fossil fuels with more environmental friendly energy sources, like solar panels and wind turbines. An alternative way would be to reduce the excessive use of fossil fuels and create passive systems require none. One of these systems is a hydraulic air compressor (HAC). The HAC compresses air by entraining it in a descending water flow causing the pressure in the flow to rise and thus the entrained air to compress. The HAC is location specific because a natural height difference is required for the water to descent from. This makes this system very interesting for example for the ski industry, which is where the initial question for a reevaluation of the HAC system arose. In ski areas there is a demand for snow machines, these require compressed air. Since a natural height is available in ski resorts the HAC can be a supply system for this compressed air. Another example can be large water dams that already have a natural height and often have systems that function on compressed air [1].

The HAC is a 120 year old concept. The entrainment of air in a descending water flow occurs in the downcomer pipe and the separation of the compressed air and water happens in a separation tank, see Figure 1. During the twentieth century hydraulic air compressors were used, mainly in mining plants. These HAC systems had multiple different proportions, having downcomer pipes varying between 2 and 100m [21]. Schulze cataloged these hydraulic air compressors and found that a maximum efficiency of 88% could be reached. An efficiency of 100% would be reached if the compression of air would be equal to the hydraulic pressure created by the descending water flow. There was at this time however no further analysis on this efficiency limit and an average efficiency of 70% was accepted. When targeting to use the hydraulic air compressor for modern day uses, a further analysis into the limits of this system is desired.

![Figure 1: Schematic of a hydraulic air compressor used in mines](image)

Rice described the basic mechanism of a HAC in 1979 by means of mass, momentum and energy conservation laws[16] and used experiments to determine any unknowns. This principle is used
as a base for most further research concerning a hydraulic air compressor. It contains, however, some unspecified assumptions and possibly unnecessary complexity. There are limited experimental results available in the literature. This makes it difficult to determine which assumptions are accurate and what can be simplified. Furthermore it indicated the necessity of new experimental results.

The goal of this study is to determine what assumptions can be made, while still accurately model a hydraulic air compressor. In this report a revision on the theory already available is made and to increase accuracy additional theory is proposed to model a HAC. In order to validate the theory discussed in Chapter 3 an experimental set up is designed and build, which is discussed in Chapter 4 and 5. Using this set-up experiments are performed and compared to models with different degrees of assumptions, which is done in Chapter 6. A better understanding of the hydraulic air compressor is achieved and a revised theory is validated.
3 Hydraulic air compressor

3.1 The basic principle

A HAC compresses air using hydraulic pressure generated by a descending water flow [5]. The air is entrained in the descending water flow in the form of bubbles, the increasing hydraulic pressure of the descending water flow compresses these bubbles. After the flow has descended and the air is compressed, air and water are separated in a separation chamber. The basic mechanism of a HAC is based on mass, momentum and energy conservation laws. To describe the change of pressure over the length of the downcomer pipe two models are available in the literature, one simplified model is described during this study. The first model described in the literature is the most commonly used and derived by Rice [16]. The second model found in the literature takes diffusion into account and is derived by Adriaensen [2].

In Figure 2 a schematic drawing of a HAC is given. In order to provide more clarity during the description of the HAC, the system is divided into four different stations which are visible in the figure.

![Diagram of a hydraulic air compressor divided into four stations](image-url)
3. Hydraulic air compressor

Station 0, inlet of water and air

Air and water are passively or actively injected into the downcomer pipe and enter the system at section 0 with an initial velocity and pressure. A change in velocity and pressure is likely to occur from the inlet to the beginning of the mixing of air and water, labeled as station 1. This change can be described by mass and energy conservation laws. The mass conservation law is defined as,

\[ \dot{m}_{a,0} = \rho_a A_{a,i} v_{a,1} \quad (3.1) \]

\[ \dot{m}_{w,0} = \rho_w A_{w,i} v_{w,1} \quad (3.2) \]

This conservation equation describes the conservation of mass flow rate (\(\dot{m}_0\)) from station 0 to 1 by means of the density (\(\rho\)), the area of the inlet (\(A_i\)) and the resulting velocity (\(v_1\)) at station 1.

In the following equation the change in pressure of both air and water is described by means of the energy conservation law,

\[ \frac{p_{atm}}{\rho_a} + g H_{a,i} = \frac{p_{a,1}}{\rho_a} + \frac{v_{a,1}^2}{2} + k_{a,i} \left( \frac{v_{a,1}^2}{2} \right) + p_{slip,a} \quad (3.3) \]

\[ \frac{p_{atm}}{\rho_w} + g H_{w,i} = \frac{p_{w,1}}{\rho_w} + \frac{v_{w,1}^2}{2} + k_{w,i} \left( \frac{v_{w,1}^2}{2} \right) + p_{slip,w} \quad (3.4) \]

The conservation of energy is described by the initial pressure (\(p_0\)), the gravitational acceleration (\(g\)), the height of the inlet section (\(H_i\)) and the loss coefficient of the inlet section (\(k_i\)). The loss coefficient can be different for water and air depending on the geometry of the inlet. Static friction (\(p_{slip}\)) loss occurring due to slip (\(p_{slip}\)) is taken into account as well.

Station 1, mixing of air and water

After the inlet air and water are mixed at station 1 they form a steady bubbly two phase flow at station 2. It is assumed that this two phase flow is almost instantly steady; the length between section 1 and 2 is assumed negligible compared to the length of the downcomer pipe. In order to describe the change in velocity and pressure from station 1 to station 2 the following equations for mass and momentum conservation are used,

\[ \dot{m}_a = \frac{p_2}{RT} (A_{dc} - A_{w,2})(v_{w,2} - v_{r,2}) \quad (3.5) \]

\[ A_{dc}(p_1 - p_2) = \dot{m}_a(v_{w,2} - v_{r,2}) + \dot{m}_w v_{w,2} - \dot{m}_w v_{w,1} - \dot{m}_a v_{a,1} \quad (3.6) \]

To solve both equations it is assumed that the air and water inlet area combined is equal to the downcomer area, \(A_{w,i} + A_{a,i} = A_{dc}\). In these equations the water flow cross-section at station 2 (\(A_{w,2}\)), the velocity of water at station 2 (\(v_{w,2}\)) and the initial relative velocity between the air and water at station 2 (\(v_{r,2}\)) are used. The relative velocity is seen as the difference between water and air velocity, \(v_r = v_w - v_a\). Both gravity and friction are not taken into account in the momentum conservation equation because the length between station 1 and 2 is neglected.

Station 2, top of the downcomer pipe

There are three different models describing the change in pressure over the length of the downcomer pipe. All three will be discussed in this section, the first one is derived during this study and the last two originate from the literature.
Current model;
When describing the change in pressure over the height of the downcomer pipe the first thing that
should be taken into account is the hydrostatic pressure rise which is described by,

\[ p_3 = p_2 + \rho_{flow} g H_{dc} \]  (3.7)

In order to determine the hydrostatic pressure difference the density of the flow \( \rho_{flow} \) is needed,
this density can be calculated by means of the void fraction \( \alpha \). The void fraction is the fraction
of the channel volume that is occupied by the gas phase and can be calculated by means of the
slip ratio \( K \). The slip ratio is the ratio between the gas and liquid velocity of the two phase flow.
A first assumption is made for simplicity reasons that the gas and liquid velocity are assumed
equal, resulting in a slip ratio of 1. This means that the void fraction becomes equal to the volume
fraction and the density of the flow can be calculated with,

\[ \rho_{flow} = (1 - \alpha) \rho_w + \alpha \rho_a \]  (3.8)

When considering velocity dependent friction from water in the pipe, provided by the friction
factor \( f \), the equation becomes,

\[ p_3 = p_2 + ((1 - \alpha) \rho_w + \alpha \rho_a) g H_{dc} - \frac{\nu_w^2 H_{dc}}{2 D_{dc}} f \rho_w \]  (3.9)

The friction factor for turbulent flow can by approximated by Colebrook et al [12],

\[ \frac{1}{\sqrt{f}} = -1.8 \log \left( \frac{6.9}{Re} + \left( \frac{\epsilon}{3.7 D_{dc}} \right)^{1.11} \right) \]  (3.10)

and uses the Reynolds number \( Re \), diameter of the downcomer pipe \( D_{dc} \) and the roughness of
the pipe \( \epsilon \). The Reynolds number is defined as,

\[ Re = \frac{\nu_w D_{dc} \rho_w}{\mu_w} \]  (3.11)

The equation above provides the Reynolds number in a HAC by means of the velocity \( \nu_w \), density
\( \rho_w \) and dynamic viscosity \( \mu_w \) of the water flow as well as the diameter of the downcomer pipe
\( D_{dc} \). The Reynolds number is calculated for the water flow in the downcomer pipe as it assumed
that the water flow is the main component of the velocity dependent friction occurring in the
downcomer pipe. The friction factor when the flow is laminar can be described with,

\[ f = \frac{64}{Re} \]  (3.12)

It is dependent on the Reynolds number which friction factor gives the most accurate approxima-
tion of the velocity dependent friction loss in the downcomer pipe.

Rice;
Model two is derived by Rice and takes velocity and temperature change into account [16]. When
the pressure, velocity and temperature of a two-phase flow are modeled the Bernoulli relations for
mass and momentum are used. To describe the development of pressure, velocity and temperature
over the length of the downcomer a discretization technique is used. The downcomer pipe is divided
in small sections and the velocity and pressure change is computed over each section. This system
has three unknowns; pressure, air velocity, and water velocity \( p, v_a, v_w \). In order to model these
three variables three relations between pressure and velocity are needed. The first is the mass
conservation equation,

\[ d\dot{m}_a = d \left( \frac{p}{RT} (A_{dc} - A_w)(v_w - v_r) \right) = 0 \]  (3.13)
This describes mass conservation neglecting diffusion over the length of the downcomer pipe. Second, the conservation of momentum is balanced by gravity and friction provided by,

\[
\dot{m}_w(v_w + dv_w) - \dot{m}_w v_w + \dot{m}_a ((v_w - v_r) + d(v_w - v_r)) - \dot{m}_a (v_w - v_r) = dp + \rho_w A_w gdz + \rho_a (A_{dc} - A_w) gdz - \frac{f}{2} \pi \rho_w D_{dc} \frac{dz}{D_{dc}}
\] (3.14)

In these equations the water velocity \((v_w)\), the air velocity \((v_a)\), the relative velocity \((v_r)\) and the pressure \((p)\) are considered to be variable. The variables preceded by a ‘d’ described the change of that variable over one section with the length \(dz\). The variables not preceded by a ‘d’ are the values of that variable at the beginning of the section. The derivation of these used equations can be found in the Appendix. In Figure 3 a visual representation of the discretization technique is shown.

![Figure 3: Visual representation of the discretization technique used](image)

The third and final relation in this model between pressure and velocity is based on the assumption made by Rice that the drag force on an air bubble is balanced by the buoyancy and gravity force. Where the drag, gravity and buoyancy is provided by,

\[
F_{drag} = \frac{1}{2} C_D \rho_w v_r^2 \frac{\pi}{4} D_b^2
\] (3.15)

\[
F_{buoy} = \frac{\pi}{6} D_b^3 (\rho_w - \rho_a) g
\] (3.16)

In these equations the diameter of the bubble \((D_b)\) and the drag coefficient \((C_D)\) are used. Bubble velocity over the length of the downcomer pipe can be described by,
It is assumed that no coalescence or break-up of bubbles occurs and all bubbles have the same shape and size. The equilibrium assumed between these forces, together with the ideal gas law, can be used to model the change in relative velocity dependent on the change in pressure. The equilibrium between the buoyancy, gravity and drag force also results in a relation for the bubble diameter,

$$D_b = \frac{3v_r^2 C_D}{4g}$$

In these relations a constant drag coefficient $C_D$ is assumed. Because the change in Reynolds number over the length of the downcomer pipe is negligible. This relation results in a dependence of the bubble diameter on the relative velocity and visa versa. For one of these an initial value is required. Rice assumes an initial value for the relative velocity of 0.244 m/s, assumed to be independent of HAC design and two phase flow properties.

Adriaensen; Model three that can be used to describe the pressure development in the downcomer pipe is a model described by Adriaensen [2]. The difference between this and the previous model is that this model takes diffusion of air into water into account. The diffusion calculation uses the diffusion constant ($F_j$) and an initial concentration ($LMCD_j$) for the four main components of air (nitrogen, oxygen, hydrogen, argon), as well as a specified average residence time ($t_{avg}$) and the bubble surface ($A_b$).

$$\Delta \dot{n}_j = 2 \sqrt{\frac{F_j}{\pi t_{avg}}} LMCD_j A_b$$

The model uses the same relation for wall friction as the model described by Rice, and the same assumption for the relative velocity and bubble diameter relation. This model contains the same discretization technique as the model from Rice which makes it possible to calculate the development of pressure over the length of the downcomer pipe.

Station 3, inlet separation tank
During the separation of air and water in the separation tank a change in pressure occurs depending on the depth of the downcomer pipe in the separation tank. This change of pressure is described by the equation below and is solely dependent on depth of the downcomer pipe in the separation tank ($h_{34}$), labeled as the height difference between station 3 and 4.

$$p_{sep} = p_{dc,o} - \rho_w g h_{34}$$

### 3.1.1 Comparison of models

The models concerning the downcomer pipe described above are implemented in Matlab. In order to determine whether this is done correctly a validation calculation is performed. Of course the pressure at the bottom of the downcomer should be equal to the atmospheric pressure when the length of the downcomer pipe is zero. For further validation and comparison the models are simulated for increasing length of the downcomer pipe. Both water and air mass flow rate are kept constant at 1000 kg/s and 1 kg/s respectively. The model from Rice and Adriaensen both assume a constant initial relative velocity determined by Rice, the model from Rice assumes for this comparison a constant temperature and the model from Adriaensen takes temperature change into account. The current model initially assumes a slip ratio of 1, resulting in the void fraction.
being equal to the volume fraction. All three models assume turbulent flow and use the same approximation for friction loss given by Equation 3.10. In Figure 4 a comparison of these models is shown.

![Figure 4: Comparison of three models for pressure development over increasing length of the downcomer pipe.](image)

Visible from Figure 4 is that the current model starts at atmospheric pressure and shows a linear increase in bottom pressure with increasing downcomer height. This behavior is expected as the calculation for bottom pressure has a linear relation to the downcomer height, see Equation 3.9. This model has compared to the Rice model two differences. The first is the neglected pressure change due to velocity decrease over the height of the downcomer pipe. The second is the difference in considering compression, the current model uses the assumption that the void fraction is equal to the volume fraction while the model by Rice uses an initial relative velocity which changes with increasing pressure.

The model by Rice computes a decrease in relative velocity \(v_r\) due to an increase in pressure over the length of the downcomer pipe, see Equation 3.17. The decrease in relative velocity is linked to a decrease in bubble diameter \(D_b\), see Equation 3.18. Both water \(v_w\) and air velocity \(v_a\) decrease over the length of the downcomer pipe. This decrease in velocities results in a pressure loss that increases with increasing downcomer height, resulting in a non linear relation between the bottom pressure and the downcomer height. This causes a difference in pressure loss between the current model and the model by Rice and should result in a lower bottom pressure for the Rice model than the current model. This is not the case when looking at Figure 4. This can be caused by the difference in assumption concerning compression between these two models. The current model assumes a relative velocity of zero, while the model of Rice assumes an initial relative velocity of 0.244 which changes with increasing pressure. This results in a different approximation for void fraction and the resulting density of the two phase flow. It appeared difficult to compare these different methods of compression which the following example shows.
When taking a downcomer height of 100m, the difference between bottom pressure calculation of the two models is 3.3 bar. When calculating the difference in friction due to decreasing water velocity this is 0.73 bar. The void fraction calculated by the hydrostatic pressure model is 0.43 which is different from the void fraction indirectly determined by the Rice model. When calculating the void fraction for this model a void fraction of 0.45 is found at the top of the downcomer and a void fraction of 0.46 is found at the bottom. There is a small difference between to the void fraction used in the current model and the void fraction derived from the Rice model which does not explain the difference visible in Figure [3].

It is likely that the different methods considering compression handled by these two models causes this difference. The model developed by Rice shows no clear relation between the relative velocity and the void fraction, making it unclear how these two are related in this model. The Rice model does make use of an initial volume fraction, see Equation 3.14 which changes due to changing relative velocity. The exact relation is unfortunately currently still unclear. There is also the possibility that the difference is cause by a fault in the Matlab script used to implement these models. There are, however, test cases performed and these show results that suggest that this is not the case.

The model by Adriaensen appears to not have a linear relation between the bottom pressure and downcomer height. When comparing this model to the model made by Rice it is visible that the difference between the models increases with increasing downcomer height. That is explained by the diffusion that the Adriaensen model takes into account. The effect of diffusion on the pressure drop over the downcomer height will increase with increasing downcomer height due to the time of air being entrapped in water increasing. Because the model by Adriaensen uses the same assumption for the relative velocity as the model by Rice the same difference between the current model and this model is visible.

### 3.2 Presumptions

In the previous paragraph a description of pressure development over the entire HAC system is given. There are three different models given that simulate pressure, and additionally velocity and temperature development over the length of the downcomer pipe. These three models contain assumptions. I propose that these can be avoided by considering mechanism that are likely to take place in the HAC such as the void fraction, flow regime and drag coefficient.

#### 3.2.1 Void fraction

The void fraction is defined as the volume fraction of the gas phase of the flow. The definition of the void fraction ($\alpha$) is provided in the following equations [11],

$$x \equiv \frac{\dot{m}_a}{\dot{m}_w} \quad (3.21)$$

$$\frac{1}{\alpha} \equiv 1 + \frac{(1 - x)}{x} \frac{\rho_a}{\rho_w} K \quad (3.22)$$

As visible from the equations, the void fraction is defined by the quality of the flow ($x$), the density of both the water and air phase ($\rho$) and the slip ratio ($K$). The slip ratio is defined as the gas velocity over the liquid velocity. When assumed as 1, the void fraction equals the volume fraction. It is however likely that water and air velocity are not equal, which is why different correlations for the slip ratio are introduced.

The first correlation for the slip ratio is based on the relative velocity that is determined by Rice et al. The relative velocity is considered to be the difference between the velocity of the gas phase and the liquid phase in a two phase flow, in this case the difference between the velocities of air and water. Due to the balance assumed between the buoyancy, gravity and drag force the initial relative velocity is only dependent on the bubble diameter. The relative velocity considered by
Rice is determined using an experimental model with a downcomer pipe of 2m and a diameter of 20mm. Considering this set-up and performing experiments with variable height difference an initial relative velocity of 0.244 m/s is determined by Rice. These results agree with experiments from Martin [15] who did experiments on bubble rise velocities in pipes with similar diameters. It is also stated that the relative velocity is solely dependent on the change of pressure. The slip ratio when considering the relative velocity determined by Rice is as follows,

\[ K = 1 - \frac{v_r}{v_w} \]  

(3.23)

This relation derived by Rice is thus dependent on the water velocity as well as the relative velocity which has an initial value of 0.244m/s and changes with changing pressure according to Equation 3.17.

The most commonly used correlation for the slip ratio is an empirical relation from Lockhart and Marinelli [11], provided by,

\[ K = 0.28 \left( \frac{x}{1 - x} \right)^{0.36} \left( \frac{\rho_w}{\rho_a} \right)^{0.64} \left( \frac{\mu_w}{\mu_a} \right)^{0.07} \]  

(3.24)

Another correlation which is widely used and valid for both vertical and horizontal flow below atmospheric pressure until a void fraction of 0.72 is the correlation defined by Armand [11], provided by,

\[ \alpha = C_A \alpha_H \]  

(3.25)

This correlation is defined by the homogeneous void fraction (\( \alpha_H \)) which is the void fraction for a slip ratio of 1, and the Armand coefficient (\( C_A = 0.83 \)). A comparison of both relations for the slip ratio is visible in the next figure.

As visible from Figure 5 the Lockhart and Martinelli correlation differs from the homogeneous void fraction at low flow quality, but this difference decreases with increasing flow quality. The Armand correlation has the same shape as the homogeneous void fraction but with a constant difference. From experiments it can be concluded which of these correlations is the most accurate.
for the experimental set-up. It should also be noted that these correlations have been determined for an upward flow and in the hydraulic air compressor downward flow occurs. This results, when assuming the downward direction to be positive, in an air velocity which is lower than the water velocity and thus a slip ratio higher than 1. In Chapter 6 the assumption that $\frac{1}{K}$ is a correct approximation for the slip ratio in downward flow is tested.

3.2.2 Flow regime

In the literature concerning a HAC the assumption is made that the flow regime in the downcomer pipe is always a bubbly flow. Meaning that the air entrained in the water only takes the shapes of bubbles. There are, however, multiple different flow regimes besides the bubbly regime. When taking a closer look at when these different flow regimes occur it is clear that there is a large possibility that the two phase flow occurring in the downcomer pipe of a HAC does not fall under the bubbly regime.

The regime where two phase flow is categorized in is firstly dependent on the size of the HAC, secondly on the void fraction and on the total mass flux as well. In order to make predictions concerning two phase flow a categorization of flow regimes is introduced. The aim is that two phase flow in a HAC can be categorized as one of these regimes or as a transition between regimes. The main flow regimes occurring are: bubbly, intermittent, dispersed and Annular, see Figure 6.

The bubbly regime is characterized by multiple air bubbles traveling through the water flow, the shape of these bubbles depends heavily on the velocity of the flow [21]. In vertical downward flow the bubbles tend to have a spherical or ellipsoidal shape at high flow rate and a more hemispherical shape at lower flow rate. At intermittent flow a combination of bubbles and slugs of air can be found. Slugs of air are defined as large pockets of air that have almost the same size as the tube diameter in question, these slugs of air will appear sporadically and their size and rate of occurrence is difficult to predict. The intermittent phase can transition into annular flow which has a consistent water film around the outer surface and a fully gaseous core. Dispersed flow is characterized by water layers that trickle down the walls and a fully gaseous core.

It appears to be difficult to predict the flow regime even when knowing the mass flux and the void fraction of the flow, which is why multiple experiments are done to create a flow regime map. Crawford, Weinberger et al. [9][8] have created multiple charts of this kind, dependent on the tube diameter, mass flux, void fraction and pressure of the two phase flow.

In Figure 7 an indication can be found for when different flow regimes might occur. It can be seen from this figure that there is a small region of void fraction and mass flux at which a two phase flow is categorized as a bubble flow. The region for intermittent, dispersed and annular are all significantly larger. The void fraction in a HAC is low [21] and the annular and dispersed regime

![Figure 6: Different flow regimes for downward two phase flow](image-url)
are not likely to occur in a HAC. The intermittent regime has a larger possibility of occurring in a HAC, especially in larger HAC systems when the mass flux increases. It is thus important to understand the effect different flow regimes have on the compression of air in the HAC system.

3.2.3 Drag coefficient

Because the air bubbles in water flow can be seen as objects in a moving medium, these bubbles experience friction due to drag. The drag coefficient describes, together with the velocity and properties of the medium, the drag an object encounters. This drag coefficient is largely dependent on the shape of the object and the behavior of the flow. In previous models this drag coefficient is kept constant at 0.44 which is the drag coefficient for a spherical bubble.

The Reynolds number gives an indication of the amount of turbulence occurring in the flow. The Reynolds number of the liquid flow surrounding the air bubble will describe the stability of the flow most accurately because the void fraction of two phase flow in a HAC is low. If the Reynolds number changes over time the size and most likely shape of the bubbles in the two phase flow will change as well. When looking at Equation 3.11 it can be seen that the Reynolds number is dependent on the water velocity. The velocity of water decreases over the length of the downcomer pipe meaning that the Reynolds number decreases over the length of the downcomer and does not remain constant.

The Eötvös number gives an impression of the gravitational forces compared to the surface forces the bubble experiences. Therefor, the Eötvös number gives an indication of the deformation of the bubbles.

\[
E\ddot{\text{o}} = \frac{(\rho_w - \rho_a)D_{dc}g}{\sigma} \quad (3.26)
\]

As visible from this equation, the Eötvös number is calculated by means of density difference, characteristic length which in this case is the diameter of the bubble, the gravitational acceleration and the surface tension (\(\sigma\)).

Looking at experiments done by Bhagwat and Ghajar [4] it is visible that the bubbles decrease in
size over the length of a downcomer pipe. Which is due to increasing pressure over the length of the downcomer pipe.

Figure 8: Behavior of bubbles in a vertical downward two phase flow

This increase in hydraulic pressure will cause a change in surface tension and thus a change in Etvos number. Both these observations suggest that keeping the drag coefficient constant could lead to potential error in calculations considering a HAC.

There are multiple different theories concerning the drag coefficient, all of which are based on experimental results. Both theories discussed here are based upon different bubble behavior. The first is derived by Cheng et al. [6] and describes the drag coefficient of one spherical particle. This relation is based on the Reynolds number of the relevant particle, and is valid for \( \text{Re} \leq 2 \cdot 10^5 \),

\[
C_D = \frac{24}{\text{Re}} (1 + 0.27\text{Re})^{0.43} + 0.47[1 - \exp(-0.04\text{Re}^{0.38})]
\] (3.27)

To accommodate none spherical bubbles Dijkhuizen [10] derived a new relation using the Etvos number. This derivation takes the influence of both the Reynolds number and the Etvos number into account and is valid for a range of \( 10 \leq \text{Re} \leq 1500 \) and \( \text{E} \dot{\text{o}} \leq 15 \). The relation derived by Dijkhuizen is valid for a single bubble,

\[
C_{D,\infty} = \sqrt{C_{D,\text{Re}}^2 + C_{D,\text{E} \dot{\text{o}}}^2}
\]

\[
C_{D,\text{Re}} = \frac{16}{\text{Re}} \left( 1 + \frac{2}{1 + \frac{16}{\text{Re}} + \frac{3.315}{\sqrt{\text{Re}}}} \right)
\] (3.28)

\[
C_{D,\text{E} \dot{\text{o}}} = \frac{4\text{E} \dot{\text{o}}}{\text{E} \dot{\text{o}} + 9.5}
\]

The relation derived by Dijkhuizen can be altered for a bubble in a swarm. This is done by Rusche and Issa [19] by defining a correction factor \( f(\alpha) \) and taking the void fraction \( \alpha \) into account,

\[
\frac{C_{D,\text{swarm}}}{C_{D,\infty}(1 - \alpha)} = f(\alpha)
\] (3.29)

\[
f(\alpha) = \left( 1 + \frac{18}{\text{E} \dot{\text{o}} \alpha} \right)
\] (3.30)

The correction factor for bubble swarm that was found most accurate by Adriaensen is the one derived by Roghair [18] and is defined by the Etvos number and the void fraction.

A comparison of these drag coefficients is made by comparing their behavior to both void fraction and Reynolds number, see the figures below. For this comparison the Etvos number is kept constant at 1 and the void fraction and Reynolds number are kept constant in turn.

As visible from Figures 3.2.3 and 3.2.3 the three drag coefficients discussed above are compared to the constant drag coefficient for spherical particles, which is 0.44. It is visible that the drag coefficient determined by Cheng is independent of void fraction and has a non linear relation to the Reynolds number. This relation determined by Cheng approaches a limit of 0.4 at a Reynolds
number of approximately 2500 and little change in the drag coefficient is noticed with increasing Reynolds number.

The relation derived by Dijkhuizen decreases linearly with increasing void fraction and appears to have a non linear relation to the Reynolds number. This relation approaches a limit of 0.34 at a relatively low Reynolds number.

The relation for the drag coefficient considering a swarm of bubbles is similar to the relations by Cheng and Dijkhuizen and has a non linear relation to the Reynolds number. This relation approaches a high limit of 0.86 compared to the other relations at a relatively low Reynolds number. This relation for the drag coefficient has a very different relation to the void fraction compared to the others. It increases with increasing void fraction until a void fraction between 0.3 and 0.4 is reached and then starts to decrease with increasing void fraction. This means that the friction a bubble experiences for these mass flow rates of air and water has a maximum at a void fraction between 0.3 and 0.4 and then decreases with further increasing void fraction. This maximum will be different for each situation dependent on the void fraction and the Etvos number. This can be caused by bubbles starting to cluster and thus drag starting to decrease at a certain void fraction.

Figure 9: Comparison of drag coefficients against Reynolds number, with constant void fraction of 0.1, and comparison of drag coefficients against void fraction, with constant Reynolds number of 5000.
4 Conceptual process design

In order to validate the theory discussed in the previous chapter experimental results are required. In order to obtain these an experimental set-up is designed and build. Previous experimental set-ups of hydraulic air compressors were passive and open systems [10] [5]. In the figure below a schematic representation of a passive open system is visible. Passive HAC systems make use of the Bernoulli effect which causes natural suction of air and water at the top of the downcomer pipe and a natural drainage at the top of the raiser pipe. Due to the additional raiser pipe the hydraulic head is no longer equal to the height of the downcomer pipe but to the difference between the downcomer and riser pipe. The hydraulic pressure that can be reached in these systems is thus equal to the hydraulic head and not the height of the downcomer pipe.

These systems have the downside that the inlet conditions can not be changed without altering the hydraulic head. Needing to change the geometry of the set-up when different flows are tested is from a time management point of view impractical. Not to mention that each time the set-up is altered risks for errors occur. The second downside of this system is that the mass flux of the flow is inevitably linked to the hydraulic head, due to the Bernoulli effect. These downsides are why for this experimental set-up a closed active system is chosen. This way both air and water inlet condition can be controlled separately from the hydraulic pressure. Another advantage of a closed system is that no water supply and drainage is necessary.

The most important element of the hydraulic air compressor is the downcomer pipe, which is where the compression of air takes place. It is important that the downcomer pipe is designed to provide experimental results that can validate the theory discussed in Chapter 3. Because a closed system is chosen it is important that at the end of the downcomer pipe air and water are separated from each other. This will be done in a separation tank. In the figure below a first schematic representation of the experimental set-up is shown.
4. Conceptual process design

4.1 Downcomer pipe
The downcomer pipe is a significant part of a HAC and thus needs to be design carefully. In order to measure a pressure difference the pipe requires a significant length. If the pipe is too short pressure sensors might not pick up on the pressure difference and if the pipe is too long it would become impractical from a spacial point of view. Because a two phase flow requires time to develop and stabilize a specific length in the pipe is needed for it to reach a stable state. If measurements take place when the flow is still unstable these measurements might not form an accurate representation of a HAC. According to experiments done by Crawford and Weisberge \[9\] a ratio of 100 between the height and diameter is sufficient to create a stable flow in time. If the size of the downcomer pipe is comparable with experimental results known in the literature, comparison between experimental results is possible. Resulting in a downcomer pipe with a length of 2m and a diameter of 2cm. In a worst case scenario the flow will stabilize at the bottom of the downcomer pipe, making measurements over most of the length of the downcomer pipe inaccurate. However, previous experiments suggest that it is most likely not the case \[21\]. The height of 2m will create a maximum pressure difference of 0.2 bar which is sufficient enough to measure with pressure sensors.

4.2 Separation tank
After the two phase flow has passed the downcomer pipe it enters a separation tank. The separation tank should be large enough to let the two phase flow fully separate. This means that there needs to be enough room in the tank for the two phase flow to become stationary and for the air bubbles to travel to the surface. The air bubbles that require to drift to the surface are doing so with a speed dependent their size. Assuming a spherical bubble with a diameter of 1mm and a balance between the drag and buoyancy force this bubble will have a drift velocity of 0.17m/s \[7\]. Larger bubbles have a larger drift velocity due to their larger size. The height to width ratio of the tank
for bubbles to rise to the surface needs to be at least the drift velocity over the pump velocity,

\[
\frac{\text{height}}{\text{width}} \geq \frac{v_{\text{drift}}}{v_{\text{pump}}}
\]  

(4.1)

The space and time it takes for the two phase flow to become stationary before air bubbles travel to the surface is hard to predict and differs with altering mass flow rates during experiments. The tank should thus be large enough to accommodate the maximum mass flow rate.

### 4.3 Inlet of water and air

In order to simulate multiple different two phase flows in a HAC an active water and air inlet system is chosen. The water inlet will be controlled by a pump that transports water from the separation tank up to the inlet of the downcomer pipe. The water flow rate needs to be variable, which is easiest to do with a valve right behind the pump. This makes sure a range of water flow rates can be reached.

The air inlet takes place in the form of injection in the water tube before the downcomer pipe. The injection of air is done earlier in the water tube. The advantage of this is that the flow has a bit more time to stabilize. This might be needed because the injected air is pressurized and has a larger initial velocity and a different pressure than the water and will need time to stabilize. Extracting air from the separation tank is necessary, otherwise the pressure in the separation tank starts rising and influences the entire system. The outlet for the air in the separation tank is controllable because then a stable situation can be created with as much air entering as leaving the system.

There are multiple different ways of injecting air into a water mixture with the goal of creating a two phase bubbly flow. The two main options are injection at the edge or in the middle of the flow. Injecting air in the middle of the water flow has the advantage of air bubbles being less likely to cling to the wall. However, placing an injection system in the center of the flow causes disturbances in the flow which influence the mass flux. This is complex to describe and needs to be measured for validation. The injection at the side of the flow can be done by using a T-piece and causes less disturbances to the flow but can cause bubbles to cling to the walls. For this experimental set-up injection by means of a T-piece is chosen as it causes less unknown disturbances and is easy to install. If it is found to be difficult to create bubbles using this method, a straightener can be used in the future to ensure a bubbly flow.

### 4.4 Measurements

The measurements that are done with the set-up cater to the theory discussed in Chapter 3. Theory for pressure development, flow regime and drag coefficient are tested with this experimental set-up.

**Pressure**

In order to validate the theory for pressure development in the downcomer pipe the pressure at the top and bottom of the downcomer pipe are measured. In order to determine whether the separation tank works properly a pressure measurement in the tank is necessary as well. This is to monitor the increase in pressurized air and to control the air outlet. In order to determine the effect of the flow regime as well as bubble size and behavior flow measurements are done as well.

**Bubbles**

A way of measuring two phase flow as well as bubble size and behavior is to film the flow in the downcomer pipe from up close. This gives the possibility to examine the development of two phase flow more precisely as well as the shape of bubbles at various places in the downcomer pipe. There is software available that measures the size of bubbles from pictures taken with high
speed cameras. It is important to film the flow at the top and bottom of the downcomer pipe to determine development of the flow over the length of the pipe. To be able to make 2D video measurements the refraction of light from the round downcomer pipe is counteracted, this is done with a square shaft around the round pipe filled with water, visible in Figure 12. The square shaft is made of the same material as the round pipe and has the same thickness. The square shaft is not placed over the entire length of the round pipe because measurements are taken at the middle and bottom section of the pipe where stable two phase flow occurs.

Figure 12: An exaggerated example of the counteraction of the refraction of light seen from the top view of the downcomer pipe

The technique used for bubble size and distribution measurement is developed by Lau [14]. This technique makes use of 2D images which are processed and analyzed by a program written in Matlab. This program is able to analyze both size and distribution of bubbles. It does this by first filtering the taken image so it can distinguish single bubbles from clusters of bubbles. The distinction between cluster and singular bubbles is made based on the roundness of the shape and is displayed differently in the results. As visible from Figure 13 both singular bubbles as well as clusters are detected and marked with either blue or red. For each of the markings the area and radius is calculated and thus the bubble size and distribution is known. The errors in size will be +2% to -6% for small bubbles (less then 30 pixels) and +0.5% to -1% for larger bubbles (more then 30 pixels). The volume of the bubbles can be calculated as well with an error of +5% and -16%.

Figure 13: Example of a result produced by the described bubble measurement technique, adopted from Lau [14]
4.5 Previous experimental set-up

First, before discussing the actual set-up, an experimental set-up is discussed which was previously used with the same goal as the set-up currently created. This previous set-up was proven to be insufficient and resulted in the suggestion for a new experimental set-up.

A previous experimental set-up is used by Adriaensen [2] to perform an experimental analysis on the hydraulic air compressor. This previous set-up is a modification of an already existing set-up which was used for bubble generation and analysis of bubble migration and distribution in vertical channels. See Figure 14 for a schematic representation of the original set-up.

Figure 14: Schematic representation of the original experimental set-up used for bubble generation and analysis

With this original set-up the inlet conditions of water and air can be controlled and different straighteners can be used to create bubbles. This made the potential for the set-up to replicate a downcomer pipe in a HAC system optimistic. The height of the original set-up was 5.5m, which was favorable as Adriaensen had the aim to simulate a hydraulic air compressor with a height of approximately 200m. The main element this original set-up was missing was a separation tank, which separates the air bubbles from the water flow. This element is essential because the main objective of the hydraulic air compressor is to generate air under pressure and the separation tank separates the air bubbles in which pressure is generated from the liquid.

The separation tank was added in the form of an old beer canister of 50L which is linked to the experimental pipe via a tube of approximately the same diameter and used to separate the gas from the liquid, see Figure 15. The air flow then exiting the separator was measured using an analog flow-meter. The reason not a more accurate flow-meter was chosen was due to the possibility of the tank overflowing and water exiting via the air outlet and damaging the flow-meter.

The experiments done by Adriaensen with this experimental set-up were found to have a large inaccuracy, mainly due to the measuring equipment. There were also other factors that had a negative effect on the measurements.

This experimental set-up appears to have some significant downsides, the first being unable to
measure the pressure of the air exiting the separation tank. It is impossible to add a pressure meter to the current set-up due to the high possibility of water exiting the air outlet. This leads to the next downside of the set-up, the possibility of the separation tank overflowing. This is to the tank being too small to handle the large mass flow rates, which are hard to avoid due to the size of the downcomer pipe. Besides the downsides of the separation tank, the inlet of the downcomer pipe has some downsides as well. The first is the air inlet. Air injection is done with a mass flow controller that has been there for over 20 years and there is a large suspicion that this mass flow controller is no longer accurate. It can be solved by disconnecting the mass flow controller and to have it calibrated again in order to improve its accuracy. Another downside is that it is not possible to create different types of flow regimes using the current set-up due to the different straighteners that are used to create bubbles. The last and most significant downside of the set-up is that it makes it impossible to measure the bubbles using 2D video techniques. This is due to the thick cylindrical pipe the flow travels through, which causes large deflection of the actual location of the bubble. Due to the size of the current set-up it is expensive to alter the entire tube section in order to take 2D video measurements. This will also result in the set-up no longer being able to be used for its original purpose.

These downsides lead to a suggestion for a new experimental set-up. It was important for the new set-up to keep as much variables as possible in order to have a wide range of measurement possibilities.
5 Experimental set-up

An experimental set-up is designed, detailed and constructed based on the conceptual design from Chapter 4. The experimental set-up is used for data acquisition concerning the workings of a hydraulic air compressor. This experimental set-up is a first version and is designed while assuming a possibility for alterations and is thus easily adaptable. In the figure below a schematic drawing of the final set-up is visible and the choices made in each element are explained below. For further specifications of the components, see the Appendix.

![Schematic drawing of the experimental set-up](image)

Figure 16: Schematic drawing of the experimental set-up

The entire set-up is placed inside a frame for support and convenience, which is provided by Hepco Motion. When designing and building the set-up it is taken into account that the possibility for alterations to the set-up is large, which is why the frame is easily adaptable. One of the constraints is that the set-up should be easily transported, resulting in a mobile frame having dimensions that fit through doors and in elevators. The weals that are used can be locked in both degrees of freedom making sure that the set-up is stable when experiments are running.

5.1 Water control

The transportation of the water flow from the separation tank to the inlet of the downcomer pipe is done by a pump, visible in Figure 16 and Figure 5.1. The pump operates at a maximum flow of approximately 6.8 L/min, the pump curve for this pump can be found in the Appendix. In order to control the water flow a needle valve as well as a flow sensor is used. Both the flow sensor (symbol V in Figure 16) and the needle valve are visible in Figure 16. The needle valve can control the flow with an accuracy of approximately 0.5 L/min and the flow sensor measures with
an accuracy of 2%. The data acquisition is done with SignalExpress which loads measurements from the flow sensor every second.

5.2 Air control

The air inlet and outlet are controlled by mass flow controllers, visible as symbol M in Figure 16. By controlling the inlet multiple flow regimes can be reproduced and by controlling the outlet a stable state can be reached in the system.

The mass flow controllers that are used are from Bronkhorst and have a range from 0 L/min to 18 L/min. They can be regulated with an accuracy of 1% and are controlled by the program LabView. These mass flow controllers need pressurized air of at least 2 bar and thus a pipeline for pressurized air is installed. The mass flow controller responsible for the inlet is connected to the pressurize air pipeline and will inject air with a pressure of 1 bar. Because the pressure in the separation tank can not be increased to 2 bar, partially due to safety measures as well as leakage from the tank, it is not possible to precisely control the outlet mass flow controller. In practice this mass flow controller can be kept open or closed.

The method of injection in the water tube is by means of a T-piece, see Figure 19. The injection is done before the beginning of the downcomer pipe to give the air and water more time to stabilize before they enter the downcomer pipe.

For safety measures a clamping tool is used when the set-up is not turned on to prevent water from entering the air tube. When the mass flow controllers were to come in contact with water it could lead to damage in the mass flow controllers.
5.3 Pressure measurements

The pressure is measured at three different locations, each location visible as symbol P in Figure 16. In order to analyze the progression of the two-phase flow in the downcomer pipe, the pressure of the flow at the top and bottom of the pipe needs to be known. The choice is made to place the pressure sensor not completely at the top and bottom of the pipe but at the edges of the chosen measurement section, visible as the square shaft around the downcomer pipe in Figure 16. This is done to make sure measurements are taken at locations where the flow is stabilized. The height difference between the top and bottom pressure sensors is 155.89 cm. The pressure sensors that are used are from Keller and have a range of 2 bar and an accuracy of 0.08%. These sensors measure relative to the atmospheric pressure.

The pressure in the separation tank is measured at the top, making sure to measure the pressure of the air separated from the water. It analyzes the effect of pressure change when the set-up runs for a longer period. The pressure sensor used at this location is also from Keller and has a range of 5 bar with an accuracy of 0.08%. This sensor measures relative to the atmospheric pressure.

The data acquisition is done with SignalExpress which loads measurements from all pressure sensors every 0.1 seconds. The pressure sensors give an output in voltage with a range of 0 to 10 V which is then converted to bar. This conversion is calibrated by the company Keller that supplied the pressure sensors, and can be found in the Appendix.

Between the pressure sensors and the actual location where the pressure is measured, tubes are placed which are filled with air. The pressure sensor measuring the pressure at the top of the downcomer pipe is placed at the same height as the measured location. This is done to prevent hydrostatic pressure differences between the pressure at the top of the downcomer pipe and the pressure measured by the sensor. The pressure sensors measuring at the bottom of the downcomer pipe and in the tank are located above their measured locations. This is done because the pressure sensors are not fit for a humid environment and cannot thus be placed inside the tank. A tube filled with air is connected between the measured locations and the sensors making sure no hydrostatic pressure difference occurs in the measurements.
5.4 Separation tank

The size of the tank is chosen on the larger side, 143 L and 575x500x500 mm, in order to accommodate larger volume flows in the future. The tank itself is made out of plastic and has a removable lid for filling and draining. In order to check whether the air fully separates from the water the tubing from the separation tank to the pump is made see through. This makes it possible to spot bubbles in the tube and to see whether all air is separated from the water. From first tests it is concluded that even with the maximum amount of air injection (18 L/min) all air separates from the water in time.

In the separation tank a height measurement is present in order to monitor the amount of water in the tank. A pressure relieve valve is installed to prevent too much pressure build up in the tank and a possible explosion. In the figures below these elements are highlighted and it is visible where they are installed.

![Side view of the separation tank](image1)

![Top view of the separation tank](image2)

In order to test the amount of leakage from the tank compressed air is injected in the system with the hope of the pressure in the system rising. As soon as a pressure increase of 0.1bar was noticed the compressed air was turned of and the pressure sensor in the tank measured the time it took for the pressure in the tank to decrease back to atmospheric.

![Pressure change in tank over time](image3)

Figure 20: Side view of the separation tank

Figure 21: Top view of the separation tank

Figure 22: Pressure leakage of the separation tank over time
In Figure 22, the pressure in the tank over time is visible. As visible from the figure the pressure fluctuates heavily around the atmospheric pressure. When plotting a linear fit through the data the fit appears to be horizontal meaning that on average the pressure in the tank does not change. The heavy fluctuation is caused by the sensitivity of the pressure sensor and the movements occurring in and around the system. From this test it can be concluded that the pressure in the tank is approximately the atmospheric pressure and it is not possible to build up any pressure in the tank itself.

5.5 Bubble section

As discussed in Chapter 4 multiple elements are necessary for the acquisition of bubble measurement data. These elements are the square shaft around the downcomer pipe, the diffused light behind it, and the cameras in front of it. In the figures below these are visible.

![Figure 23: A picture of the square shaft around the downcomer pipe with the diffused light on](image)

![Figure 24: A picture of the camera set-up](image)

The cameras are mounted on a flexible stand making it possible for them to move both horizontally and vertically. The cameras that are used are from the company Stemmer and have a frame rate of 74 frames/second. Data acquisition from the cameras is done with the program ePub, one of the images obtained is visible in Figure 24.
Unfortunately is it not possible to use the program previously discussed to measure the bubble size due to distortions from other object in the frame. This can be solved by zooming in more on the images and altering the program to fit the intensity of the images made.

5.6 Flow regime map

The experimental set-up has it limitation in regards to flow regimes that can be reproduced. With use of the T-piece and no straighteners three flow regimes are possible to reproduce; the slug, intermittent, and annular regime. It was found that injecting the air before the measurement section gave the two phase flow enough time to stabilize and the measurements are done at locations where the flow is stable.

In Figures 26, 27, 28 and 29 the different flow regimes are visible. The flow regimes appear to be solely dependent on the volume fraction and mass flux of the flow. It is also noticed that the bubble regime can not be made in this experimental set-up. This is most likely due to the narrow downcomer tube making it easy for bubbles to cluster together. It is also noticed that no matter the regime a thin film of water is always present at the wall of the downcomer tube. In order to make it more visible which regime can be expected Figure 7 is altered to show the range of this experimental set-up.
Shown in Figure 30 the current set-up is mostly limited by the mass flux. This can be improved by enlarging the pump that is used. It is also visible that the bubbly regime can not be reach but a regime consisting of small slugs can. These slug decrease in size with decreasing void fraction.
5. Experimental set-up
6 Experimental results

Experimental results are obtained with the set-up described in the previous chapter. The results are used to compare and validate the theory described in Chapter 3. The comparison of theory and experimental results is done with an increasing level of detail, such that the required level of detail can be determined.

The first step is the determination of friction and pressure losses occurring in the downcomer pipe, that is done with full liquid flow measurements. After, measurements with two phase flow are done and are compared to calculations. Then the flow regime most likely to occur in the HAC is highlighted and analyzed. Finally the repeatability of the measurements and the dependence on environmental conditions is quantified.

6.1 Friction determination

The first experiments that are pressure and flow measurements with liquid flow. These measurements function as a baseline measurements to determine the workings of the system. The friction losses occurring in the downcomer pipe can be determined from these. The pressure drop for full liquid flow without friction, can be determined by the hydrostatic pressure difference,

\[ \Delta P_h = \rho_w g H_{dc} = 0.1525 \text{bar} \]

In Figure 31, the measurements are shown (the blue circles) as well as the calculated hydrostatic pressure difference (the black line). A difference between the calculations and the measurements is present, which is caused by friction in the system.

In order to determine the pressure losses a fit is made through the measurements, the slope calculated from this fit is seen as the velocity dependent friction. The faction \( f \) describing the velocity dependent friction, discussed in paragraph 3.1, can be determined from this slope. This can be compared to the friction factor calculated using Equation (3.10) or (3.12) dependent on whether the flow is laminar or turbulent. The Reynolds number calculated from the measurements with the lowest flow is 452 and the Reynolds number calculated from the measurement with the highest flow is 2000. Meaning that the flow is thus laminar for all measurements. Having a laminar flow...
results, according to Equation 3.12 in a linear relation between the velocity and the pressure loss. When using Equation 3.12 the friction factor has a value of 0.14 at the minimum velocity and decreases with increasing velocity. In Figure 32 a comparison of the friction determined from the experiments and the theoretical friction is shown.

![Comparison of experimental and theoretical friction factor against volume flow and a zoomed in figure of the theoretical friction factor](image)

Figure 32: Comparison of experimental and theoretical friction factor against volume flow and a zoomed in figure of the theoretical friction factor

As shown in Figure 32 both the experimental and theoretical friction factor have the same relation to the volume flow. However, the experimentally determined friction factor appears to be larger than the theoretical friction factor, with a factor of 37. This difference between the calculated and measured friction factor is likely due to an assumption made for calculation of the friction factor. It is assumed that the friction factor is solely dependent on the velocity and all other factors are constant. This difference is likely caused by the pressure sensors being partially extruded in the flow causing additional friction. Salcudean et al. [20] as well as Smith et al. [22] determined that extrusions in flow can cause significant pressure losses possibly explaining this large difference between the measured and theoretical friction factor.

In Figure 33 the pressure difference over the downcomer pipe taking velocity dependent friction loss into account is visible as the purple line. The overall friction found in the system is displayed in Equation 6.2

\[
P_f = 0.5 \nu_w^2 \rho_w f \frac{H_{dc}}{D_{dc}}
\] (6.2)
6.2 Two phase flow measurements

After determining the friction in the downcomer pipe, measurements are done for two phase flow. These measurements are done over the entire available volume flow range of air, 0 to 18 L/min. In Figure 34 the pressure measured at the top and bottom of the downcomer pipe are plotted against an increasing volume flow of air.

It can be seen that the pressure at the top of the downcomer pipe increases significantly with
increasing volume flow until a volume flow of approximately 10 L/min where the curve starts to stagnate. The pressure at the bottom of the downcomer pipe increases slowly with increasing volume flow and the curve stagnates as well around a volume flow of 10 L/min. This stagnation of the curve is most likely caused by the two phase flow stabilizing in the annular regime around 10L/min and no longer changing flow regime with increasing volume flow of air, see Figure 30. It is also noticed that both top and bottom pressure are negative. There is thus a system pressure lower then the atmospheric pressure.

Two shifts in regime are visible when increasing the volume flow of air from 0 to 18 L/min. The first being the shift from the slug regime to an intermittent regime where both slugs and bubbles are visible. This happens between the 0.9 L/min and 1.8 L/min, pictures made with the top camera of this shift in regime are visible in Figure 35.

![Figure 35: Transition between slugs and intermittent two phase flow regime, volume fraction of 0.19.](image)

The change from intermittent to annular flow occurs between the 3.6L/min and 5.4L/min where the flow goes from a mixture of slugs and bubbles to a pipe filled with air and a thin film of water forming around the walls of the pipe. Pictures made with the top camera of this regime shift are visible in Figure 36.

![Figure 36: Transition between intermittent and annular two phase flow regime, volume fraction of 0.83](image)

When considering shifts in regime the Reynolds number should be considered as well, as this determines whether the flow is laminar or turbulent. It is however difficult to accurately calculate the Reynolds number of a two phase as both the density, the dynamic viscosity and velocity should be determined for the mixture of liquid and gas components. The approximation for the density and dynamic viscosity of the mixture can be done using an approximation for the void fraction. Approximating the velocity of the two phase flow is more complex to describe, as most likely liquid and gas components will have a different velocity and descent separately while influencing each other.

This is why for this comparison the Reynolds number for both the water and air flow is calculated separately. Shown in Figure 37, the Reynolds number of the water is flow much higher then the Reynolds number of the air flow. This can be explained by the higher density and dynamic viscosity water has. It can be noted that both the water and air flow are laminar, making it likely to assume that the two phase is laminar as well.

It is visible that the Reynolds number of air increases with increasing volume fraction. This is caused by the velocity of air increasing with increasing volume flow of air. The velocity of water decreases due to the volume flow in the downcomer pipe staying almost constant and the volume flow of air increasing. Resulting in a decreasing water flow when air flow is increasing. This is
visible in Figure 37 as a decreasing Reynolds number for water flow with an increasing Reynolds number for air flow.

6.2.1 Calculation of two phase flow

The model that is used to predict the pressure difference in the downcomer pipe is the current model discussed in Chapter 3. This model consists of the hydrostatic pressure, the friction determined in Section 6.1 and the void fraction of the two phase flow:

$$\Delta p = \left((1-\alpha) \rho_w + \alpha \rho_a\right) g H_{dc} - \frac{v_w^2}{2} \frac{H_{dc}}{D_{dc}} f \rho_w$$  \hspace{1cm} (6.3)

In order to test whether this model sufficiently compares to the measurements the calculating is divided and compared. The first part of the model that is compared to the measurements is the calculation for the hydrostatic pressure. Which for two phase flow contains the void fraction to determine the density of the two phase flow. For the first comparison it is assumed that the void fraction is equal to the volume fraction, thus a slip ratio of 1. In Figure 38 a comparison between this model (black line) and measurements (blue circles) is shown.
Figure 38: Pressure drop against volume flow air, compared to current model

Shown Figure 38 is a difference between the calculation and the measurements. It is visible that the slope of the measurements is much steeper than the slope of the model. This can be explained by not taking friction losses into consideration. The friction is computed as explained in Section 6.1. It is assumed that friction is independent of the void fraction. This results from the observation that there is always a film of water on the walls of the downcomer pipe and the assumption that friction is predominantly wall friction.

The purple line in Figure 39 displays the comparison of this model to the measurements.
Figure 39: Pressure drop over volume flow air, compared to current model with and without friction

There is still a difference between the calculations and the measurements visible, see Figure 39. This difference increases with increasing void fraction, which indicated that possibly the calculation for the void fraction is incorrect. In Section 3.2.2, different methods for approximating the void fraction are discussed. In order to determine which approximation is most accurate all are compared to the measurements.

Figure 40: Pressure drop over volume flow air, compared to different approximations for the void fraction
Shown in Figure 40 is that the approximation for the void fraction determined by Lockhart and Martinelli and the one determine by Armand differ more from the measurements then the homogeneous approximation assumed by the current model. This suggest that possibly these approximation are incorrect for downward flow. Which is why the following assumption for downward flow is tested, $\frac{1}{K}$.

![Graph showing pressure drop over volume flow air compared to different approximations for the void fraction with the assumption for downward flow.](image)

Figure 41: Pressure drop over volume flow air, compared to different approximations for the void fraction with the assumption for downward flow.

From Figure 41 it is shown that both approximation of Armand and Lockhart and Martinelli are closer to the measurements when using the assumption for downward flow. This suggest that this approximation is correct in this situation.

Both Armand and Lockhart and Martinelli seem accurate at low volume flow of air. The Armand approximation appears to become less accurate with increasing volume flow. Overall the Lockhart and Martinelli approximation seems the most accurate approximation when comparing with the measurements. A comparison between the calculation and measurements using this approximation is shown in Figure 42.
Figure 42: Pressure drop over void fraction, compared to the model using the Lockhart and Martinelli approximation

Shown in Figure 42 is that the model compares a lot better with the measurements, there are however still some differences. These differences are especially clear at a void fraction higher than 0.5. The inaccuracy in the sensor is too small to explain this difference. The most likely cause would be the inaccuracy of the Lockhart and Martinelli approximation for the void fraction. It is also possible that unknown friction losses occur in two phase flow that are not yet taken into account in the current model. The relation concerning the initial relative velocity stated by Rice is tested against the measurement data as well. In Figure 43, a comparison of this assumption to the measurements is visible.
As shown in the figure the model considering the assumption made by Rice differs significantly from the measurements. The pressure difference calculated seems to almost linearly decrease with increasing volume flow of air while the measurements curve stagnate around a pressure difference of zero. This large difference between measurements and calculations might be caused by the difference in set-up at which Rice measured this initial value for the relative velocity. Rice has also stated that this initial relative velocity is determined considering a fully developed bubbly regime, something that is not possible with this experimental set-up. At low volume flow of air an intermittent regime that comes close to the bubbly regime can be reached, however at low volume flows the model does not come close to the measurements. One can conclude that this initial relative velocity is thus not correct for this experimental set-up and these measurements.
6.3 HAC calculation

The next figures shown are done in the slug and intermittent regime. This regime comes closest to the assumed bubbly regime in the literature concerning the HAC. A full bubble regime could unfortunately not be reached with this set-up mainly due to the small diameter of the tube and the absence of a straightener that breaks up the slugs into bubbles. In Figure 44 the current model for pressure drop in the downcomer is compared to measurements is visible.

It is shown in Figure 44 that a difference between the measurements and calculation occurs, mainly at low void fractions. The inaccuracy of the sensors is too low to explain this difference. An inaccuracy in the approximation for the void fraction could be the cause for this difference. Therefore, a more accurate relation considering flow regime is required for the void fraction to improve the results. Further research should be done after this.

Another reason for this difference can be the changing air density. Due to compression of the air bubbles the density will increase and thus differ from the density at atmospheric pressure currently used for the model. The reason that this is only considered in these regimes is when further increasing the void fraction, as shown in Figure 34, the pressure in the downcomer pipe reaches almost atmospheric pressure. Therefore the change in density will not be significant. Taking the density change into account the following comparison between calculation and measurements is made.

In Figure 45 it is shown that the difference between the current model considering density change and not considering density change is insignificant. When looking at the measurement with the lowest void fraction, the density at the top of the downcomer is 1.0301 kg/m³ and at the bottom of the downcomer 1.1987 kg/m³. This is a change of 14% in density. This results in a small change visible in the model, see Figure 45. This change will decrease with increasing void fraction. In Figure 46 the density of air at the top and bottom of the downcomer over increasing void fraction is shown.

Figure 44: Pressure drop over void fraction, compared to current model in the HAC regime
Visible from Figure 46 is that the top density increases significantly more than the bottom density. This is in line with the pressure measurements visible in Figure 45.
6.3.1 Drag coefficient test

In Section 3.2.3 multiple different drag coefficient theories are discussed concerning the drag on an air bubble. Because it is not possible to create a bubbly regime in the experimental set-up it is difficult to determine the right coefficient by comparing calculations to measurements. When looking at pictures taken of the bubbles in the experimental set-up an approximation for the most accurate drag coefficient can be made. In Figure 47 relatively large bubbles are shown as they travel through the downcomer pipe. From this figure it is clear that the three bubbles that are visible are not round and change in shape over time.

![Figure 47: Bubble development in the downcomer pipe, for volume fraction of 0.1](image)

It is also visible that the intermittent regime, which has the most bubbles, experiences bubbles from all different sizes. Pictures of a bubble cluster over time are shown in Figure 48. The smallest bubbles are approximately round, the larger bubbles are not. From the figure it is also visible that the bubbles interact. The bubbles tend to cluster together which happens most often at the top of the downcomer pipe where the flow start descending.

![Figure 48: Bubble cluster development in the downcomer pipe, for volume flow of 0.28](image)

Visible from these results is that there is a wide spread in bubble size and especially larger bubbles can not assumed to be round. It is also noticed that the bubbles occur in clusters, and they tend to interact with each other and sometimes merge together. Based on these conclusions the relation derived by Dijkhuizen using the correction factor for bubble swarm by Roghair gives the most accurate approximation for the drag coefficient. For the calculation of this drag coefficient the bubble diameter needs to be known. The bubbles occurring the most frequent in the intermittent regime range from 1 to 5mm. When considering these values the drag coefficient experienced by bubbles in the intermittent regime is visible in Figure 49.

![Figure 49](image)

Visible from Figure 49 is that the drag coefficient has the same non linear relation to the void fraction independent on the bubble size. The peak increases with increasing bubble diameter and shifts to a lower void fraction.
6.4 Fluctuation dependent on environment

The measurements are done in a laboratory which has an ambient temperature that fluctuates heavily during the day and season. In order to determine the effect of the environment on the system the same measurement is done on multiple different days and thus different ambient temperatures.

Figure 50: Pressure measurements for change environment and water temperature

Shown in Figure 50 is the environmental temperature changing more than the water temperature.
This is due to the temperature change being much slower in water than in air. The pressure measurements appear to fluctuate with changing temperature. It is also visible that the fluctuation due to temperature change falls within the fluctuation caused by the inaccuracy of the pressure sensors. It is thus impossible to conclude whether this fluctuation is indeed caused by temperature change.
7 Conclusion

It became apparent that both the models by Rice and Adrieansen contain a presumption that is not fully understood. This regarded the initial relative velocity that is assumed constant. The relation between the void fraction and flow velocities used in literature produces a results inconsistent with results from both these models. It is also found that the assumptions considering flow regime and drag coefficient are inaccurate. This resulted from observations found in literature considering the HAC and descending two phase flow.

The pump used for water transport is considered as insufficient, due to the inability to reach the previously designed volume flow. This has an impact on the set-ups ability to recreate the bubbly regime, which is required to validate certain assumptions. It is found that pressure measurements are more complex than initially anticipated. The top pressure sensor has to be located next to the measured location in order to prevent errors in the measurements. The pressure sensors inevitably extrude into the flow causing additional pressure losses, which became apparent in Section 6.1.

The friction factor determined from experiments is found larger than theoretically predicted. This is most likely due to the previously mentioned extruding pressure sensors. It also became apparent that the assumption for homogeneous void fraction was incorrect and the approximation made by Lockhart and Martinelli gave a better representation of the experimental results.

When looking at the range the HAC is most likely to operate in, a difference between model and measurements occurs at low void fractions. The assumption that this is caused by changing density of air due to compression was incorrect. This appeared to have an insignificant impact on the model. The approximation by Lockhart and Martinelli is possibly not accurate enough as it does not consider flow regime. Considering different flow regimes will increase the accuracy of the current model.

The approximation by Dijkhuizen regarding the drag coefficient for air bubbles is likely the most accurate. It is found difficult to compare this assumption to experimental results due to the inability to reproduce the bubbly flow regime. This conclusion is based on the observed shape and mutual interaction from bubbles in the experimental set-up.

The current model taking friction and void fraction into consideration reproduces the pressure change measured in the experimental set-up of the HAC with a relatively high accuracy, see Figure 44.
7. Conclusion
8 Recommendation and Discussion

A few recommendations can be mentioned after completing this study as well as a few points of discussion.

The unclearity still present in the models by Rice and Adrieansen makes it impossible to properly compare this model to the current model. It is recommended to look further into these models regarding the compression of air and the assumed initial relative velocity.

As previously mentioned it is not possible to reach the bubbly regime with the experimental set-up, which makes it impossible to test certain assumptions concerning the HAC. Concluded from Figure 30 an insufficient mass flux is the problem. Increasing the volume flow by replacing or adding a pump will make it possible to reach the bubbly regime. If increasing the pump power appears to be insufficient the next step would be to add straighteners to the experimental set-up. These straighteners will brake up slugs of air into smaller bubbles and thus forcing a bubbly flow regime. Another addition to the experimental set-up would be a pressure sensor in between the top and bottom sensors on the downcomer pipe. This makes it possible to measure pressure development, which will give a better inside in the relation between pressure development and two phase flow properties. When adding velocity sensors as well the influence of velocity development can be determined.

If the images currently made with the high speed cameras mounted on the set-up are altered, as well as the program needed to develop these images, it will be possible to determine bubble size and behavior. This will lead to a better understanding regarding flow regime and transition between flow regimes. The approximation now used by Lockhart and Martinelli does not take the flow regime into consideration which leads to inaccuracies in the current model. When a more accurate flow regime map is drawn and more inside on the flow regimes is gathered it becomes possible to take the flow regime into account. This will increase the accuracy of the current model.
A Appendix

The derivation of the mass and momentum conservation equations used by Rice [5] is described by,

\[ dv_w = \left( \frac{\dot{m}_w}{v_w-v_r} + \frac{\dot{m}_w}{v_w} \right) gdz - f \frac{v_w^2}{8} \pi \rho_w dz D_{dc} \]
\[ \left( A_{dc} + \frac{\dot{m}_w v_r}{6p} \right) \left( \frac{\dot{m}_w + \dot{m}_a}{A_{dc} + \frac{\dot{m}_w}{6p}} - p \left( \frac{1}{v_w-v_r} + \frac{\rho_w v_r^2 A_{dc} - m_w v_w}{1+ \frac{v_r}{6(v_w-v_r)}} \right) \right) \]

\[ dp = -pdv_w \left( \frac{1}{v_w-v_r} + \frac{\dot{m}_w}{\rho_w v_r^2 A_{dc} - m_w v_w} \right) \left( 1 + \frac{v_r}{6(v_w-v_r)} \right) \]

In Table 1 a list of parts for the experimental set-up can be found. In Table 2 the calibration of the pressure sensors can be found.

<table>
<thead>
<tr>
<th>Table 1: List of parts</th>
</tr>
</thead>
<tbody>
<tr>
<td><strong>Frame</strong></td>
</tr>
<tr>
<td>Hepco profile</td>
</tr>
<tr>
<td><strong>Downward pipe</strong></td>
</tr>
<tr>
<td>downward pipe</td>
</tr>
<tr>
<td>square shaft around the pipe</td>
</tr>
<tr>
<td><strong>Water control system</strong></td>
</tr>
<tr>
<td>pump</td>
</tr>
<tr>
<td>tubing</td>
</tr>
<tr>
<td>needlevalve</td>
</tr>
<tr>
<td><strong>Separation tank</strong></td>
</tr>
<tr>
<td>tank</td>
</tr>
<tr>
<td>pressure relieve valve</td>
</tr>
<tr>
<td><strong>Air control system</strong></td>
</tr>
<tr>
<td>mass flow controller inlet</td>
</tr>
<tr>
<td>air tubing</td>
</tr>
<tr>
<td>mass flow controller outlet</td>
</tr>
<tr>
<td>reduction valve</td>
</tr>
<tr>
<td><strong>Pressure measurements</strong></td>
</tr>
<tr>
<td>pressuresensor 1</td>
</tr>
<tr>
<td>pressuresensor 2</td>
</tr>
<tr>
<td>pressuresensor 3</td>
</tr>
<tr>
<td><strong>Flow measurements</strong></td>
</tr>
<tr>
<td>flowsensor</td>
</tr>
<tr>
<td><strong>Bubble measurements</strong></td>
</tr>
<tr>
<td>light diffuser</td>
</tr>
<tr>
<td>lights</td>
</tr>
<tr>
<td>Camera</td>
</tr>
</tbody>
</table>
Table 2: Calibration pressure sensors

<table>
<thead>
<tr>
<th>Top downcomer pipe</th>
<th>2</th>
<th>Pressure (bar) = -0.0022*Voltage + 5.0022</th>
</tr>
</thead>
<tbody>
<tr>
<td>Bottom downcomer pipe</td>
<td>1</td>
<td>Pressure (bar) = -0.0012*Voltage + 5.0016</td>
</tr>
<tr>
<td>Separation tank</td>
<td>0</td>
<td>Pressure (bar) = -0.021*Voltage + 2.0010</td>
</tr>
</tbody>
</table>

A circulation pump is used with a maximum volume flow of 58 l/min and head of 3.6m, in practice the maximum flow in this experimental set-up is around 6.8 l/min. Using these known points a pump curve for this pump is determined. Due to the lack in measurement points the pump curve will be inaccurate but a first approximation can be made, see Figure 51.

Figure 51: The pump curve measured for the circular pump used in the experimental set-up

In Figure 52 the schematic drawing of the downcomer pipe used in the experimental set-up is shown.
Figure 52: Schematic drawing of the downcomer pipe
References


[6] Li-Ting Chen, W. Rice Properties of air leaving a hydraulic air compressor (HAC), Arizona State University, Department of mechanical and energy systems engineering. 1983.


