MULTIPLE LIFT TUBE BUBBLE PUMPS TO INCREASE THE CAPACITY OF DIFFUSION ABSORPTION REFRIGERATORS

Prepared By: A J Bennett, masters student in the Department of Mechanical Engineering, UCT.

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Declaration

I Andrew James Bennett hereby declare that the work presented in this thesis is my own work unless otherwise stated. It is presented as a partial fulfilment of a masters degree in mechanical engineering.

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Synopsis

Diffusion absorption refrigeration is a method of refrigeration that uses heat to drive the solution circulation in refrigeration system and produce cooling. It therefore has the potential to be used in areas where there is no electricity or where a cheap heat source is available. This system has suffered from two main disadvantages, its small size and a low efficiency. This thesis concentrates on one component of the unit, the bubble pump, because it has the potential to enable larger more efficient refrigeration units to be build and used.

It is believed that the use of addition lift-tubes in the bubble pump will enable it to use heat energy more effectively and efficiently. The performance of the bubble pump has been predicted using a mathematical model for a single substance pumping system, and not for the binary substance used in the refrigeration system.

Two experimental units were built to test the viability of the multiple lift-tube pumps. One used only water as a working fluid, and its results were compared to the results predicted by the mathematical model. The other was a commercially available diffusion absorption refrigerator that was fitted with a multiple lift-tube pump and used to test its affect on the unit's performance.

The experiment with the diffusion absorption rig found that the addition of extra lift-tube reduced the boiler temperature for the same heat input and allowed the boiler to use larger quantities of heat energy. The lower boiler temperature allowed the other components in the system to work more efficiently at heat inputs that were greater than when a single lift-tube was used.

The multiple lift-tube bubble pump was found to be a viable and workable solution that will allow new designs in diffusion absorption refrigerators to have increased capacity and performance compared to their predecessors.

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Nomenclature

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A	Area of lift-tube	m^2
C_p	Heat capacity under constant pressure	J/kgK
d	Diameter of the lift-tube	m
f	Friction coefficient	
g	Gravitational acceleration taken as 9.81 m/s	
h	Convection coefficient	W/m^2
h_{fg}	Enthalpy of vaporization	J
H	Liquid head in the lift-tube	m
k	Conduction coefficient	W/mK
K	Loss coefficient	
L	Height of the lift-tube	m
$\dot{m}$	Mass flow rate	m^3/s
P	Pressure	Pa
Q	Heat energy	Watt
Re	Reynolds number	
s	Slip ratio	
t	Time	s
T	Temperature	K or °C
Th	Temperature hot line	°C
Tc	Temperature cold line	°C
v	Specific volume	m^3/kg
V	Velocity	m/s
Vol	Volume	m^3
$\dot{V}$	Volume flow rate	m^3/s
x	Quality	
Δ	Change or difference	
ε	Effectiveness	

ρ	Density	kg/m^3
σ	Surface tension	N/m

Subscripts

abs	Absorber
amb	Ambient
boil	Boiling
con	Condenser
d	Diameter
evap	Evaporator
ext	Outer surface
f	Fluid
gen	Generator/ Boiler
int	Interior surface
mid	Mid interface
R	Volume flow rate
rec	Rectifier
s	Steel
surf	Surface
v	Vapor
w	Glass wool insulation
1	Position 1 or inlet
2	Position 2 or outlet

Chapter 1

Introduction

Refrigeration is a technology that is an integral part of our modern lives. It has improved the living standards of many people by allowing, easy, long-term storage of perishables, and living and working areas with air conditioning. It is however, not a new technology, and has been in use for a long time, with evaporative cooling being the early forerunner to modern refrigeration. By far the most common refrigeration cycle in use today is the vapour compression cycle. This cycle is well known and provides reliable and efficient refrigeration. But it requires the use of a mechanically driven compressor, usually driven by an electric motor that requires access to electricity. There are, however, a number of other cycles that can also be used to produce cooling or refrigeration.

With the concern for the environment and the sustainability of the planet's energy resources growing, there is an increased focus towards energy efficient and environmentally friendly technologies. There is also an increased focus on technologies that will increase living standards in rural or third world areas. When looking at refrigeration technologies, a cycle with potential in this regard is the absorption refrigeration cycle. It has the ability to use heat energy to produce refrigeration. Thus cooling can be achieved by using waste heat or sufficient solar radiation. This gives the cycle the potential of producing refrigeration at very low cost to the user [5,7,19,27]. The heat energy needed to produce refrigeration can also be obtained from a number of other more standard sources, examples being: LPG (Liquefied Petroleum Gas), natural gas or kerosene flames and electricity [2,7,10,21].

A variation of the absorption refrigeration cycle is the diffusion absorption refrigeration *cycle*, which does not require a mechanical pump to drive the liquids in the system. The diffusion absorption cycle makes use only of heat energy to operate and drive the system. It uses a heat-activated pump called a boiler pump or bubble pump, which consists of a boiler and a lift-tube. The pump works by the vapour produced in the boiler, which then pushes the liquid up the lift-tube, thereby driving the fluids up to a head. Gravity then maintains fluid circulation through the system. It also has the ability to use different heat sources, making it a very adaptable refrigeration system, ideal for use by campers and in rural areas, which need to be able to utilise various available energy sources e.g. solar gas, electricity, wood etc. [7,21].

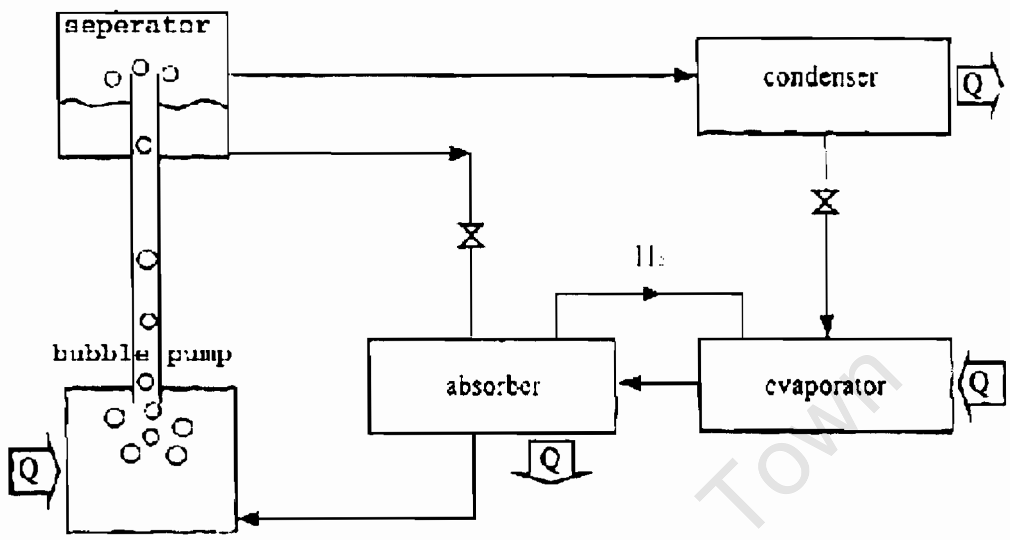

Figure 1.1) Simple line diagram of the diffusion absorption refrigeration plant.

The diffusion absorption refrigeration cycle is simply explained as follows, with figure 1.1 being a simplified diagram, only intended to give an overall concept of the system. On the left hand side of the diagram is the bubble pump with heat (Q) being added to a fluid, which consists of a refrigerant and absorbent. The heat releases the refrigerant as a vapour that then lifts the remaining fluid up the lift-tube. The vapour and fluid mixture is then separated into two streams in the separator. From there the gas goes into the condenser to form the refrigerant and the fluid moves down into the absorber. The condensed refrigerant then moves into the evaporator, where it receives energy and boils at low pressure. Thereafter the low-pressure refrigerant gas is directed to the absorber where it is absorbed into the absorbing fluid, from where it flows by gravity to the bubble pump to start the cycle again. A third substance in the system is a gas that acts as a pressure equaliser between the refrigerants low pressure and the condensers high pressure. A more detailed explanation of how the cycle works is given in the chapter 2.

From the above explanation it is noted that the system has no moving parts and there are a number of advantages associated with this.

- There is no potential for frictional wear, which therefore allows for reliable long-term operation [2,7,21].
- It allows for almost total vibration and noise free operation, which is good for sound sensitive areas like hotel rooms.
- Reduced complexity of a system's production. The result is that the diffusion absorption system is relatively simple and cheap to manufacture when compared to a vapour compression refrigerator [7]. It requires only steel tubing, welding equipment, the systems fluids of liquids and gasses and a method of charging the system, to manufacture it.

The most commonly available diffusion absorption refrigerators use hydrogen, ammonia and water as working fluids. This type of system has been in use, and produced for domestic use, by various companies since Platen and Munters patented the system in 1929 [2] and it is this system that this dissertation will concentrate on. It was chosen because of its applicability to the environmental conditions, the availability of expertise and equipment, and because the fluids used in the system are environmentally friendly. The results attained should however be usable in the design of other absorption type systems, for example the Lithium bromide/water system [16.].

The objective of this dissertation is to study one of the components of the system, the bubble pump. This is because it is the prime mover of the fluids around the system and vital to the production of refrigerant. It is foreseen that by studying this component there is the potential for increasing the capacity and efficiency of the diffusion absorption refrigeration system. This is desirable because the small capacity and low efficiency of the system are two of its major disadvantages, relative to the conventional vapour compression cycle [2,20]. It is proposed that the use of extra lift-tubes will improve the system's performance in both these regard.

For this study two apparatus were built, and a mathematical model developed by the author. The first apparatus used only boiling water and steam as its working fluids, to evaluate and test multiple lift-tube bubble pumps. The data collected was compared with the mathematical model, which predicts the volume flow rates of the liquid versus the quantity of steam produced (which is related to the heat energy input). The second apparatus was a diffusion absorption refrigerator fitted with a multiple lift-tube bubble pump. It was tested over a range of boiler heat inputs and a number of lift-tubes in order to collect data for a wide range of working conditions and establish a pattern of its performance and refrigeration capacity.

The following document has been divided into chapters as follows. Chapter 2 will explain in detail how the diffusion absorption refrigerator system works. It will also give a detailed description of the working of the bubble system and a summery of test results. Chapter 3 begins with the reasoning for testing using a multiple lift-tube bubble pump. Thereafter, it explains the design and construction considerations of the two rigs, and minor adjustments for smooth operation. Chapter 4 gives a detailed description of the mathematical model used to predict the bubble pump's performance. Chapter five and six presents the collected data, results and detailed discussions on the apparatus performance. The conclusions and recommendations follow in chapter 7.

Chapter 2

Background and Literature review

This chapter will give a detailed description of how the diffusion absorption refrigeration system operates. Following this is a study on the theoretical and empirical models on the boiler or bubble pump.

2.1) Background on the absorption refrigeration system

The diffusion absorption refrigeration system's performance is very dependent on how the components of the system interact with each other [19,21,23]. Therefore, to understand the role and workings of the bubble pump, it is beneficial to understand how the diffusion absorption system works. All the concepts and phenomena used in the systems operation are simple, but combining them within one system where they can affect and interact with each other, result in a complex system the behaviour of which can be difficult to predict and analyse.

A description of the fundamental principles used in the diffusion absorption system is presented first. Thereafter, a detailed description follows on how the diffusion absorption system works and produces cooling. Specific intricacies of the system are highlighted, particularly those that are key to important design areas in absorption cooling.

2.1.1) Operational processes of a diffusion absorption refrigerator

The diffusion absorption refrigerator uses a number of different fluid properties, but this section will only focus on two key areas namely, Dalton's law of partial pressures and absorption. The mechanics of the bubble pump subsystem is looked at in more depth in section 2.2 of this chapter.

2.1.1.1) Dalton's law of partial pressures

The low temperature required for refrigeration is achieved by the use of Dalton's law of partial pressures, which states, "The total pressure of a gas mixture is the sum of the partial pressures of the individual gasses in the mixture'' [3]. The result of this law is used instead of the expansion valve found in vapour compression refrigeration cycles. In the diffusion absorption refrigerator, it allows the reduction of the refrigerant's pressure by the addition of an inert gas, thereby making the total pressure equal to the sum of the gas pressure and the refrigerant pressure. The subsequent decrease in the refrigerant pressure allows the refrigerant to vaporise at low temperatures.

2.1.1.2) Absorption systems and cycle

The "absorption" component of the name refers to the reaction between two of the fluids in the system, which are known as a binary system. In absorption refrigeration systems, they consist of a *refrigerant* and an *absorbent*, with a common example being ammonia and water. Binary systems are well documented with some 40 refrigerants and 200 absorbents known at present [20]. There is ongoing research directed at binary systems, with the aim to find better absorbents and refrigerants for use in refrigeration and air conditioning applications [20]. At present though the most commonly used binary systems are ammonia/water and Lithium bromide/water [20].

The absorption process/cycle can be simply explained using two vessels that are connected to each other as seen in figure 2.1. The left vessel contains refrigerant and the right one the absorbent. The solution on the right will absorb the refrigerant vapour from the system, thereby dropping the system's pressure. As a consequence of the drop in pressure, the refrigerant will then be able to vaporise at a lower temperature. The amount of energy received by the refrigerant, from its environment, at low temperature is known as "refrigeration effect". As this occurs, the solution in the right container becomes more concentrated in refrigerant. The absorption process on the right is usually exothermic, with heat being rejected to the environment. There is however, a limit to the concentration of refrigerant in the absorbent. Beyond this limit, no more refrigerant can be absorbed. It is then possible to separate the refrigerant from the absorbent, which is usually done by heating the solution container. The heat, forces the refrigerant out of the absorbent and into a gaseous phase, which increases the system pressure. It is subsequently possible to condense the refrigerant vapour in the refrigerant side by transferring heat out of the refrigerant vessel and into the environment, thereby completing a simple, intermittent, absorption cycle.

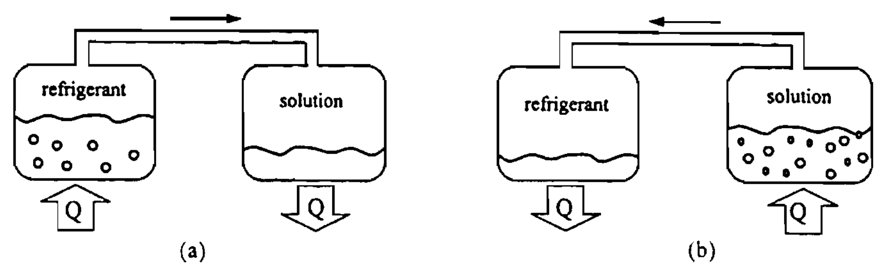

Figure 2.1) Basic absorption process a) solution absorbing refrigerant and causing refrigeration effect in the left container b) reversing the process

2.1.2) Description of the ammonia/water diffusion absorption refrigerator cycle.

A diffusion absorption refrigerator system is a hermetically sealed unit, in which all the components are connected together so that there are no pressure differences throughout the system. The explanation to follow refers to figure 2.2, which shows a schematic of a common diffusion absorption refrigerator layout. This type of system is usually charged with a 25-35% ammonia/water solution by mass [2,21,23], and is pressurised with hydrogen gas to the desired working pressure, usually between 6 and 25 bar, depending on the design requirements. Helium gas is an alternative to hydrogen if added safety is desired, but it is less effective [21].

2.1.2.1) System Operation and Layout

This is a step-by-step description of the system with each component discussed sequentially, starting with the component to which energy is added to activate and run the system. Then the discussion will move to the other components of the system, in the same order that the systems fluids flow through. Heat is applied to the generator/boiler where it desorbs ammonia gas out of the strong ammonia/water solution. (The strong solution is the liquid coming from the storage tank with a high concentration of ammonia.) As the near pure ammonia gas is produced it escapes the boiler up the lift-tube. On its way it entraps slugs of liquid in the lift-tube and drives them up into the liquid-vapour separator, where as the name implies, the liquid and the vapour streams are separated.

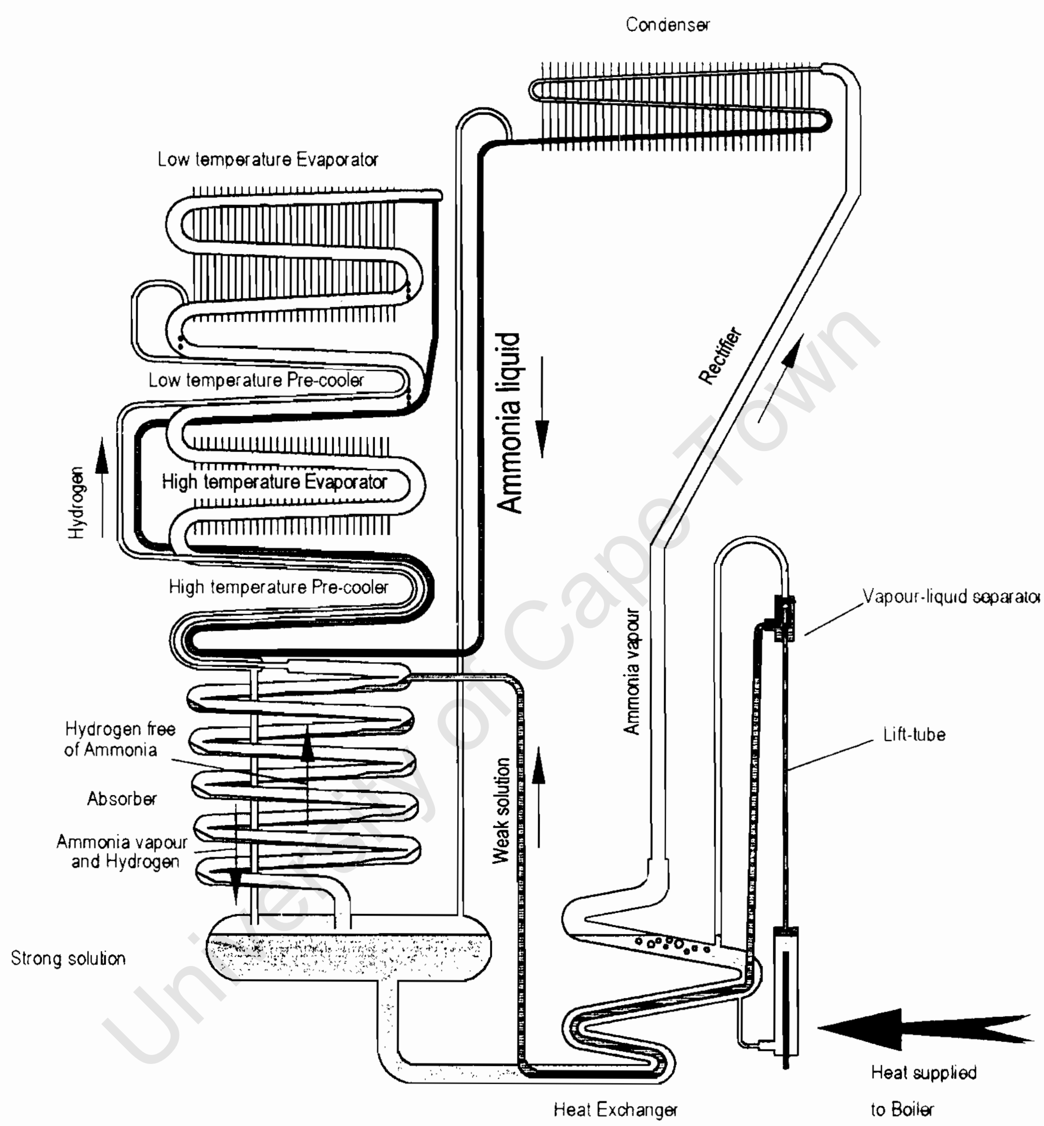

Figure 2.2) Diagram depicting the diffusion absorption refrigeration system.[24]

The liquid stream coming from the separator is called the weak solution, because it has a lower concentration of ammonia than the strong solution. The weak solution stream moves into the solution heat exchanger, where heat is transferred to the cooler strong solution coming from the storage tank. This is done to make the system more energy efficient, and has the added effect of reducing the physical size of some of the components. After the heat energy has been transferred, the weak solution is fed into the top of the absorber from where it descends the absorber's tube and comes into contact with the refrigerant vapour.

From the separator, the gas stream moves via an analyser upward into the rectifier where it is cooled and excess water vapour is condensed out. This is necessary because, during boiling of the ammonia/water solution, a quantity of water vapour is released along with the ammonia vapour, which if unrectified, would yield a refrigerant of poor quality. During rectification a calculated portion of the ammonia gas is also condensed which acts as a reflux, necessary to ensure the purity of the condensed refrigerant. This reflux from the rectifier is returned to the analyser, and joins the weak solution stream before it goes through the solution heat exchanger.

The remaining vapour stream moves into the condenser, (the highest point in the system), where the purer ammonia vapour is condensed to form the refrigerant. After condensation the refrigerant leaves the condenser and moves by gravity through a liquid vapour lock and through heat exchangers into the evaporator.

In the evaporator the liquid refrigerant is brought into contact with hydrogen gas and as a result of the effect described by Dalton's law, the partial pressure of the refrigerant drops. This allows it to absorb heat energy from the environment, and thus increasing the ammonia concentration in the vapour. In addition to the increased concentration, the vapour mixture is also colder, and denser than the purer hydrogen entering simultaneously, and thus, as a mixture begins to sink to the bottom of the evaporator.

The vapour mixture at the bottom of the evaporator is piped down into the absorber and made to flow in a counter-flow direction over the down flowing cooled weak solution. Because the weak solution is cool it can absorb the ammonia out of the vapour mixture and from the strong solution. This absorption process produces heat that needs to be removed to allow the continued absorption of ammonia gas into the solution. This extraction of heat is the absorber's principal function. It has been assumed that the absorber has two functions

1. To act as a heat exchanger for heat transfer
2. To provide a large enough surface area for mass transfer

Of the two the heat transfer is a more important function. As the ammonia gas is absorbed into the absorbers solution, the solution is turned into the strong solution, which is piped down into the storage tank.

After the ammonia gas has been absorbed and thus removed from the hydrogen/ammonia mixture in the absorber, the remaining hydrogen gas is piped into the evaporator via a series of heat exchangers to the evaporator, and rejoins with the refrigerant. The heat exchangers are necessary because the hydrogen has picked up heat in the absorption process.

2.1.2.2) Environmental effect

The system described above is one of the more widely used arrangements for diffusion absorption refrigeration systems. For most conventional applications of this system, all the heat that is rejected is done via natural convection to the environment. The refrigerator is therefore very dependent on the environmental temperature to define its effectiveness. A larger temperature difference between the components and the environment will allow for a greater amount of heat to be rejected, and thereby improving the absorption and condensation processes and reducing the systems physical size. This is also a factor that must be kept in mind when placing the refrigerator in a certain area, for example a warm area with direct sunlight, or restricting the airflow over the absorber and condenser, which will negatively affect the refrigerators cooling ability.

2.2) The bubble pump

In many applications a more powerful refrigeration unit is desirable. A logical solution would be to increase the heat input, and produce more vapour, and thereby increase the refrigerant flow rate. But this will not necessarily work because of the design and physical limitations of the refrigerator system as a whole. These limits only allow the refrigerator to work effectively within a narrow band [19]. Therefore, increasing heat input is not a straightforward solution.

The component of the diffusion absorption refrigerator to which heat is added is the bubble pump. How the pump will react to the increased heat input is therefore the first concern when an increase to the cooling capacity of the system is required. The bubble pump is thus studied in more detail, specifically at how it operates and which parameters affect its performance. This will be useful in understanding how to adapt the pump to different operating conditions.

2.2.1) The bubble pump applications

The bubble or boiler pump is the heart of the diffusion absorption refrigerator. It is the prime mover of the fluids within the system, and the component to which heat is added to desorb the refrigerant from the absorbent. It plays a vital role in the system, and should be designed accordingly. Most designing to date has been done by trial and error and/or copying existing systems [21]. This has lead to the manufacture of some refrigerators that do not operate effectively, or at all [19,23]. However, not all poor refrigeration performances are attributed to the bubble pump design.

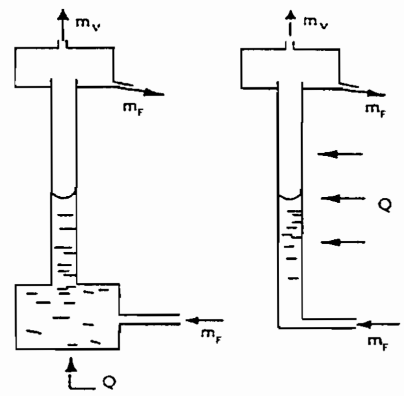

Figure 2.3) Two examples of how energy can be added to a bubble pump.

Little is written about bubble pump application in diffusion absorption refrigerators. There is, however, research available concerning air-lift pumps abilities, due mainly to the many applications for which air-lift pumps are uniquely suited, which vary from dredging, underwater mining and exploration to pumping volatile, explosive or corrosive liquids [6,10,12,22]. Bubble pumps and air-lift pumps are comparable, (both exhibit the same flow patterns) with the major difference being the way by which the gas used to pump the liquid is added [8,14,17]. A bubble pump used for refrigeration produces its own gas, whereas an air-lift pump has the gas added from an external source.

2.2.2) How the bubble pump operates

The bubble pump consists of a vertical tube into which liquid and vapour is added at the lower end. This is known as the lift-tube. The liquid fills the lift-tube to a specific depth, governed by the design of the system. Then, to pump the liquid, a gas is added or created at the lower end of the lift-tube. The gas bubbles quickly coalesce to form bubbles of larger size, and these act like pistons in the lift-tube, driving liquid slugs up the tube until the liquid is ejected out at the top of the tube. This can be seen graphically in figure 2.4. The driving force for pumping comes from the buoyancy of the bubbles and friction between the bubbles and the liquid [4,23].

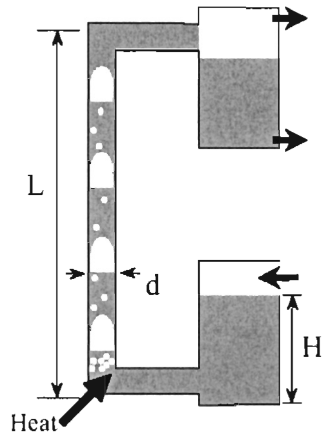

Figure 2.4) Diagram of a bubble pump showing relevant dimensions.

The pumping process can alternatively be looked at as a drop in density in the lift-tube [15]. The gas and liquid mixture in the tube has a lower density than only liquid, and thus creates an imbalance of pressures that will push the less dense gas/liquid mixture up the lift-tube [15]. A large enough imbalance will cause the mixture to be ejected out the top of the lift-tube. This concept is the same as the U-tube manometer principal of balancing pressures.

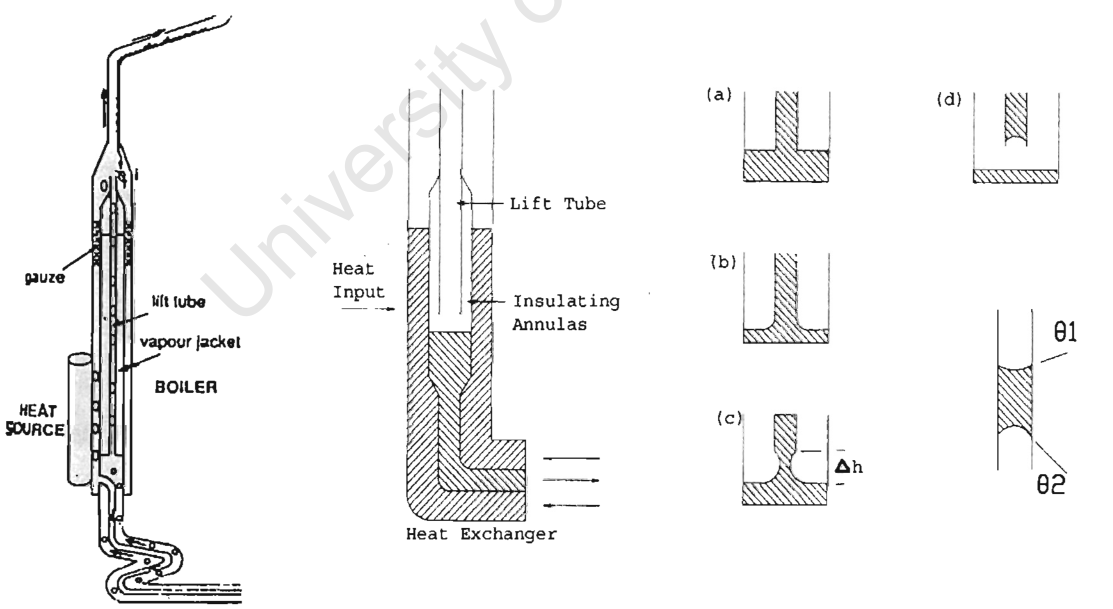

Figure 2.5) An alternate bubble pump design in the form of a surface tension pump. Left is the pump and right is a diagram of the pumping cycle

2.2.3) Practical bubble pump arrangements

There are number of different bubble pump arrangements that can be used for the diffusion absorption refrigerator. The type of arrangement used for each application is made to suit the type of heat source used, and the volume of liquid and gas pumped.

The surface tension pump can be seen in figure 2.5. The figure shows the pumping action of a surface tension bubble pump that is still used for small refrigeration units. As can be seen from figure2.5 the design is complicated, and if not designed and manufactured correctly it will not work satisfactorily [19]. A common handicap of this type of refrigerator is its inability to survive movement. In addition, because of its design characteristics, heat transfer to the liquid can be difficult. For both the reasons given above this type of design is not regarded as robust and reliable.

Another arrangement is to simply heat the lift-tube to cause boiling to occur. This method is the simplest, and can be effectively used if there is a hot vapour or liquid available for generating gas. An example would be condensing steam, or using the exhaust gas of a combustion process [17,23].

A third method is to use a boiler with a lift-tube attached to the top of the boiler [14]. This allows the bubbles formed in the boiler to naturally migrate up the lift-tube to perform the required pumping. This arrangement is very flexible, and is particularly recommended if electricity is used to perform the heating, because it can be designed to have the heat source immersed in the liquid, with the advantage of transferring the heat energy with minimal losses.

2.2.4) Flow patterns for two phase flow in a vertical tube

Two-phase flow of vapour and liquid in a vertical tube can take on a number of different flow patterns [14,17,23]. These patterns are dependent on the geometry, pressure, heat-flux and flow rates within the tube. It is generally agreed that there are five distinguishable flow patterns based on visual observations for two-phase flow in a vertical pipe. They are,

- Bubble flow
- Slug flow
- Churn flow
- Annular flow
- Mist flow

As the gas flow rate is increased, the flow pattern changes from bubble to mist flow, see figure 2.6 [4,23].

Figure 2.6) Flow patterns in a vertical tube.

Bubble flow is when bubbles of gas are contained in a continuous liquid phase. Bubble shapes ranging from one extreme with small spherical bubbles, to another of larger bubbles with a spherical cap and flat tail, not occupying the full diameter of the tube.

Slug flow refers to bullet shaped slugs of gas contained within the liquid. The bullet-shaped bubble is sometimes known as a Taylor bubble. Their bubble diameter is approximately the same diameter as the pipe, and is separated from the pipe wall by a thin film of liquid that slowly descends. The flow pattern is of liquid slugs separated by bubbles, with the distance between successive slugs relative to the velocity of the bubbles up the tube.

Froth or churn flow is a highly oscillatory flow, with a tendency for each phase to become continuous with irregular interfaces. This happens due to the breakdown of large unstable bubbles in slug flow. This flow has an oscillatory or time varying nature.

In **annular flow** the liquid forms an annulus against the wall with a high velocity gas core. At lower gas velocities the annulus is thicker and as the velocities increase it becomes thinner with more liquid being entrained in the gas as droplets. The gas core may contain liquid droplets entrained in it due to the breaking up of waves formed on the tube's wall.

Mist flow is characterised by the total dispersion of the liquid in the gas. It happens at very high gas flow rates, where the liquid annulus has disappeared.

2.2.5) Flow pattern prediction/estimation

The prediction of these flows and flow rates in a tube is not trivial, and current research is aimed at allowing the researcher to accurately predict when specific flows will be prevalent, and also to predict the transition between different flow patterns. One method is the use of flow maps that graphically show how a system will respond to different conditions. There are a number of different flow maps available, which have been used with varying degrees of success [4,7]. They can however, be misleading, because they give the flow pattern transition as a definite line, while in reality the transition is usually over a wider range of flow rates.

From previous work done it is noted that slug flow will be the flow pattern that is useful within a bubble pump [5,7,15,16,18,22,23]. This pattern works over the widest range of flows and is the flow pattern within which bubble and air-lift pumps are most effective and efficient. Other flow patterns have been documented in bubble pumps, but these are less effective and their presence is undesirable [14]. Knowing this, it narrows the focus of the theoretical work, and the prediction of bubble pump's performance using mathematical models can be made using only the slug flow regime.

2.2.5) Analytical theories used to predict the flow in bubble pumps

As has been mentioned in 2.2.4 to model the flow pattern in a bubble pump is not easy. All the early air-lift pump models used empirical equations or energy balance equations [15]. These models are, however, not flexible and could only predict accurately for the systems from which they have been derived. If they were used for different systems then the empirical equation's solutions are not reliable [6,15,22]. A more universally applicable approach is to use two-phase flow theory, which physically describes the flow within the tubes. Using this approach a number of different prediction models have been designed. The models have mostly been designed to predict performances of air-lift pumps but being very similar in nature to bubble pumps the predictions have and can be used to work for bubble pumps [7].

The early work on air-lift pumps using two-phase flow was done by Nicklin [15]. Nicklin's method used point conditions in the lift-tube, instead of viewing the system as a single unit. Because of this, changing gas volumes and the resulting velocity changes could be directly accounted for. To simplify the equations Nicklin made use of some assumptions.

- The entrance and exit effects had a negligible effect on the efficiency. Khalil et al [12] who subsequently did tests using different inlet configurations found only marginal improvements to efficiency, and therefore validates the assumption.
- Slug flow occurred throughout the pump tube. This method found good correlation with existing empirical data, but the equation, predicting the velocity of the rising bubbles becomes invalid when evaluating small bore tubes (<12.5mm).

This was followed by the work of Stenning and Martin [22]. Who developed a simple prediction model for use where the lift-tube is relatively short, up to 10m. This model is able to predict pump characteristics successfully and allows for theoretical proof of previous guesswork designs. This model is used by Abed [1] for optimising a wind driven air-lift pump, and the model has been adapted for use within a bubble pump system by Delano [7] and Schaefer [17], in their work on the Einstein refrigeration cycle. They found good correlation between the predictions and the tests performed using small diameter tubes. By this model they were able to adjust the lift-tube pump to the perfect tube diameter and achieve a marginally better pumping efficiency.

Whalley and Butterworth [26] presented a paper on a homogeneous model of two-phase flow that can be used to predict the circulation rates in a vertical thermosyphon reboiler. Their concept was to have the mathematics simple enough to be done by hand calculation. The results attained have reasonable agreement with experiments and other iterative methods. Jurng and Park [11] have subsequently used this model to predict the flow in an improved boiler pump. The pump was designed for a Li/Br water refrigeration system and was built to use between11-23 kW of heating produced by a gas flame. The results were good and added credit to the method for use on a binary system.

Clark and Dabolt [5] proposed a general design equation that would be able to account for all air-lift pump design needs. This equation works best in the region where the pump works most efficiently. They also conducted some experimental work using different liquids, and took some comparisons to existing theories, with favourable results. Their equation does not have a height limitation, unlike the Stenning and Martin method and can be used for flow predictions of a lift-tube of any length.

Pfaff et al [16] modelled a bubble pump using a manometer principal and intermittent slug flow regime. This method looks at the pressure difference within the pump, a result of the difference in the densities of the fluids in the inlet and lift-tubes. The inlet tube is filled with liquid and the lift-tube filled with the less dense gas/liquid mixture. From observations they noted that the pump only worked intermittently, and built a time factor into their model. Stenning and Martin [22] and Lister [14] also noted the intermittent or oscillating flow, but did not take it into account. This is the weakness of the Pfaff et al model; the cycle times need to be known or estimated to proceed with the calculations and get predictions. This model is acceptable in that it follows the trends from their experimental work, although predicting about 20 % higher flow rates throughout.

2.2.6) Experimental observations of bubble pumps

Early bubble pump and air-lift-tubes have been designed by trial and error methods, and numerous observations and principles in their operation give rise to empirical models [5,15,22]. This section will look at some of the observations and findings that have been made and used.

2.2.6.1) Effect of the bubble pumps dimensions on its performance

It has been shown through experimental and theoretical considerations that the diameter of the pumping tube has very little effect on the pumping efficiency, provided the pump is operating in the slug or churn flow regimes, see figure 2.7 [14]. Bubble pumps that have small diameter lift-tubes are affected by the surface tension of the fluid, and this can become an important factor in the pump's design, for example in the surface tension pump mentioned in 2.2.3. The results found by Lister in figure 2.7 [14], show no change in the initial slope of the power input vs. liquid volume pumped curve for a variety of different diameter lift-tubes. With the smallest being 6mm, which indicates that the effect of surface tension is negligible for tube diameters as small as 6mm.

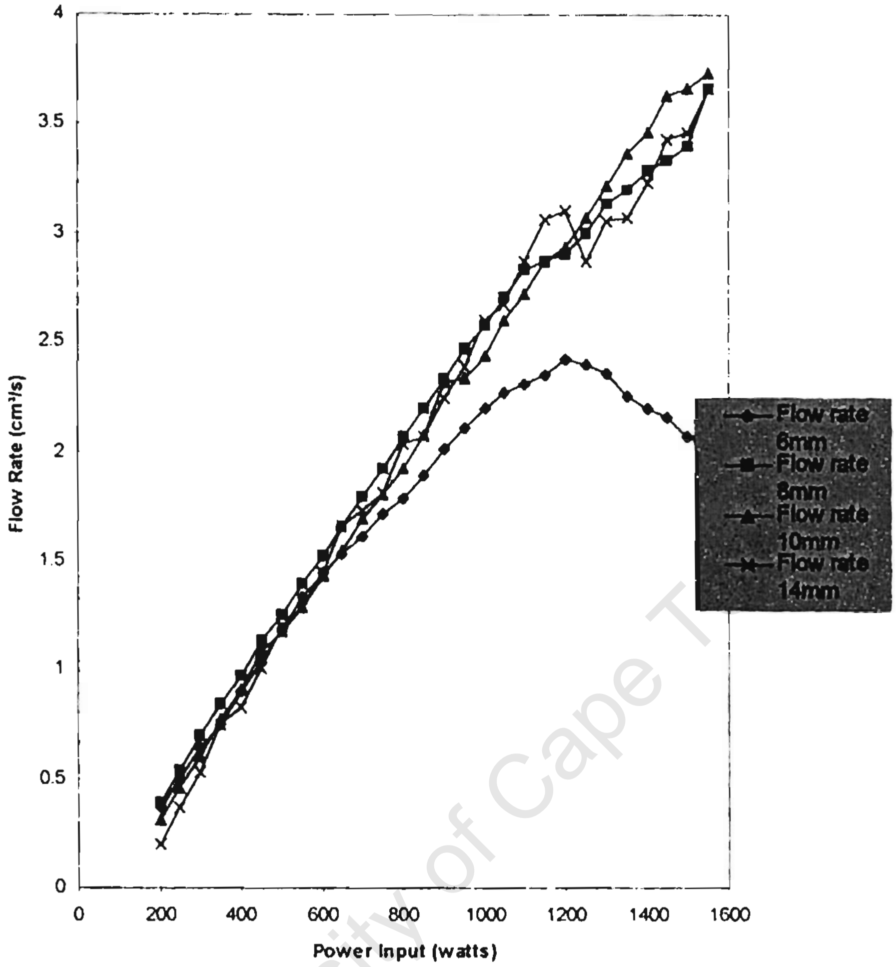

Figure 2.7) Graph showing pumped flow rates for different lift-tube diameters [14]

The pumping efficiency is however affected by the submergence ratio [10,14,15,22,27], which is the ratio of the pump's lift height to the driving head of the fluid in the lift-tube. The bulk of the experiments found in literature were conducted on gas injection systems. When these results are compared to experiments conducted using water as the liquid medium and the production of steam due to a heating the results show no difference, and therefore it is assumed that the same will hold true for a binary liquid and its vapour, such as the ammonia/water mixture found in the absorption refrigeration unit.

The pumping efficiency can be regarded as a measure of the volume of gas needed to pump a certain volume of liquid. This will have a direct correlation to the supplied energy to create the gas flow into the liquid medium, regardless whether it is gas injection or vaporisation of the liquid due to heating.

It is observed in figure 2.7 [14] that the efficiency of the pump will eventually decrease as the volume of vapour released is increased. This occurrence is observed for the 6mm-diameter tube first. The reason for this decrease is the small area of the tube and the large volume flow rate of the gas. This forces the flow pattern to change from the slug/churn flow to the annular flow regime, allowing the larger gas volume to pass through, which results in a reduction in the liquid pumping efficiency.

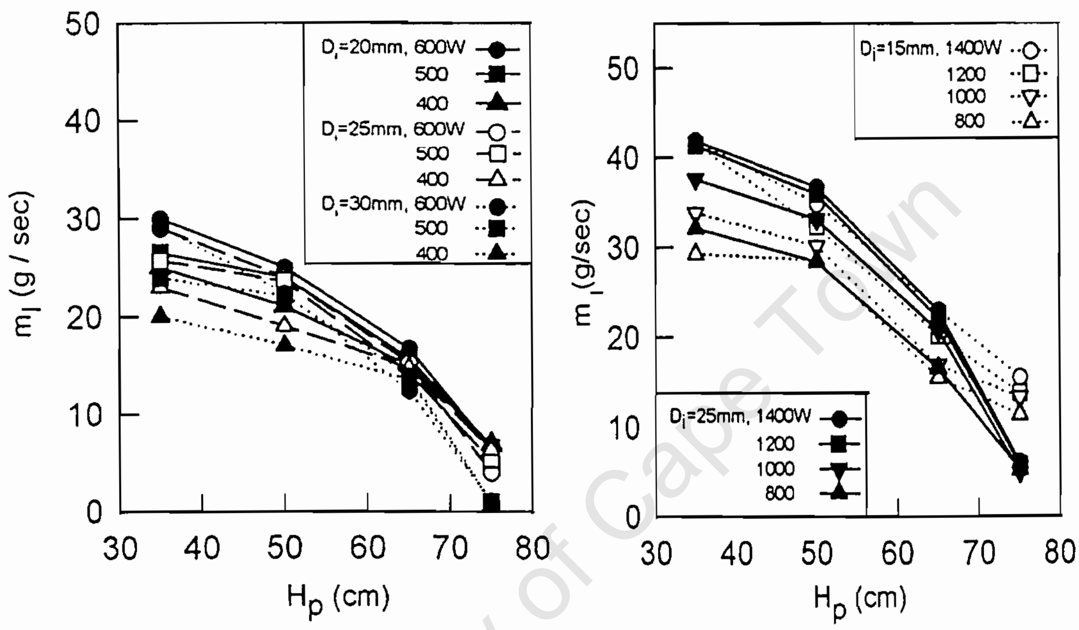

Figure 2.8) Effect of pumping height on the liquid flow rate. Left, for a water system. Right, for a Li/Br system.

There is a maximum tube diameter above which slug flow will not occur. Equation 2.1 predicts this behaviour, with σ being the surface tension and ρ_v and ρ_f being the density of the gas and fluid respectively [4]. This indicates a restriction on the use of larger diameter lift-tubes to deal with higher gas volume flow rates in terms of retaining pumping efficiency.

$$d = 19\left(\frac{\frac{\sigma}{\rho_f}}{g\left(1 - \frac{\rho_v}{\rho_f}\right)}\right) \qquad (2.1)$$

It was found that after a specific pipe diameter has been exceeded there is a change in the flow pattern from slug to an intermittent churn type flow [10,13,16]. It was also found that after a still larger tube diameter, pumping stopped altogether. Pfaff et al [16] found this diameter to be 18mm for their particular set-up.

Jeong et al [10] found similar type of phenomena and they called it a "discharge limit", see figure 2.8. They found that after a certain pumping height is exceeded the pumping stopped, and noted that as the diameter increases the maximum pumping height decreases, despite the amount of heat, or gas that is added. They have put forward that there could be a cut-off ratio between pipe diameter, pump lift and the amount of gas processed that will predict if a pump will work. Their observations also show that water and binary systems have very similar responses.

2.2.6.2) Effect of heat inputs on a bubble pumps performance

Other basic observations found that an increase in heat input increases the amount of vapour released, which then increases the liquid volume pumped and decreases the size of the slugs [10,14,16]. It was observed that the amount of vapour released is due mostly to the heat input and is not significantly affected by the pumps lift height, as seen in figure 2.9 [10]. This only holds true if the pump is physically able to cope with the increases in the gas/vapour flow rates, which gives a limiting factor of heat input versus liquid flow rate in the slug flow regime.

Form the literature it is evident that bubble pumps appear to be quite robust. They are easily adaptable to many situations, although accurately predicting their performance has not been as easy. With the development of the mathematical model studied and described in this thesis it will be possible to design a bubble lift-tube pump for diffusion absorption machines and predict not only the pump's performance but also the overall performance of the refrigeration unit.

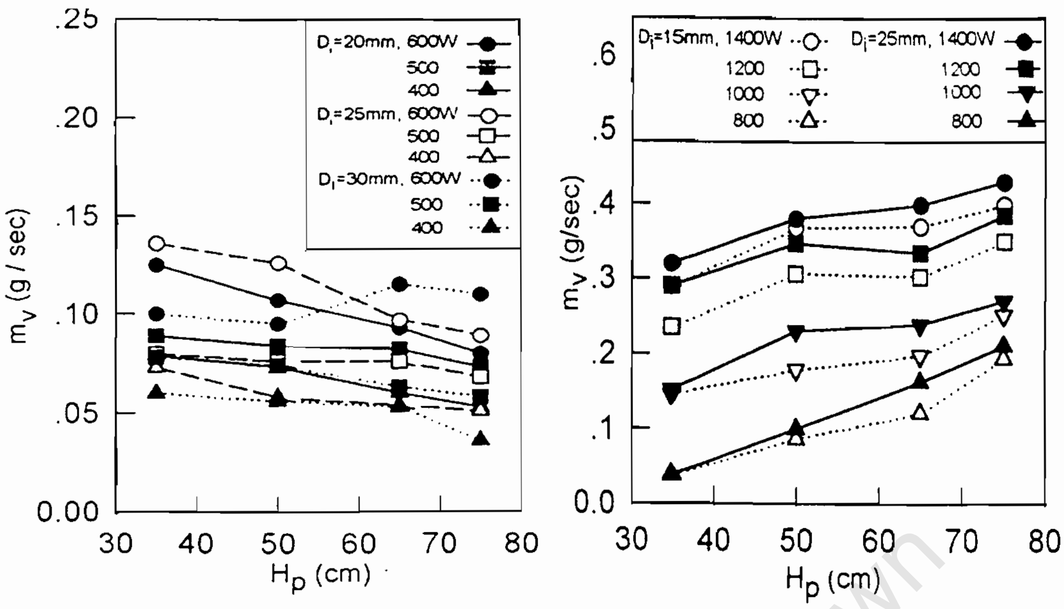

Figure 2.9) Effects of pumping height (H_p) on vapour release (m_v). Left, for a water system. Right, for a Li/Br binary system.

However, the latter is not implemented as this work focuses on the bubble pump alone. There are methods available that predict the COP's of absorption machines in general, such as Vicatos [23], assuming that the pump can deliver that desired solution flow rates. This alone has been the sole and major limiting factor on the capacity and size of the diffusion machines. With the proposed design modification, incorporating the multi tubular lift pump this limitation is eliminated.

Chapter 3

The Apparatus

Two pieces of apparatus were built and adapted for use with a multiple lift-tube bubble pump. This chapter documents the design criteria and observations that were desired from each piece of equipment. Included are a number of design problems and solutions that arose after the apparatus had been built and the initial testing had begun.

3.1) Justification of the multiple lift-tube bubble pump.

The main focus of this project is to increase the capacity of diffusion absorption refrigerators. To achieve this it is necessary to increase the amount of refrigerant produced by the boiler, which necessitates an increase in supplied heat. The first result of the increased heat input is the necessity to redesign the system as a whole, in order for all the components to accommodate the larger heat transfers and resulting increases in solution flow rates [19].

Looking at the bubble pump specifically, the increased heat input changes the bubble pump's task in the following ways:

- The pump will need to accommodate an increase in heat input.
- Will need to facilitate the higher vapour and liquid flow rates.
- The pump will need to pump to a higher head, because of the necessary increased size of the system to accommodate the increased heat inputs.

From the observations and data given in chapter 2, there are some physical restrictions to the workings of a bubble pump, with regards to the increase in vapour and liquid flow rates and pumping head. Firstly, it must be remembered that there is no major change in liquid or vapour flow rates for different diameter lift-tubes, given the same heat energy input. The larger flow rates do, however, require larger diameter lift-tubes, because smaller diameter lift-tubes have a limit to the vapour flow rate that can be used before the flow pattern within the lift-tube changes from slug to annular flow, and the liquid flow rate begins to reduce, see fig 2.7. The larger diameter tube has limits that place a restriction on the maximum lift-tube diameter and maximum pumping head. This meaning, that to pump to a higher head, a narrower diameter tube is needed, but as already mentioned, a narrow lift-tube is unable to handle the larger flows. This leaves only one solution, the use of a number of lift-tubes.

The remainder of this dissertation will examine the use of a multiple lift-tube bubble pump, concentrating specifically on its ability to increase the flow rates, and cope with larger heat inputs. Attention is also paid to how this will affect the diffusion absorption refrigeration plant.

3.2) Multiple lift-tube bubble pump: water experimental apparatus

The water rig was designed to test for the effects on pumped volume flow rate by varying the number of lift-tubes used. Water was used as the working fluid, in order to gain insight and confidence that this type of pump would work as desired. The design was also required to have similar dimensions to an earlier rig, which was designed and used by Lister [14], which would allow the use of data collected for comparative purposes.

3.2.1) Reasons for testing with water

Water has a number of advantages that make it a good test fluid. They are:

- Water is non-toxic.
- Water is freely available.
- The properties of water are well known and lend themselves to calculations.
- In the literature the trend is to use water as the initial testing fluid, thus giving results that are more easily compared to data found in previous experiments [5,6,7,10,12,14,15,16,21,22].

- Water has similar properties to that of ammonia, which can be seen in the table in Table 3.1.

Table for the comparison of water and ammonia properties				
Liquids				
	Water		Ammonia	
	Atmospheric	Atmospheric	8 bar	Pressure
Temperature	25	100	25	Deg C
Density	997	955.1	602	kg/m^3
Heat capacity	4184	4216	4800	J/kg.K
Dynamic viscosity	1.00E-06	1.67E-04	3.59E-07	m^2/s
k (conductivity)	0.604	0.684	0.521	W/m.K

Gas Properties (100°C, atm)			
	Steam	Ammonia	
Density	0.586	0.559	kg/m^3
Heat capacity	2.06	2.236	kJ/kg.K
Viscosity	1.27E+07	1.29E-05	kg/m.s
k (conductivity)	0.0246	0.0327	W/m.K

Table 3.1) Comparison some properties of water and ammonia [3,9]

The major difference between pure and binary systems is the *temperature glide* of the boiling point in binary systems. The binary system is different from a pure liquid, where the liquid boils at a set temperature, within a constant pressure environment. In a binary mixture, one component will be more volatile than the other, and will vapourise faster. The faster vapourisation of the one component results in the reduction in its concentration in the solution. This changes the boiling temperature of the remaining solution. For the binary system used in this project the boiling temperature is increased by the decreased refrigerant concentration in solution. It is believed that this effect will allow for a smoother pumping action, because the vapour is released over a wider temperature range, thereby making pumping more independent of constant pressure and temperature within the boiler than with a single fluid. Therefore, by testing with water, the multiple lift-tube bubble pump is undergoing a more difficult test, assuring that if it works with water it will work with a binary system.

3.2.2) Water rig design considerations and features

There are a number of different aspects that had to be considered when designing and manufacturing the test rig. Four broad points were considered during the design of the water rig and are as follows.

- Control and safe working at elevated temperature and pressure
- Heating of the fluid
- Control of fluid levels
- Basic layout

Control and monitoring of each of these points bears directly to the quality of the attained results. Figures 3.1 and 3.2 are referred to in the following sections.

3.2.2.1) Control and safe working at elevated temperature and pressure

It was desirable to test the bubble pump with different internal pressures, in order find a trend of how increased pressure would affect the pumping efficiency. The rig was designed to withstand a maximum working pressure of 6 bar safely, as this would allow a wide enough pressure range to establish any trends. Appendix A demonstrates the design calculations of minimum wall thickness allowed for the different tube diameters that would make up the system. It was found that commonly available seamless steel, copper and glass tubes would be more than sufficient for the design. The valves and fittings used were also specified to withstand pressures well above what they would encounter during testing.

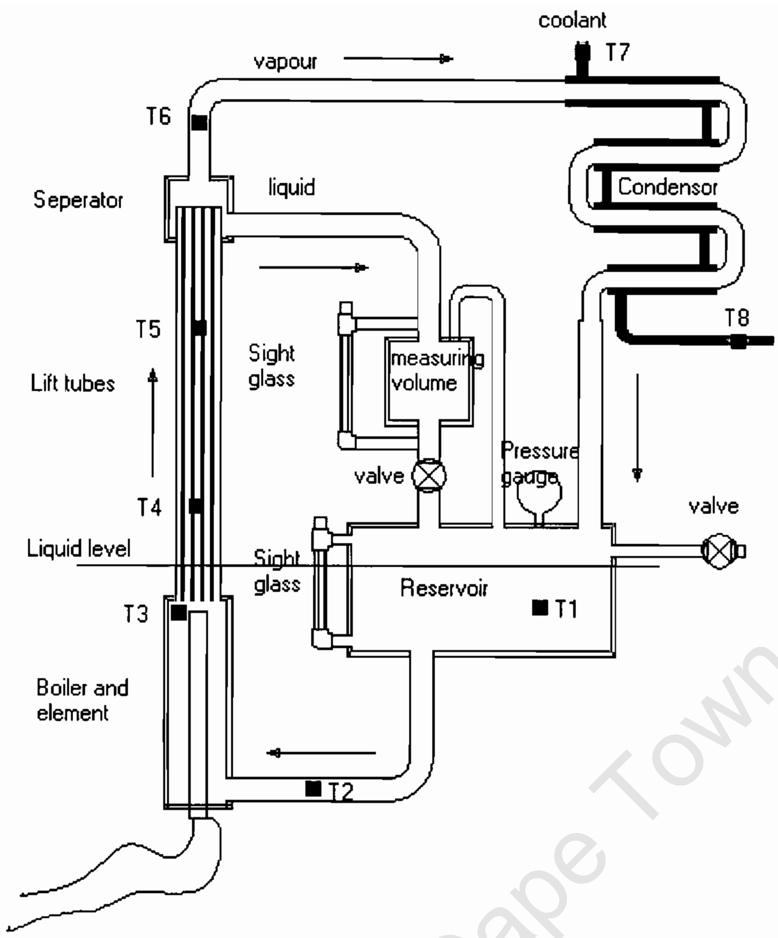

Figure 3.1) Diagram of the water rig showing the liquid level and the thermocouple points.

During operation, the system is heated and vapour is released. This results in a pressure increase, because the rig is a sealed unit. In order to minimise any thermal oscillations two additional components were implemented:

- A constant pressure *valve* that allows the pressure to be kept below a specific maximum relative to atmospheric pressure by allowing some vapour to escape if the maximum pressure is exceeded, very similar in action to a safety relief valve.
- A condenser, of the shell and tube type heat exchanger. It allows the vapour to condense, and return to the reservoir, maintaining a constant pressure. The

cooling necessary to condense the vapour water is provided by piping cool water through the outer shell of the heat exchanger.

3.2.2.2) Heating of the fluid

The heating to the boiler is provided by using an electrical cartridge element. The energy input is controlled using a variable transformer. The electrical cartridge is placed inside the boiler and comes into direct contact with the fluid and therefore maximises the heat transfer. The maximum power rating of the cartridge was chosen as 1.2kW, because it could exceed the power required to allow the bubble pumps flow regime to change from bubble flow to annular type flow, if a single lift-tube were being used, as seen in figure 2.7.

To increase the range and efficiency of the system the water heated in the boiler was not cooled before being returned to the reservoir. This increased the testing range by using recovered heat from the ongoing heat input in the boiler. To further increase the efficiency, the whole system was insulated with glass.

Thermocouples are installed at various points on the system, see figure 3.1. Eight channels were used with a Pico TC-08 data logger, which allows for quick and easy real time monitoring of temperatures as heat inputs are changed.

3.2.2.3) Fluids and control of fluid levels

The water charged in this system had an additive of potassium dichromate and sodium hydroxide, which was also used in the subsequent absorption diffusion refrigeration plant [28]. The potassium dichromate crystals used are approximately 1.5% the weight of water and acts as a rust inhibitor, it is however acidic and therefore needs to be neutralised by the addition of sodium hydroxide. Sodium hydroxide is added until the mixture changes colour from orange to completely yellow giving a visual indication of the neutrality of the substance. The colour of the water has the added advantage of making the water levels easily visible in the sight glasses.

The submergence ratio is a critical variable in the pumping performance of the bubble pump, and needed to be kept as constant as possible to give reliable test results. For this reason a large reservoir of 5 lt. maximum capacity and a measured volume of 0.5 lt. was chosen, that allowed only small changes to this ratio. During each flow measurement the submergence ratio moved between 0.16-0.14 with an average of 0.15, which was the desired ratio. The boiler itself will also have an effect on pumping, because it is a tube. The boiler tube diameter, thought, is very large when compared to the lift-tubes and it will not work in the slug flow regime. The vapour liquid mixture within it does however increase the effective submergence ratio, by lowering the density of the fluid within the boiler thereby increasing the initial head in the lift-tube.

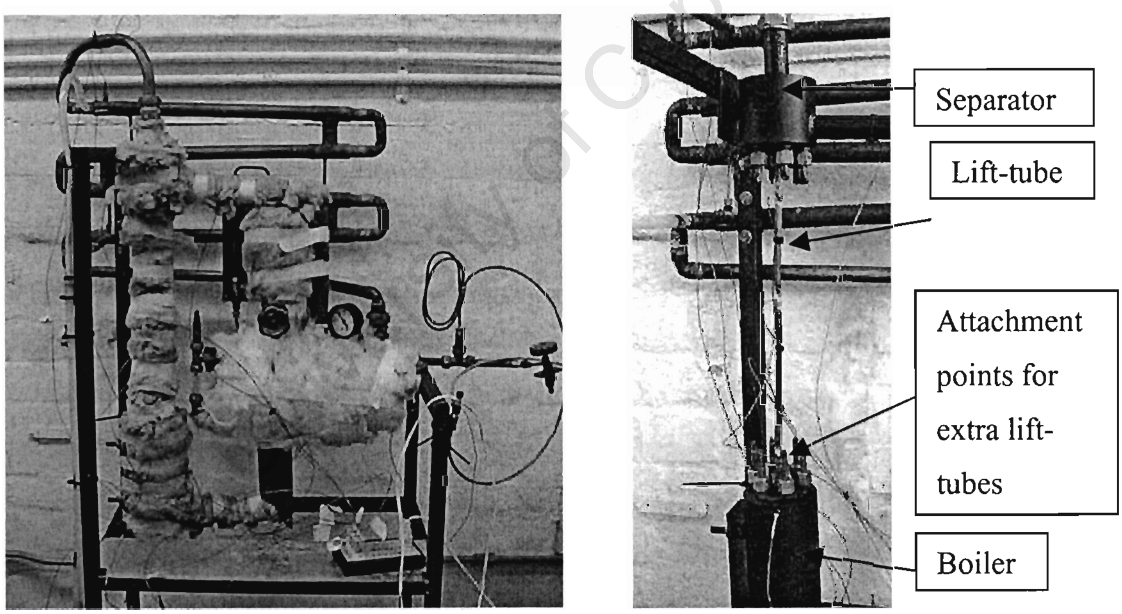

Figure 3.2) Pictures of the water rig. Left, the whole rig clad in glass wool. Right, a close up of the boiler, lift-tube and separator showing positions for additional lift-tubes.

In order to determine the liquid flow rate a vessel of known volume was used to trap and measure the liquid pumped. This measuring volume was made large enough to minimise volume fluctuations thus increasing the accuracy of the flow rate measurements. A sight glass was used to visually assess the liquid level within the measuring vessel and also ascertain the timing of the vessel's filling period, see figure 3.1.

3.2.2.4) Layout of the water rig

The water rig was made to test the bubble pump performance and assess the possibility of using a multiple lift-tube bubble pump on a diffusion absorption refrigerator. The rig had to therefore have a boiler, lift-tube and separator that are the same as those to be used on the refrigerator. Figures 3.1 and 3.2 are pictures and schematics for the rig that was built and used. The bubble pump used in the rig was designed to work in the same way as in the refrigerator. By applying heat directly to the liquid in the boiler, vapour, pushes the liquid up the lift-tube and into the separator, where the vapour and liquid are separated. Thereafter, to complete the cycle, the vapour is condensed into a liquid state and then returned to the reservoir.

The rig's design allowed for the addition of extra lift-tubes as can be seen in figure 3.2. They could be added as desired by removing the plugs and inserting additional lift-tubes. This method however, proved to be difficult to use. To make the addition and subtraction of lift-tubes easier, ball valves were added into the permanently fitted tubes. The addition of a ball valve into a lift-tube did not result in any major discontinuities within the tubes, because they were chosen to have the same internal diameters as the lift-tubes, and therefore their effect on system function, if compared to a continuous tube, is believed to be minimal.

3.2.3) Initial rig problem

After the first tests were completed, it was noticed that there was a large amount of temperature and flow rate oscillations during pumping. The pattern can be seen in Figure 3.3, which gives a temperature vs. time response of the system. It can be observed that when the boiler temperature reaches 100 °C, the components after the boiler begin heating up, indicating that pumping is occurring. Then a drop in the inlet temperature begins, followed by a drop in the temperature of the rest of the components, indicating that pumping had ceased and the components were then loosing heat to the environment.

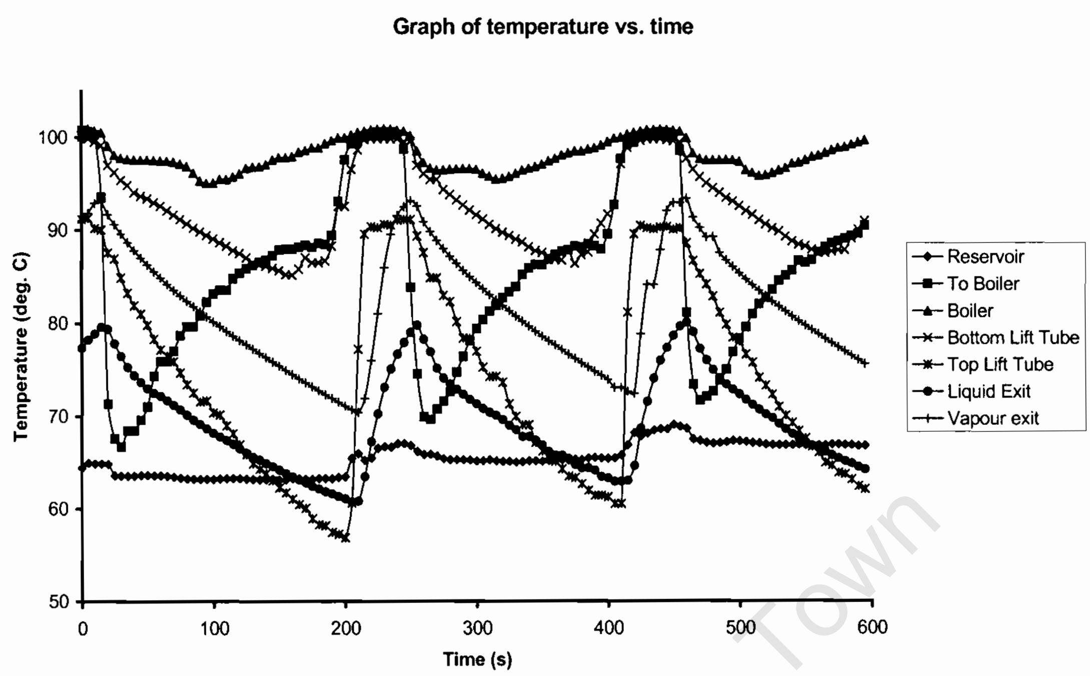

Figure 3.3) The water system's temperature response before the boiler was adjusted.

The reason for this was believed to be that the boiler chamber was too large, and thus allowed the cooler water entering the boiler from the reservoir to mix too easily with the hot water, thus causing a drop in temperature within the boiler. Two design upgrades were used to prevent this and obtain a steadier pumping action.

- The first was to partially fill the boiler with glass beads and reduce the volume of liquid held in the boiler.

- The second method was to direct the cooler liquid entering the boiler down to the bottom of the boiler, where it could begin heating up without disrupting the hotter fluid above it.

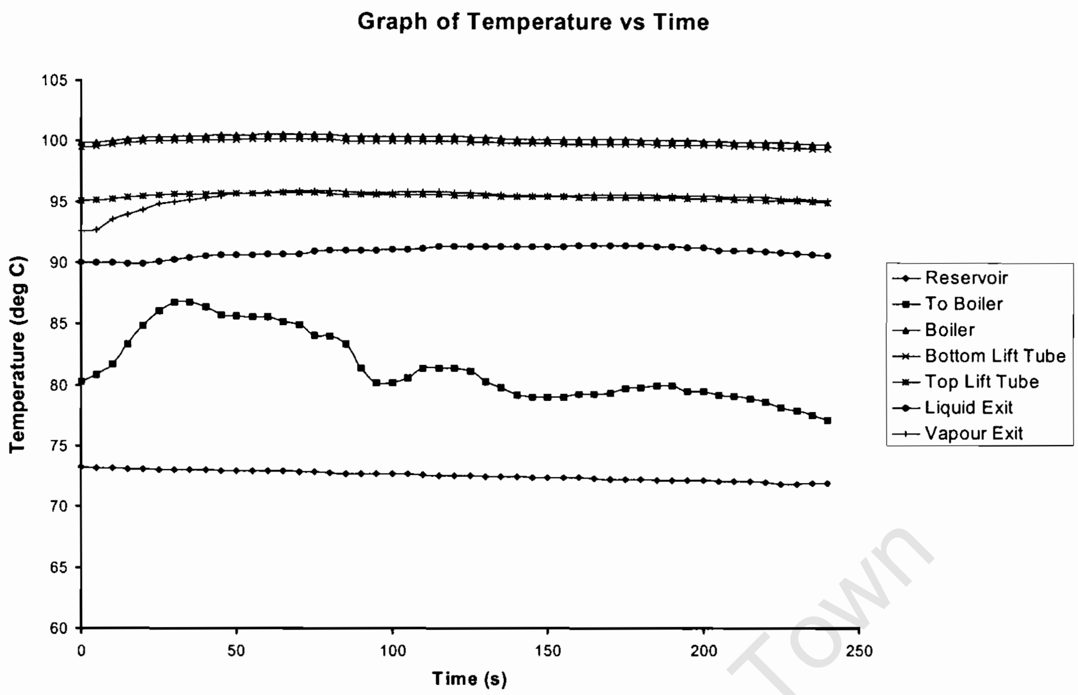

Figure 3.4) Temperature response of the system after adjustments had been made

The result of these adjustments can be seen in figure 3.4, which shows that almost all of the cyclic activity has been eliminated. The boiler now holds its temperature along with the other components in the system, indicating that pumping is regular and continuous.

3.3) Diffusion absorption refrigeration plant

The refrigeration plant used was an existing refrigerator made by a refrigeration company called Zero, and was designed to use a 25% ammonia 75% water solution by mass. The use of an existing system reduced the manufacturing time of an experimental rig. The changes made to the system were minimal and are described in this section.

3.3.1) Safety considerations

The safety considerations that must kept in mind when using the refrigerator are: that hydrogen is extremely flammable; and that ammonia is poisonous, flammable at specific concentrations with oxygen, and is corrosive to copper and its alloys. With these properties in mind, care was taken when working with this refrigeration plant to avoid hazards and/or injury.

3.3.2) Modifications to existing absorption refrigeration plant

The following additions were to the plant.

- A new multiple lift-tube bubble pump.
- A heater to act as a load for the evaporator.
- A pressure gauge.
- Filling and discharge valves and level indicator.
- Thermocouples at specific points.

3.3.2.1) Multiple lift-tube bubble pump

The major modification to the system was the use of a multiple lift-tube bubble pump. The existing bubble pump system was replaced with the boiler, lift-tubes and separator used for the water rig. The modification was simple and involved cutting off the original bubble pump and welding the new bubble pump in place. It also involved the fitting of a new down-comer tube, to bring the pumped liquid down from the separator to the entrance of the liquid heat exchanger.

3.3.2.2) Heater for the evaporator

In order to test the system, a controllable and measurable load is applied to the evaporator, which will enable the efficiency and effects of the added lift-tubes to be calculated and studied. A coiled copper tube was soldered on the evaporator coils to form a heat exchanger. Heat could then be added to the evaporator by pumping a liquid through the copper pipe. The whole evaporator was then insulated with polyurethane foam. Thus, by changing the flow rate of the liquid, the temperature of the evaporator could be kept within a desired range. The applied load can then be calculated from the temperature drop between inlet and outlet of the warming fluid.

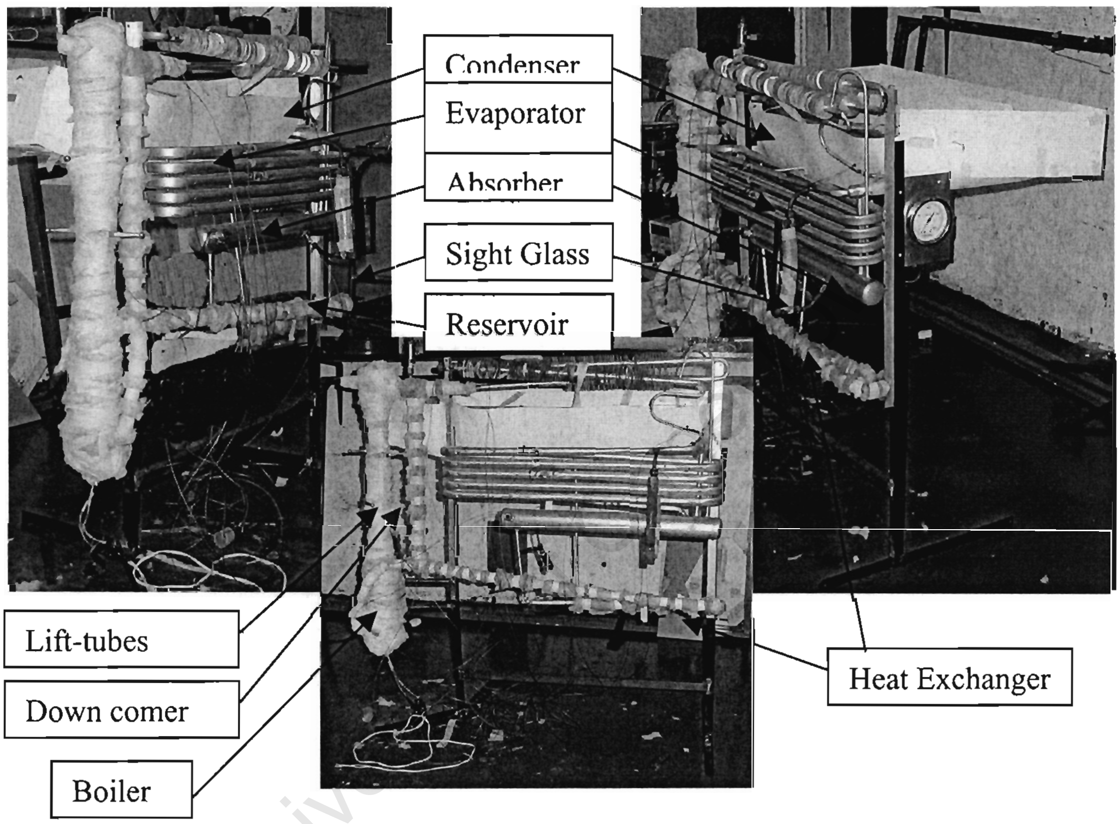

Figure 3.5) Pictures of the actual diffusion absorption refrigerator.

3.3.2.3) Pressure measurement

The pressure within the rig was measured with a bourdon tube pressure gauge, with a 2% accuracy rating. It was attached to the reservoir, although the pressure could have been measured elsewhere in the system with little change to the pressure reading.

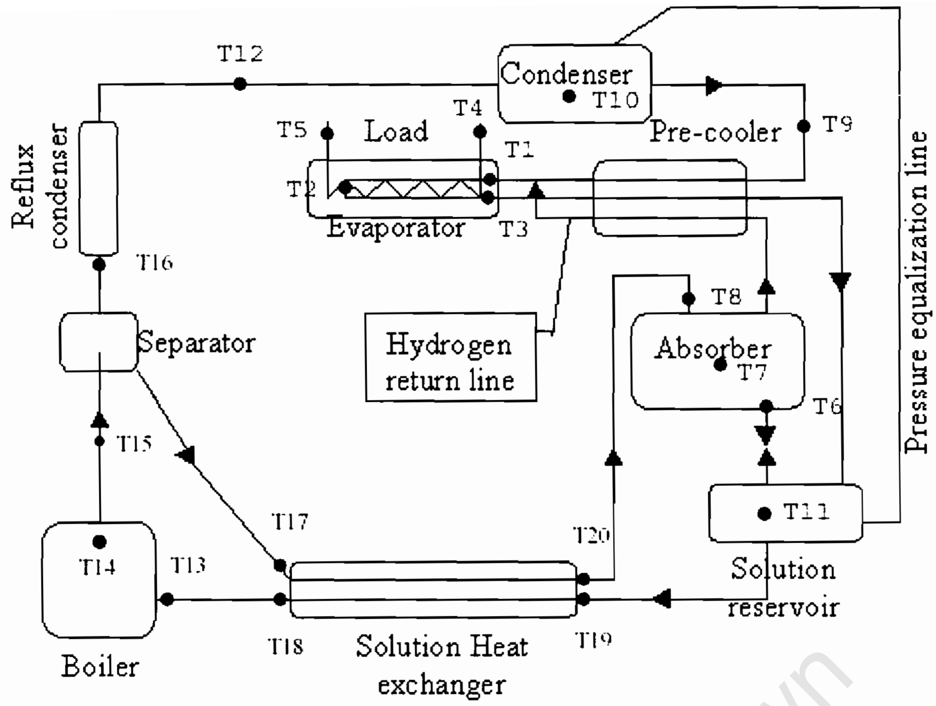

Figure 3.6) Schematic of diffusion absorption plant showing the locations of the thermocouples.

3.3.2.4) Charging and discharging

To charge and discharge the rig a number of valves were added to different areas. These were added to areas that would enable the total draining of liquids from the system. One was also added to the highest point in the system, where mostly hydrogen gas would collect, and it thereby facilitated its easy addition or release.

To control the level of fluid within the system, a sight glass was fitted to the solution reservoir, which enabled the level of the strong solution within it to be ascertained.

3.3.2.5) Temperature measurement

Thermocouples were installed to measure the temperature at points of interest throughout the rig, see figure 3.6. The number of points was governed by the usefulness of the attained readings, which transpired to be 20 points to give a full picture of the system's workings. Two temperature data loggers were used in tandem, one providing 12 channels (Yokogawa data recorder) and the other providing 8 channels (Pico TC-08). J type (iron vs. Constantan) thermocouples were used and the thermocouples were silver-soldered directly onto the tubes.

3.3.2) External modification to the diffusion absorption rig

It was found that at larger heat inputs to the system, the absorber quickly became inadequate to dissipate its heat load. Its temperature would rise to levels that would retard the absorption process, thereby stopping the effective working of the refrigerator. During initial testing when the system was under original design heat load, it was found that the exit wall temperature of the absorber was 40°C. This meant that to allow experiments to be carried out at higher heat inputs, the absorber would need additional cooling. The additional cooling was achieved using a mobile fan to create a forced convection over the absorber and increase the heat transfer to the environment.

3.3.3) Charging method

To effectively charge the system a number of items need to be known.

- The concentration of the charge solution. The manufacturer recommends a solution of 25% ammonia and 75% water by mass for this refrigerator.
- The volume of liquid needed to allow the system to work effectively and at the desired submergence ratio. This would be done by filling the system with water to the desired level, and then draining the water and measuring the volume. It was then possible to calculate the masses of water and ammonia to be added to the system, by knowing mass concentration. The masses were calculated to be 1.3kg of water and 0.433kg ammonia.
- Final system pressure, after hydrogen is added.

The procedure is as follows:

Step 1: The system was evacuated to remove the air. It was then pressurised with hydrogen and re-evacuated.

Step 2: The distilled water was then added to the system while it was boiling, in order to remove as much dissolved air as possible. Potassium dichromate and sodium hydroxide crystals were added to the boiling water in the same ratios as used for the water system. A weighed mass of the mixture was transferred into the evacuated system.

Step 3: Ammonia was transferred into the refrigerator from a continuously weighed ammonia cylinder. The transfer was stopped once the desired weight reduction in the ammonia cylinder had occurred. As a test to the correct ammonia/water ratio the system was allowed to reach ambient temperature, and sufficient time to allow the concentration of the ammonia/water solution to reach homogeneity. The system pressure was 0.5 bar absolute at 20°C, which confirmed the 25% ammonia concentration.

Step 4: The remaining system pressure came from the hydrogen that was added to lift the system up to the desired working pressure, which in this case was 20 bar abs. With the charging complete the system was ready for testing.

Chapter 4

Bubble pump mathematical model

The bubble pump has been modelled using the analytical air-lift pump model of Stenning and Martin [22]. This model assumes that the pump operates in the slug flow regime. The air-lift pump equation has been adapted to allow its use for a multiple lift-tube bubble pump, which is followed by the density and energy calculations that allow a comparison to be made between the model and rig data.

4.1) Modelled solution

4.1.1) Analytical solution

The equation given below describes the behaviour of an air-lift pump working in the slug flow regime and concentrates only on the lift-tubes. It uses one-dimensional continuity and momentum equations, and the results of two phase flow experiments to find the solution.

$$\frac{H}{L}-\left(\frac{1}{1+\dfrac{\dot{V}_g}{s\times\dot{V}_f}}\right)=\frac{V_1^2}{2\times g\times L}\times\left((K+1)+(K+2)\times\frac{\dot{V}_g}{\dot{V}_f}\right) \tag{4.1}$$

Equation 4.1 gives the dependence of the submergence ratio (H/L) on the pump's dimensions, and properties of the fluid in the bubble pump. It is evaluated by Microsoft Excel 2000's goal seek function, which adjusted the volume rates of vapour and liquid until the desired submergence ratio was obtained and thereby predicted the flows in the experiment.

For n number of lift-tubes, the factor n is introduced into equation 4.1, which becomes,

$$\frac{(H+\Delta h)}{L}-\left(\frac{1}{1+\frac{\dot{V}_g}{s\times\dot{V}_f}}\right)=\frac{\left(\frac{\dot{V}_f^2}{n\times(A_l^{\ 2})}\right)}{2\times g\times L}\times\left((K+1)+(K+2)\times\frac{\dot{V}_g}{\dot{V}_f}\right) \tag{4.2}$$

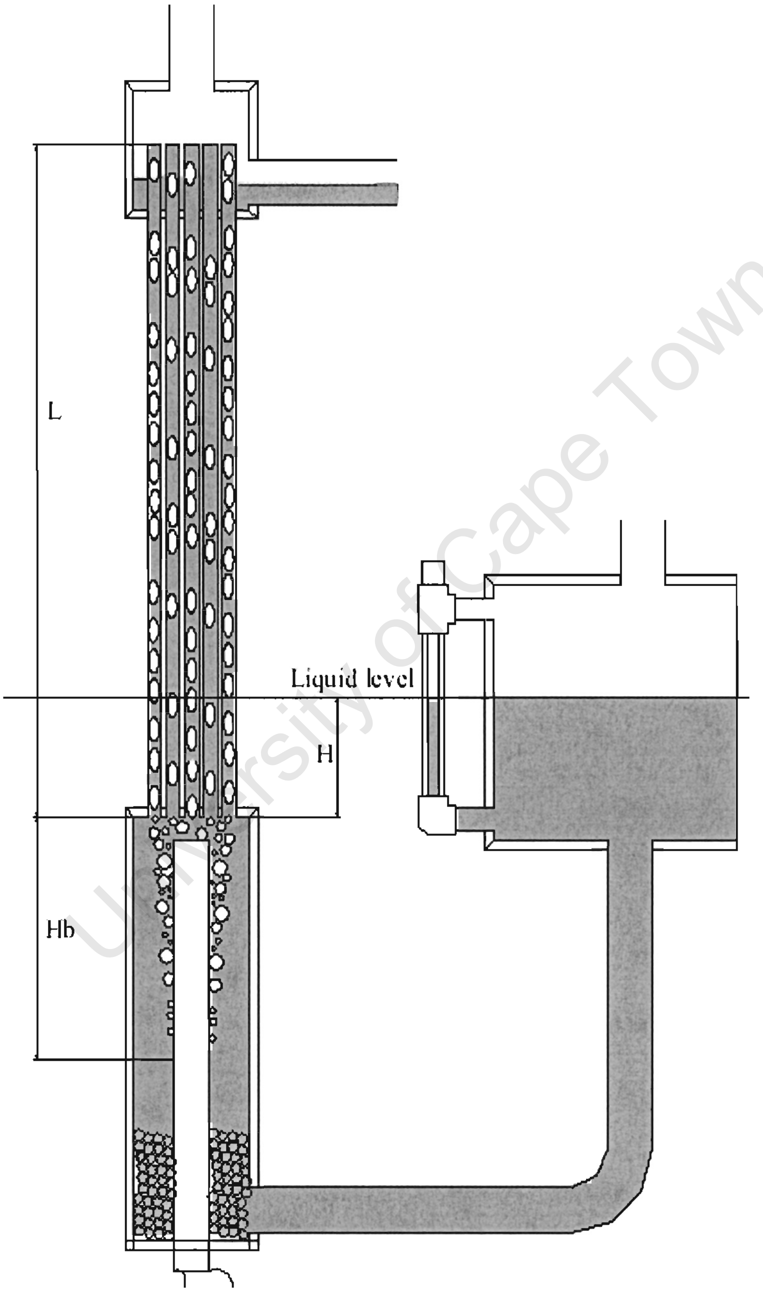

Figure 4.1) Schematic of the bubble pump dimensions used for the model

Where s is the slip ratio defined as,

$$s = \frac{V_g}{V_f} \tag{4.3}$$

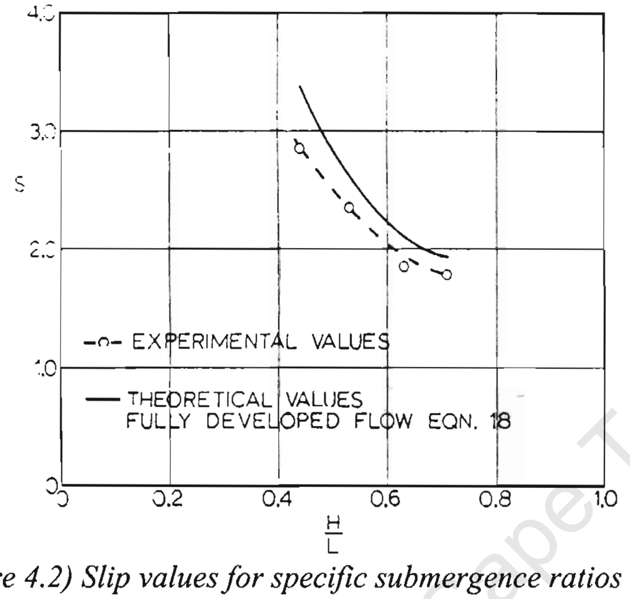

Figure 4.2) Slip values for specific submergence ratios [22]

It has been suggested that for optimal performance, the slip ratio s for a bubble pump be between 1.5 and 2.5 [7]. However, for the smaller submergence ratio that is used in this case, figure 4.2 indicates a value that is much higher; by extrapolation it shows s to be in the region of 4 or 5 for submergence ratios between 0.1 and 0.2. This indicates that less liquid will be pumped for a given gas volume flow rate. The slip ratio can also be found using the following correlation [17],

$$s = 1.2 + 0.2 \times \frac{V_g}{V_f} + \frac{0.35 \times A \times \sqrt{g \times D}}{V_f} \tag{4.4}$$

The factor K, which is the loss coefficient and accounts for the losses on the pump, is also affected by the number of lift-tubes. These losses can be entrance, exit, elbow and friction losses etc. According to Delano [7], the factor K in the equation 4.2 can be determined experimentally and is defined as,

$$K = \frac{4 \times f \times L}{n \times d} \tag{4.5}$$

The friction factor f for two phase flow is given as [17],

$$f = \frac{0.316}{\mathrm{Re}_d^{\ 0.25}} \tag{4.6}$$

With the Reynolds number for the tube given as,

$$\mathrm{Re}_d = \frac{4 \times \rho \times \left(\dot{V}_g + \dot{V}_f\right)}{n \times \pi \times \mu \times d} \tag{4.7}$$

Equations 4.5-4.7 indicate that the loss factor K will not remain constant over the flow range of the pump. Experiments and analysis have shown that the K value does not vary significantly for a H/L ratio [22]. Therefore by using a constant value for K, there will not be a significant affect on the results.

4.1.2) Gasses effect in the boiler

The bubble pump tested consists of a large diameter boiler and relatively thin lift-tubes. The solution in 4.1.2 is able to account only for the lift-tubes pumping effect. There is also a lift effect created by the vapour within the boiler, which is a result of the decreased density found in the boiler, because of the vapour liquid mixture.

Because the area of the boiler is larger than that required, allowing slug flow to occur, it is assumed that the generated bubbles will simply flow upward with little reaction between them and the liquid in the boiler. The effect of the decrease in the boiler fluid density can be seen by a "U tube" manometer in figure 4.3. The decreased density on the left side results in the liquid level on the left being higher than that on the right, due to the balance of pressures. This increase in the liquid level in the lift pump results in increasing the submergence ratio thereby increasing the liquid pumping capacity.

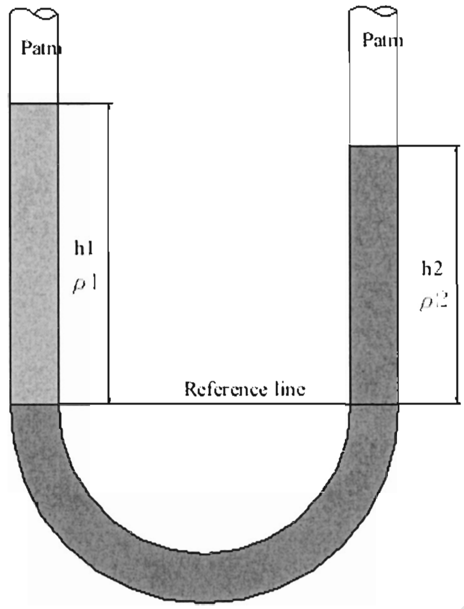

Figure 4.3) Diagram of a u-tube Manometer

For each arm of the u tube in figure 4.3

$$\rho_1 \times g \times h_1 = \rho_2 \times g \times h_2 \qquad (4,8)$$

which reworked to use the boilers terminology gives

$$\Delta h = h_b \times \left(1 - \frac{\rho_b}{\rho_f}\right) \qquad (4,9)$$

This difference in height is then added to the pump's initial head, used in equation 4.2, thereby accounting for the decreased density in the boiler.

From the vapour volume flow rate we know that,

$$A_{bg} = \frac{\dot{V}_g}{V_{bg}} \qquad (4.10)$$

Where A_{bg} is the superficial area the gas occupies within the boiler and V_{bg} is the gas rising velocity. This can be found from the following relationship [4],

$$V_{bg} = 1.53 \times \left(\frac{g \times \sigma \times (\rho_f - \rho_g)}{\rho_f^{\,2}}\right)^{\frac{1}{4}} \qquad (4.11)$$

The gas volume rate will also vary according to its position within the boiler, starting at the bottom with no gas and ending at the top at full flow rate before it enters the lift-tubes. A linear increase in gas volume is chosen to represent this, because the gas production is a function of the heat released from the resistance heater, which will have a regular heating rate over most of its heating surface, figure 4.1 graphically represents this.

If the gas volume increases linearly a cone of vapour/gas mixture will form about the heating element. Beginning with the volume of the cone,

$$Vol = \frac{\pi \times r^2 \times h}{3} \tag{4.12}$$

From equation 4.10 the radius is found to be,

$$r = \sqrt{\frac{\dot{V}_g}{V_{bg} \times \pi}} \tag{4.13}$$

The average density within the boiler can be expressed as

$$\rho_1 = \frac{Vol_g \times \rho_g + Vol_f \times \rho_f}{Vol_{total}} \tag{4.14}$$

And by combining equations 4.12, 4.13 and 4.14,

$$\rho_1 = \frac{\frac{\dot{V}_g \times h_b}{3 \times V_{bg} \times A_T} \times \rho_g + \left(A_T \times h_b - \frac{\dot{V}_g \times h_b}{3 \times V_{bg} \times A_T} \right) \times \rho_f}{A_T \times h_b} \tag{4.15}$$

Giving,

$$\rho_1 = \frac{\dot{V}_g \times \rho_g}{V_{bg} \times A_T \times 3} + \rho_f - \frac{\dot{V}_g \times \rho_f}{V_{bg} \times A_T \times 3} \tag{4.16}$$

To allow a solution to equation 4.9 to be found, a guessed vapour flow rate is used to find the average density within the boiler, equation. 4.16. The Δh value from equation 4.9 is then calculated and added to the design h resulting in a new submergence ratio that is used instead of the "at rest" submergence ratio in equation 4.2. The Δh is dependent on the volume rate of the vapour and therefore it will be updated to the correct value during the iteration process used to find the submergence ratio.

4.1.2) Modelling the heat input for the water rig

The bubble pump in a diffusion absorption refrigerator uses heat to produce the vapour that drives the system. This heat must be calculated in order to complete the prediction model.

The pressure in the system is directly linked with the density of the vapour and liquid in the pump, and boiling temperature. From the aforementioned properties, the property values needed to calculate the necessary heat input to produce the desired volume of vapour could be found.

There are three separate aspects of the energy added to produce the vapour:

- The energy needed to raise the liquid's temperature up to saturation temperature.
- The energy needed to boil the liquid to release vapour.
- The heat lost to the environment.

These three can be rewritten as:

$$Q_{total} = Q_{temp\_rise} + Q_{boil} + Q_{loss} \tag{4.17}$$

4.1.2.1) Energy needed for temperature rise

Equation 4.18 gives the energy needed to bring the liquid into boiling point.

$$Q_{temp\_rise} = \dot{m} \times Cp \times (T_{saturation} - T_{inlet}) \tag{4.18}$$

However, Cp for water is not constant, but varies with temperature. Its behaviour with respect to temperature is given in equation 4.19 [3]:

$$Cp = -1\times10^{-7}\times T^3 + 4\times10^{-5}\times T^2 - 0.0021\times T + 4.2145 \quad \text{(KJ/kg. K)} \tag{4.19}$$

Equations 4.18 and 4.19 are combined and integrated over a temperature range:

$$Q_{Temp-rise} = \dot{m}\times \int_{T_{inlet}}^{T_{saturation}} \left(-1\times10^{-7}\times T^3 + 4\times10^{-5}\times T^2 - 0.0021\times T + 4.2145\right)\Delta T \quad \text{(kW)} \tag{4.20}$$

Resulting in,

$$Q_{temp\_rise} = \dot{m} \times \left(\frac{-1\times10^{-7} \times T_{saturation}{}^{4}}{4} + \frac{4\times10^{-5} \times T_{saturation}{}^{3}}{3} - \frac{0.0021\times T_{saturation}{}^{2}}{2} + 4.2145\times T_{saturation} \right) \text{(kW)}$$
$$- \dot{m} \times \left(\frac{-1\times10^{-7} \times 20^{4}}{4} + \frac{4\times10^{-5} \times 20^{3}}{3} - \frac{0.0021\times 20^{2}}{2} + 4.2145\times 20 \right)$$

(4.21)

Equation 4.21 accounts for the heat energy needed to bring the all liquid to boiling point. $T_{saturation}$ is the saturation temperature or boiling temperature of the fluid at a known pressure. The inlet temperature (T_{inlet}) to the boiler was kept constant at 20 °C, to allow a comparison between the predicted and experimental data.

4.1.2.2) Energy needed to vaporise liquid

After the liquid has been brought to saturation, continued heating will produce boiling and this can be accounted for with the following equation:

$$Q_{boil} = \dot{m}_g \times h_{fg} \tag{4.22}$$

With m_g being the mass flow rate of the released vapour, and h_{fg} the enthalpy of vaporisation for the liquid.

4.1.2.3) Energy lost to the environment

The heat lost to the environment is to a large extent due to natural convection cooling from the surface of the boiler and the lift-tubes. Radiation heat loss will be insignificant because the surfaces are insulated and the external surfaces are at a relatively low temperature compared to the environment. Convection heat losses are given as [9],

$$Q_{conv} = h \times A \times (T_{surf} - T_{amb}) \tag{4.23}$$

The surface temperature can be found by finding the amount of heat that is conducted through the steel boiler wall and the outer insulation covering.

$$Q_{cond} = \frac{2\times\pi\times k\times L\times\left(T_{\text{int}} - T_{ext}\right)}{Ln\left(\frac{r_{ext}}{r_{\text{int}}}\right)} \tag{4.24}$$

Equation 4.24 is the basic equation for energy transferred by conduction though the wall of a tube. But because there are two layers, namely a steel layer and an insulation layer, that make up the wall, equation 4.24 is rewritten as:

$$Q_{cond} = \frac{2\times\pi\times L\times(T_{\text{int}} - T_{surf})}{\frac{\ln\left(\frac{r_{mid}}{r_{\text{int}}}\right)}{k_{inner}} + \frac{\ln\left(\frac{r_{surf}}{r_{mid}}\right)}{k_{outer}}} \tag{4.25}$$

Knowing that the values for Q_{cond} and the Q_{conv} must be the same, equations 4.23 and 4.25 are combined, and assuming the inner wall temperature is the same as the fluid, Q_{loss} can be found from:

$$Q_{loss} = \frac{T_{\text{int}} - T_{amb}}{\frac{\ln\left(\frac{r_{mid}}{r_{inner}}\right)}{k_s\times 2\times\pi\times L} + \frac{\ln\left(\frac{r_{surf}}{r_{mid}}\right)}{k_w\times 2\times\pi\times L} + \frac{1}{h\times A_{surf}}} \tag{4.26}$$

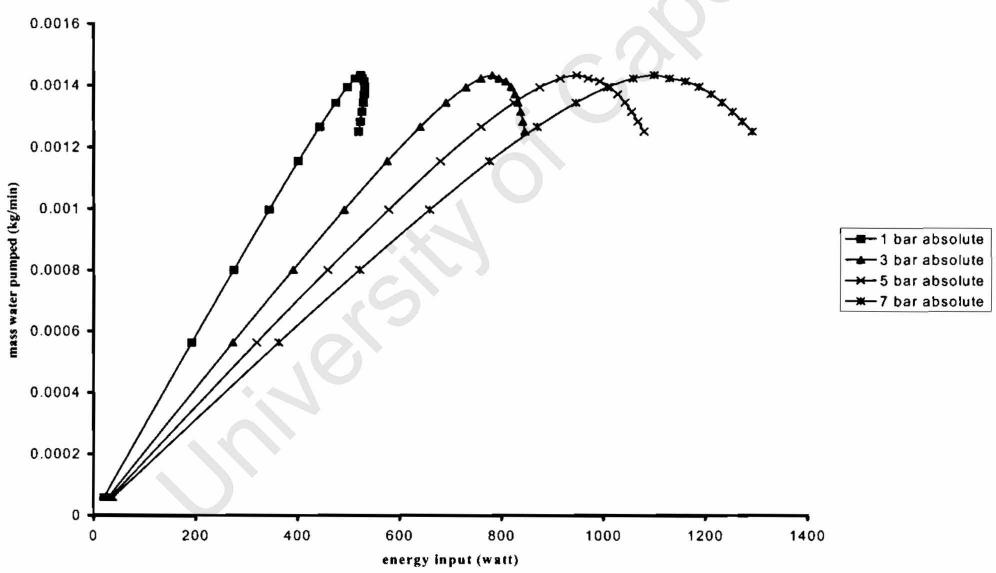

Figure 4.4) Mass flow vs. energy input as predicted by the mathematical solution for increasing pressure

4.1.2.4) Combining the energy calculations with the analytical model

The total energy used by the bubble pump at a specific condition is summed and the typical results can be seen in Figure 4.4. It shows the results achieved by the model, with different system pressures and a constant submergence ratio of 0.15. It is noted that for increased pressure the pump's volume flow rates are the same with only the energy needed to pump the same liquid mass flow changing. This is because more vapour needs to be released due to the increased vapour density, and the increased saturation temperature. It indicates that the pump relies on a specific, vapour to liquid volume ratio.

Chapter 5

Findings and Discussion of the water lift tube results

The bubble pump in the three fluid system has a double fold purpose.

- Create vapour for pumping action.
- Rise the solution to higher level in order for gravity effects to maintain solution circulation.

The vapour released will eventually become the refrigerant condensed in the condenser. The mass flow rate of which will dictate the refrigeration capacity of the refrigerator. According to established theory of absorption refrigeration [23] this mass rate of refrigerant will be supported by a) concentration difference between strong and weak solution b) a mass rates of the solutions. It is upon the bubble pump to maintain such a circulation, mainly of the weak solution. Therefore the main purpose of this study and in particular of the experiments performed is to investigate the possibility of increasing the pump rate by a combination of multiple lift tubes and increased power input. A fact, which is the very limiting factor of these absorption units.

The results obtained compared favourably to previous experiments and theory. In this chapter the results recorded from the water experiments will be viewed and discussed, and compared with the results obtained by the simulation model of it.

5.1) Adjusting the heat input to standardise the recorded results

The data collected from the water rig is in the form of volume flow rate and electrical power to the heating element. This data when plotted in the graph shown in figure 5.1, shows that there is a significant discrepancy when compared with the graph produced by Lister [14], figure 2.7 This discrepancy is accounted for by the fact that the water entering the boiler is heated from recovered heat and therefore there is no direct correlation of results. Therefore it is necessary to account for this recovered heat value and make adjustment to the heat input.

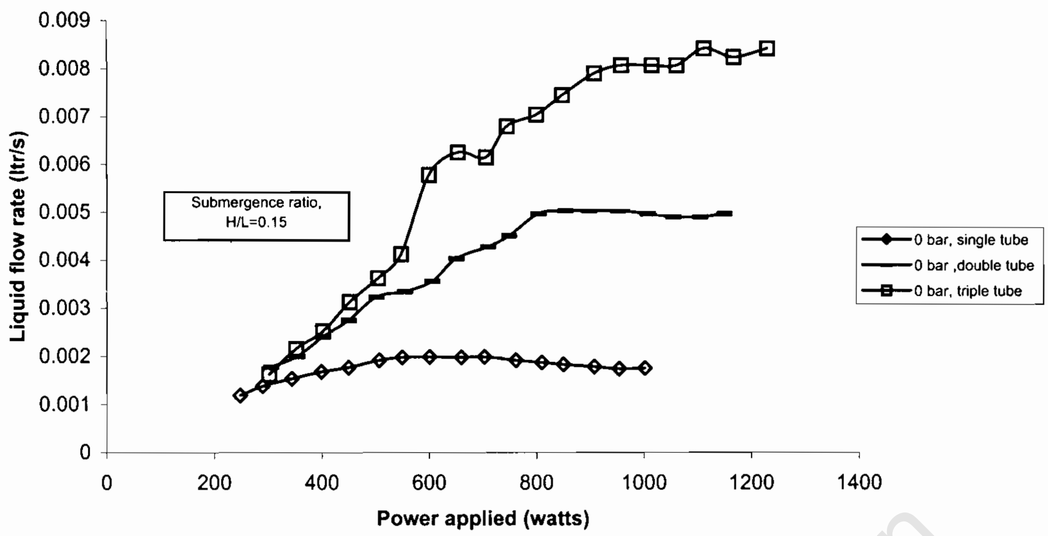

Figure 5.1) Flow rate vs. power input for unadjusted data for the water rig

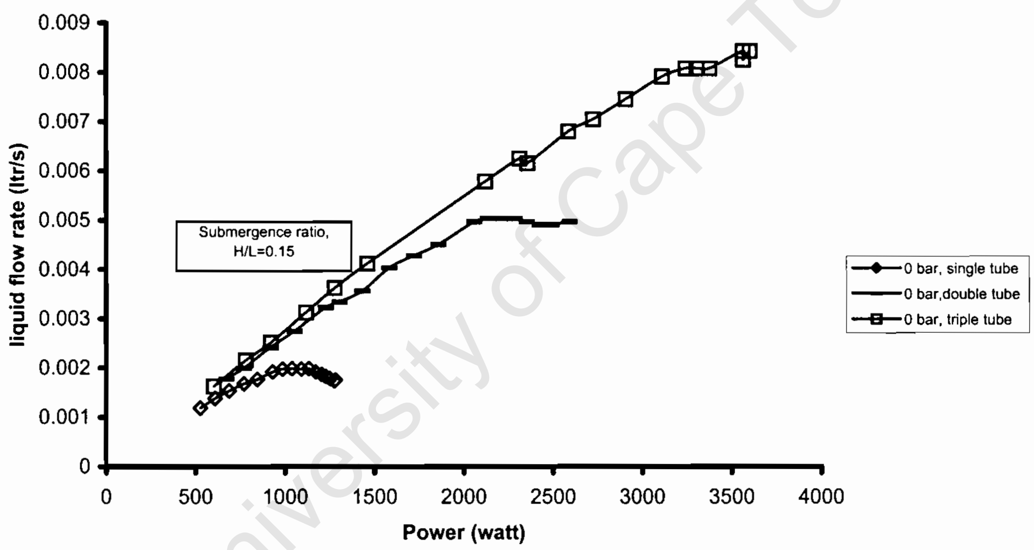

Figure 5.2) Flow rate vs. power input for the water rig data after it has been adjusted. It shows the pumping characteristics for various numbers of lift tubes at 0 bar gauge pressure.

A base temperature was chosen for the liquid input to the boiler, and all the heat inputs are adjusted to produce this base temperature, by adjusting the heat input from the electrical resistance heater. This base inlet temperature was chosen to be 20°C, which is the same as experiments done by Lister [14], and therefore will assist in correlating results.

To make the calculation of the recovered heat possible for the water rig, it is assumed that the mass flow rate of vapour in the pump is negligible, if it is compared to the mass flow rate of the liquid. And therefore the error involved due to not including the vapour in the energy calculation is also negligible.

The basic equation used to calculate the heat input to the water rig is as follows.

$$Q_{input} = Q_{elec\_heater} + Q_{recovered} \quad (5.1)$$

where

$$Q_{recovered} = \dot{m} \times Cp \times \Delta T \qquad \text{(Cp and } \Delta T \text{ are known)} \quad (5.2)$$

In order to ascertain the mass flow rate of the fluid a measuring cylinder and a stopwatch are sufficient to measure the volume flow rate. The temperature of the fluid in the measuring cylinder will assist in the determination of the fluid's density and thus the mass flow rate according to equations 5.3.

$$\dot{m} = \frac{V_f}{\rho \times t} \quad (5.3)$$

Then, knowing the real inlet temperature to the boiler and the desired base temperature, it is possible, using equation 5.2, to calculate the heat saved by recycling the hot fluid. The calculated recovered heat is then added to the electrical heat input to give a value for the total heat input in equation 5.1. The results of this adjustment can be seen when comparing figures 5.1 and 5.2. The adjusted data shown by the curves in figure 5.2 is the type that is expected taking into consideration the trends in figure 2.7.

5.2) Results and discussion on data collected from an increased number of lift tubes

Figure 5.2 also shows, that a single lift tube pump will become choked as heat input is increased above a certain threshold. This is seen as a drop in the volume of liquid pumped. It is a result of the gas volume flow rate exceeding the lift tubes limit, forcing a change of flow pattern in the lift tube from slug flow to annular flow. It is evident that by adding an extra lift tube it will increase the pump's ability to handle larger heat loads and flow rates. The trend is also seen in graphs 5.3-5.5 which are for increased internal pressure, it noted that the change in flow pattern becomes more pronounced with increasing pressure. These graphs form the focal point of this investigation because they can be extrapolated to cater for more tubes, more power input and therefore more liquid flow rate.

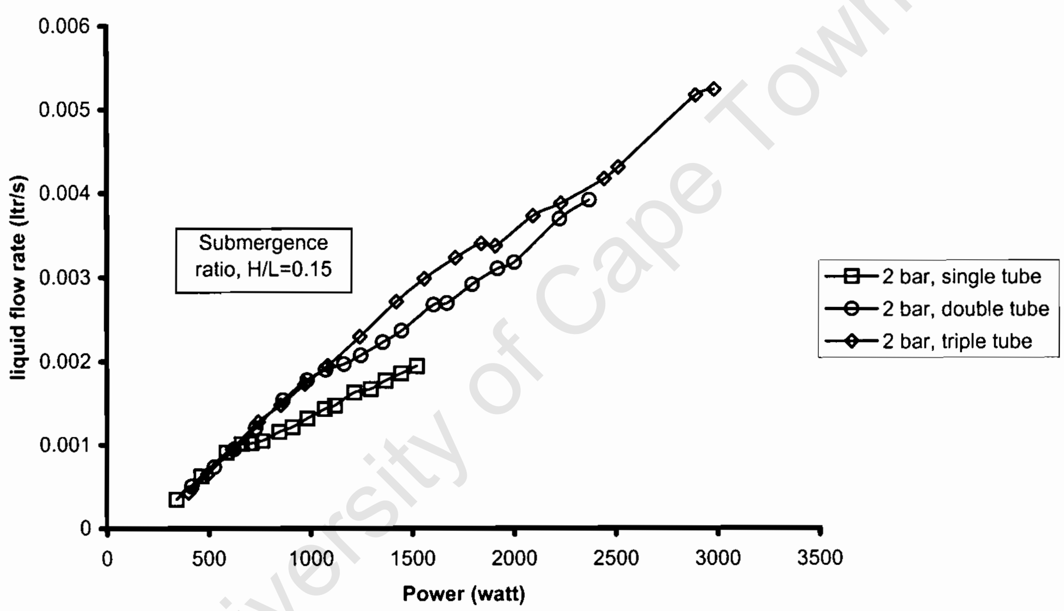

Figure 5.3) Flow rate vs. power input of 2 bar tests

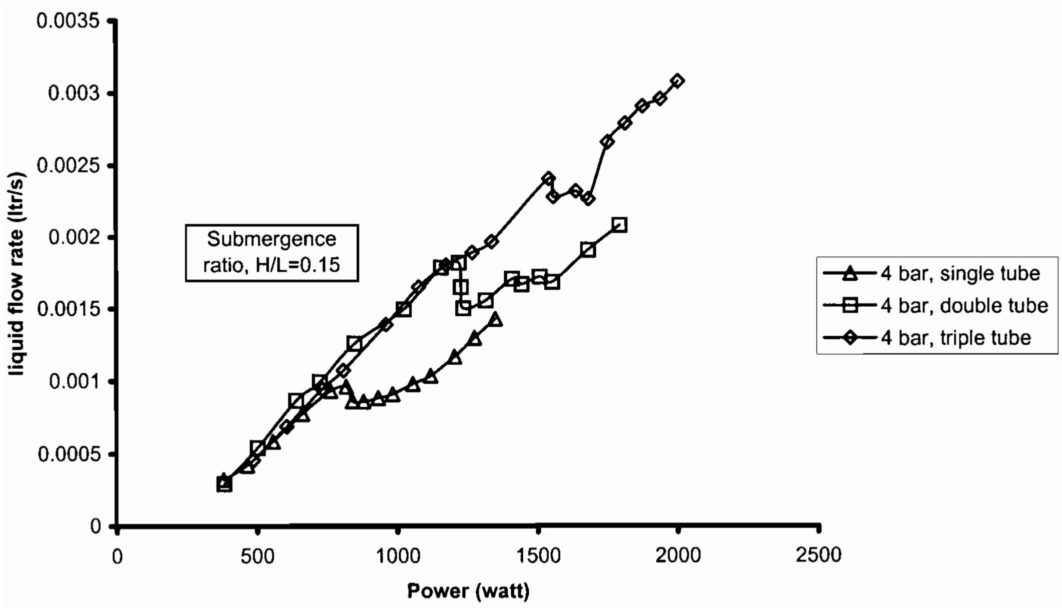

Figure 5.4) Flow rate vs. power input for 4 bar tests

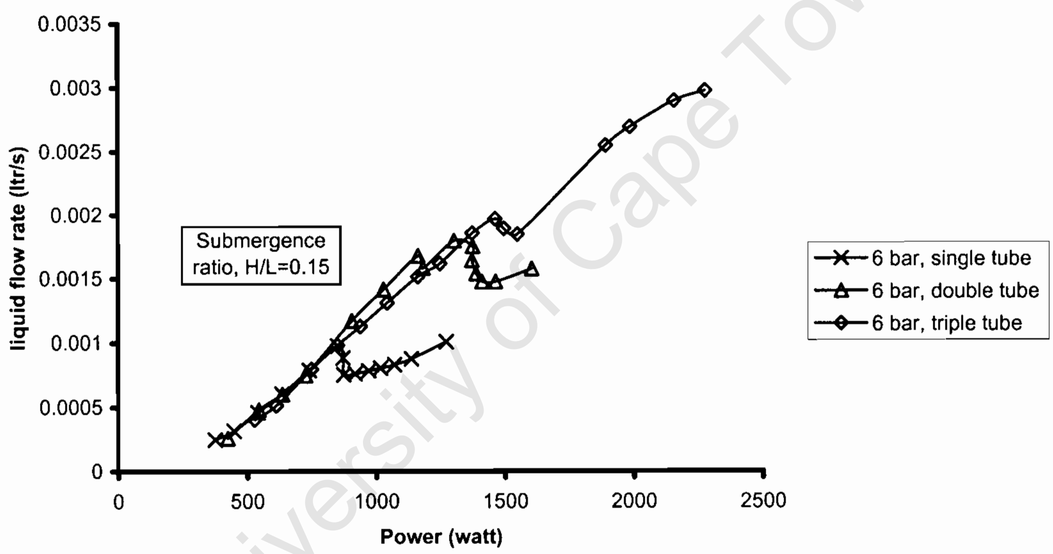

Figure 5.5) Flow rate vs. power input for 6 bar tests

However, the addition of extra tubes does not increase in the effectiveness of the system when it is operated within the slug flow regime. This is clearly seen in figures 5.2-5.5 by the curves grouped together at the lower range of power input. This trend is the same as that found when increasing the lift tube diameter, according to Lister's experiment [14][1].

It is clearly seen that all graphs show a linear relationship between liquid flow and power input until the threshold point for each set-up has been reached and a discontinuity in the flow rate is observed. Comparing the results of figures 5.2-5.5 it can be seen that more energy is needed for a given flow rate, as the pressure is increased. This can be ascribed to three main reasons:

- Increased boiling temperature
- Increased density of vapour
- Increased heat loss.

Each of these is discussed below.

- Boiling temperature

With increased system pressure, the boiling temperature is increased, which means that more energy is required to raise the temperature of the inlet liquid from the standard inlet temperature up to boiling temperature. However, for water, the vaporisation energy needed at a higher temperature is reduced, but it is insufficient to offset the increased energy required to raise the temperature to the higher levels required.

- Vapour density

The bubble pump works with an almost fixed ratio of vapour volume to liquid volume, whilst pumping in the slug flow regime [7]. As a result of the increased pressure the density of the vapour is increased, therefore requiring more liquid to be vaporised to

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[1] The curves initially form straight lines indicating a linear relationship between the vapour and liquid volume flow rates, and hence to the power input. In refrigeration practice the bubble pump will operate at its maximum efficiency in this linear region.

achieve the same volume of vapour in order to pump. This results directly in more energy being needed to vaporise a larger mass of liquid.

- Heat Loss

The elevated temperature difference between the pump and the environment will result in increased heat loss via conduction and convection. This heat loss has been kept to a minimum in this experiment by insulating the boiler and lift tubes. This loss will still have an effect, albeit small as calculated in appendix II.

The system pressure also has an effect on the flow transition. Figure 5.6 indicates a region of transition from what is believed to be either slug or churn flow to annular flow. The region's beginning is indicated with a line running through the transition areas. This line should have been horizontal indicating a constant volume ratio between vapour and liquid pumped. However it slopes because of the small changes in the liquid/vapour properties (i.e. decrease of density of the liquid, increase of vapour density, changes in the viscosities of both fluids etc., as the temperature increases).

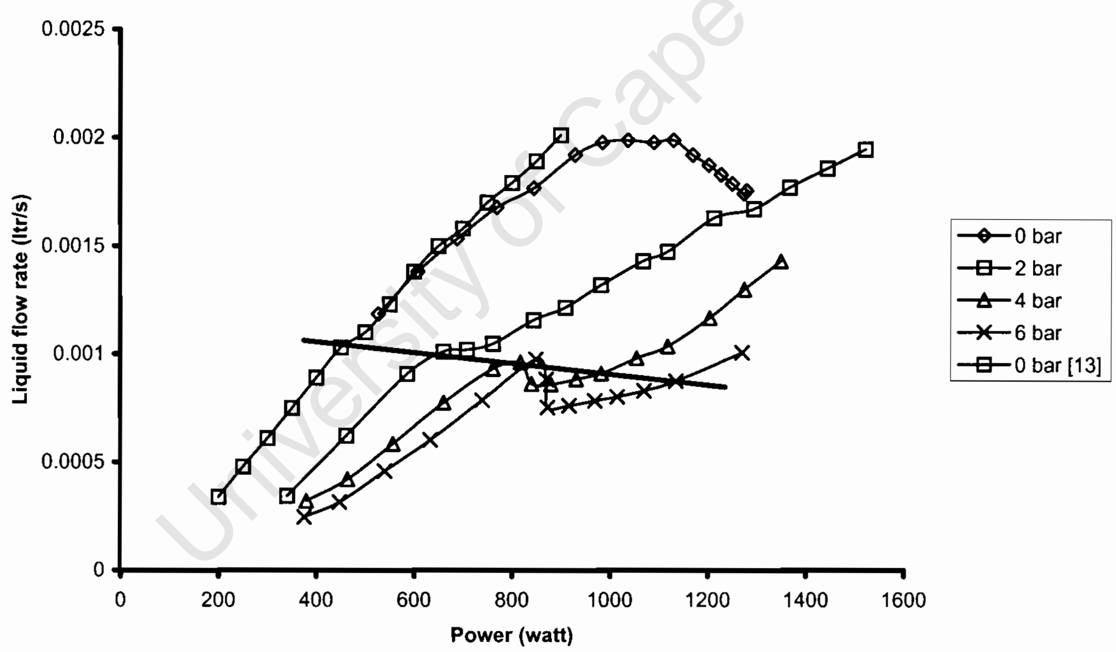

Figure 5.6) Flow rate vs. power input for the single lift tube. With a transition line indicating a change from bubble flow to churn flow and ultimately annular flow.

It is also noted that with an increase in pressure, the curves indicate an increase in the discontinuity at the transition region. It is therefore expected that at even higher pressures (the operating pressure of the fridge) the discontinuity to be severe. This region is to be avoided in the operating conditions of the unit, and the prediction model must indicate this critical region and calculate the bubble pump performance at maximum liquid flow.

5.3) Comparison of test data with the mathematical solution.

The graphs plotted in figures 5.7, 5.8 and 5.9 are for 1 tube, 2 tubes and 3 tubes respectively. The data points have been plotted against the mathematical model predictions. The graphs show the model's capability in predicting flow rates for multiple lift tube pumps.

The average difference between the analytical data and the real tests have been calculated for the three tube data using the relatively linear sections before the flow pattern experiences a change, see table 5.1

Internal Pressure Bar abs.	% Difference between Model and experiment
1	3
3	8
5	14
7	15

Table 5.1) Three tube experiment. Difference between the data from the mathematical simulation and experiment.

From the table and graphs it is seen that the model follows the experimental data fairly accurately in all pressures and tube set-ups within the region of linear relationship between flow rate and power input. The simulation model does predict when the flow regime will change, thus setting the upper limit of operating power input.

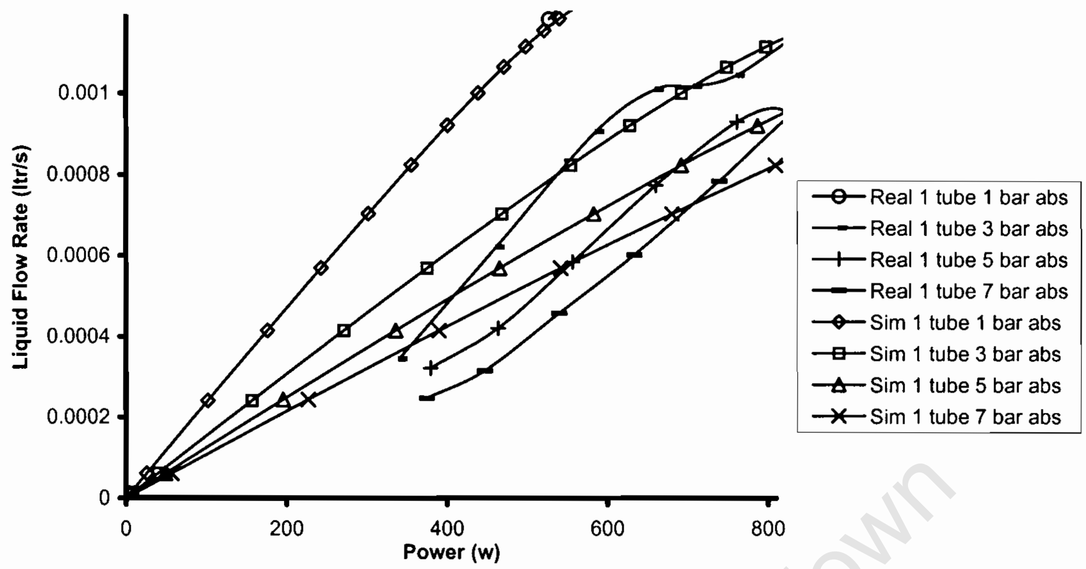

Figure5.7) Flow rate vs. power input, showing the test data with simulated data for 1 lift tube and increasing pressure, for the linear liquid/vapour ratio region.

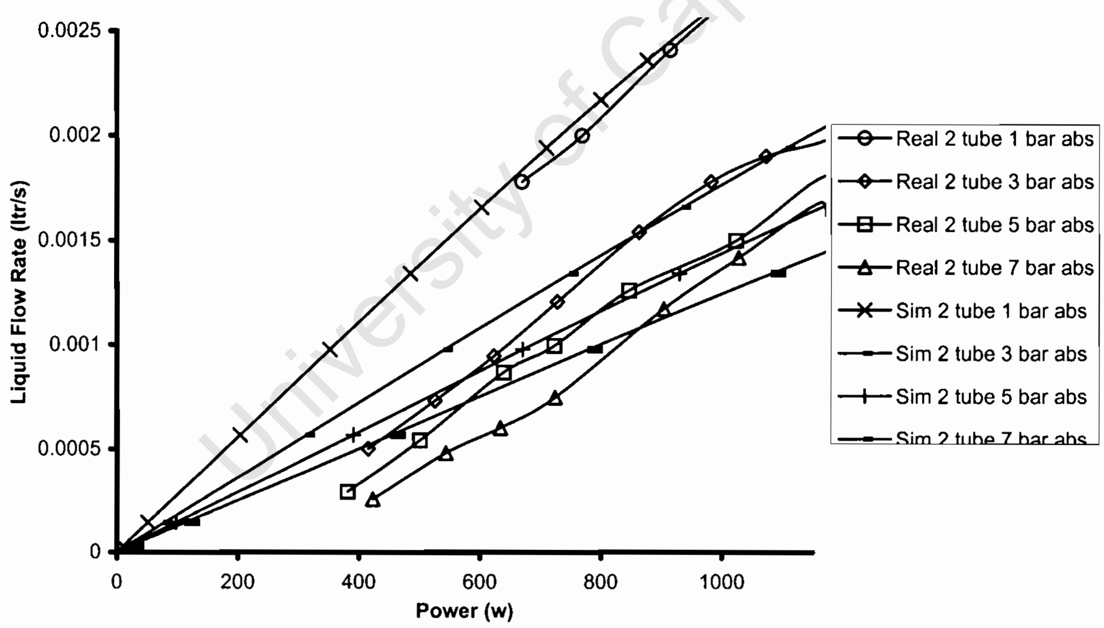

Figure5.7) Graph comparing the test data with simulated data for 2 lift tubes and increasing pressure, for the linear liquid/vapour ratio region.

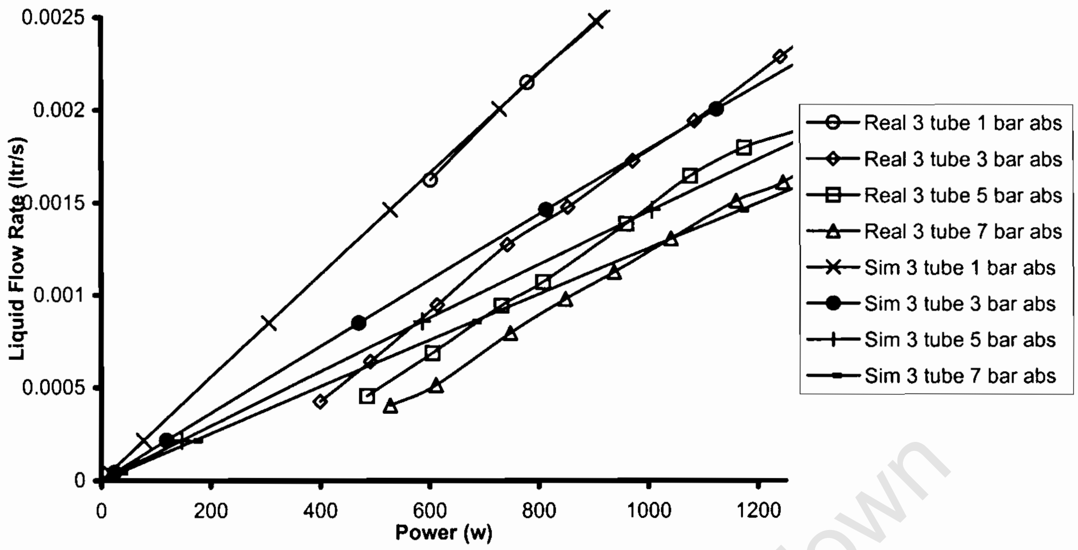

Figure 5.9) Graph comparing real and simulated data for 3 lift tubes and increasing pressure, for the linear liquid/vapour ratio region.

The difference in values indicated in table 5.1 are potentially due to heat losses to the environment that were not taken into account, which become more pronounced as the boiler temperature is increased. An additional factor to these differences is that pumping at low power was not consistent. This may be due to the heat energy supplied, being insufficient to release vapour in large enough quantities to start the pumping action. This statement is clearly supported in figure 5.9 in which, at lower power the real values deviate from the simulated values by a larger degree.

For refrigerator design purposes it is necessary to know the vapour volume flow rates. As the vapour mass flow rate will usually be the direct result of the desired cooling required from the design. The vapour volume flow rate will also dictate the liquid volume flow rate. The mathematical model will be able to use these values to predict the required tube diameters, tube numbers and submergence ratio that will result in a successful pump design.

Chapter 6

Findings and discussion of the refrigerator performance.

Testing was conducted on the modified refrigerator using different numbers of lift tubes, and varying the heat inputs to the boiler. The evaluation will begin with observations on the working conditions of the experimental rig. Thereafter each component's performance will be studied in turn, looking in particular at the results of additional lift tubes at various boiler heat inputs.

6.1.) Observations

The system was charged with a 25% ammonia solution by weight, which at an ambient temperature of 20°C maintained an equilibrium pressure of 0.5 bar abs [13]. The system was charged also with hydrogen (above the equilibrium pressure of the solution) to a total pressure of 20-bar abs.

During no load operation of the refrigerator, and 450W heat input, the system pressure was 22.5 bar abs. This indicates a refrigerant partial pressure in the evaporator of approximately 2.5 bar. The minimum evaporator temperature achieved experimentally in this no load condition was –15°C, which together with its partial pressure of 2.5 bar supports a refrigerant concentration of 99%. .

With a constant evaporator load and with the evaporator temperature varying between 0 and 5°C, the operating pressure was 25 bar abs, and was consistent throughout the heat input range between 450W and 700W. It did however increase by 0.5 bar at heat inputs range between 800W to 1100W. Under these conditions the refrigerant partial pressure was 5 bar, according to the solution's equilibrium data [13]. The agreement of the rig's operating conditions with the enthalpy concentration diagram indicates that the experimental unit was working correctly, and that the data obtained would be reliable.

6.2) Performance of the condenser

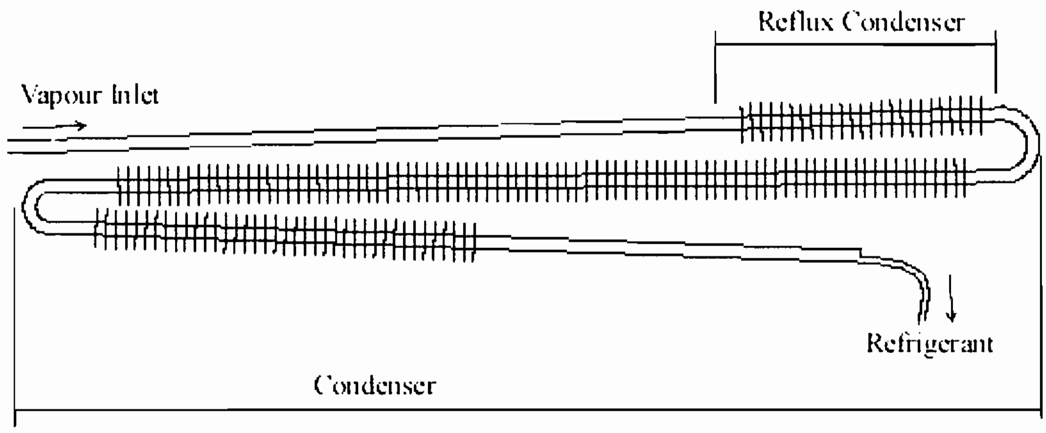

Figure 6.1) Diagram of the condenser

According to Vicatos [24], a refrigerant concentration of between 0.99 and 0.995 in ammonia is easily achieved and therefore quite common in refrigerators. A vapour with such a concentration will condense in the condenser and produce the liquid refrigerant. The condensing temperature of the refrigerant and its concentration will establish the operating high pressure of the system.

Vicatos [24] used this approach to determine the working conditions of a unit having only as known requirements the condenser temperature the evaporator temperature and the concentration of the refrigerant. Using this approach one can determine the refrigerant's properties from a commercially available unit. From paragraph 6.1 it was found that that refrigerant entering the evaporator had an ammonia concentration of 0.99.

Figure 6.6 shows the construction that graphically evaluates the temperature of an ammonia/water vapour that condensed to form a 0.99 pure ammonia solution refrigerant, given a system pressure of 25 bar abs. These two conditions indicate that the temperature of the condensate is 60°C in the condenser. This temperature was verified experimentally and it will be explained in the paragraphs to follow.

A further drop is encountered in the condenser due to sub cooling of the refrigerant in the condenser's tubes.

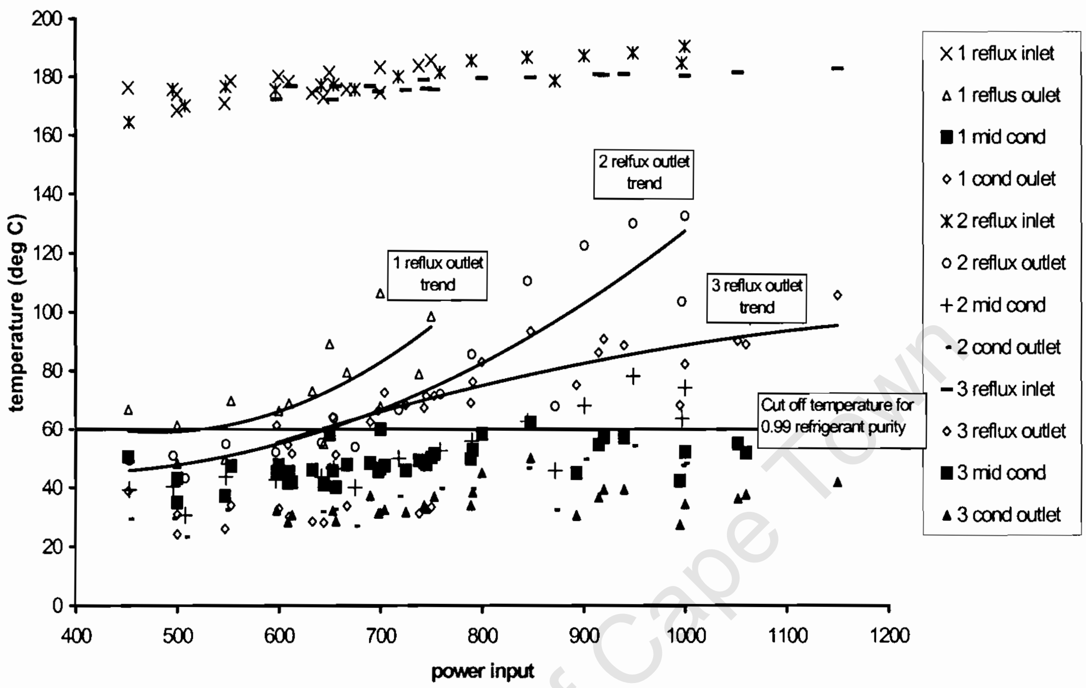

Figure 6.2) Graph showing the performance of the condenser and the cut off temperature for quality refrigerant.

The chart in figure 6.2 shows the temperatures found in the condenser during testing. It also shows a plot indicating the vapour temperature at the exit of the reflux condenser. It is important that this temperature remain constant at 60°C (for this experiment) because together with the operating pressure established by the condenser, the reflux condenser maintains a 0.99 refrigerant concentration. The trend lines in the figure indicate the temperatures at the exit of the reflux condenser for different heat inputs and number of lift pump tubes. It allows a clear indication of the operating conditions, whether the vapour temperature leaving the reflux condenser is higher or lower than 60°C. It is seen that for increasing heat input, the exit temperature of the reflux condenser increases above the 60°C temperature line. A probable explanation to this is that it was subjected to an increased heat flux from the variable heat in the boiler, more than it was able to reject.

Because the high pressure remained constant during experiments, a higher temperature input to the condenser would indicate that a poorer refrigerant in ammonia would condense. This poorer refrigerant would negatively affect the refrigeration conditions, particularly the operating temperatures of the evaporator (input and output temperatures due to the temperature glide).

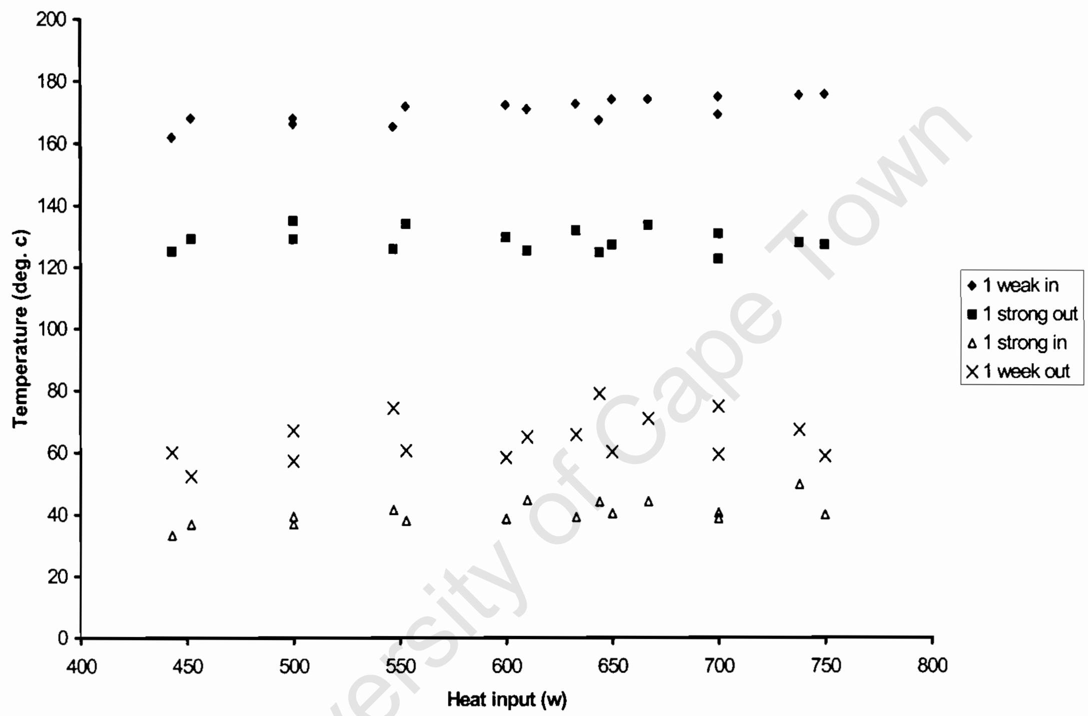

Figure 6.3 a) Heat exchanger temperature response for 1 lift tube.

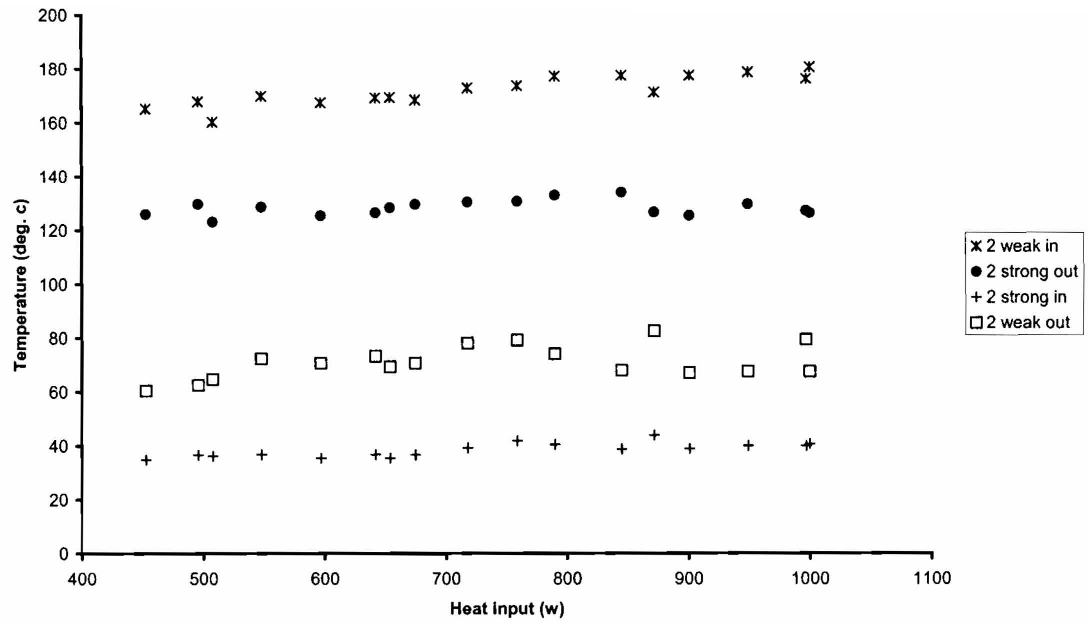

Figure 6.3 b) Heat exchanger temperature response for 2 lift tubes.

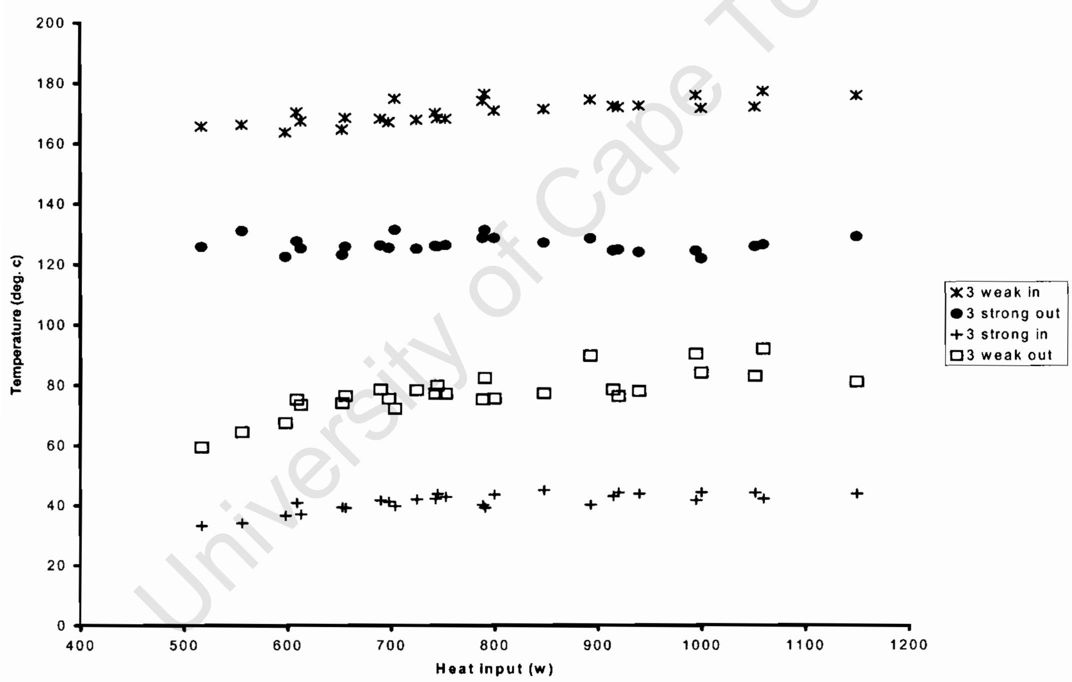

Figure 6.3 c) Heat exchanger temperature response for 3 lift tubes.

6.3) Performance of the liquid heat exchanger

The function of the liquid heat exchanger is to recover heat energy from the weak solution exiting the separator, and to transfer it to the strong solution entering the boiler. This transfer enables a large heat input saving, making the system more efficient.

The effectiveness of a counter-flow heat exchanger is given as, Holman [9]:

$$\varepsilon = \frac{(Th_1 - Th_2)}{(Th_1 - Tc_2)} \qquad (6.1)$$

Th_1 is the temperature of the entering weak solution.
Th_2 is the temperature of the weak solution exiting.
Tc_2 is the temperature of the entering strong solution.

The use of equation 6.1 is based on the hot, weak solution coming from the separator being the fluid with the least energy available to transfer[2]. The data used can be seen in figure 6.3 a, b, c.

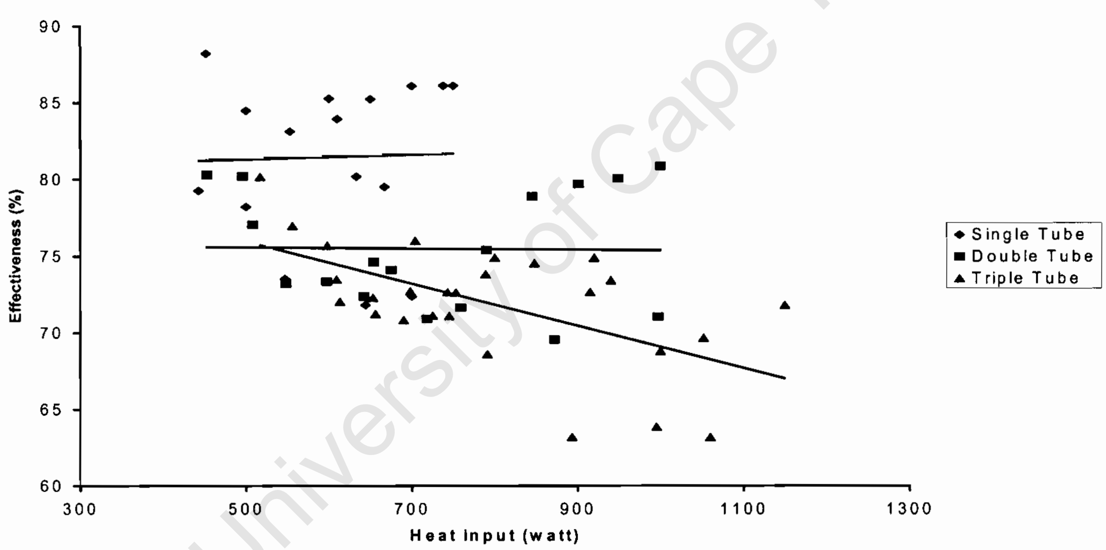

Figure 6.4) Effectiveness vs. heat input of the solution heat exchanger

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[2] The weak solution is used because it has both lower mass flow rate and a lower concentration of ammonia, which has a higher heat capacity than water

From figure 6.4 it can be seen that the effectiveness of the heat exchanger drops as the number of lift tubes increases. With exception of the single lift tube operation, in which it shows a slight increase in effectiveness, it shows a decrease for the two and three lift tubes operation.

Evidence from figure 6.4 indicates to a poorly designed heat exchanger. One would expect a liquid-liquid heat exchanger to have an average effectiveness higher than 0.95 but only 0.82 was achieved. A possible explanation is that a commercially available unit was used for experimentation in which (knowing to the author) the heat exchanger had the hot weak solution flowing in the outside annular space, thus loosing a considerable amount of heat to the environment.

Vicatos and Zulu [24] experimented with a similar three fluid system, (also commercially available) to which modifications were made to the heat exchanger among other components. From the temperature response of the modified heat exchanger, one can deduct by using the Effectiveness NTU method (eq. 6.1) that an effectiveness approaching unity was achieved.

For a single tube operation the heat flux through the heat exchanger was within the operational limits. Therefore the slight increase in effectiveness shown in figure 6.4 it is noted and accepted.

The drop in effectiveness that is shown for the two and three lift tube operations and also the continuous drop in effectiveness with an increase of heat input, shows that the heat exchanger was outside its designed characteristics. This will have an effect to all the other components of the unit, which will be discussed later.

This affects the system in two ways.

- The absorber will have to dissipate more heat than is necessary, which wastes heat and the absorber will become unable to continue its task properly.

- The boiler will have to supply unnecessary heat to the strong solution in order to lift it up to boiling temperature.

It is believed, as a result of the reasons stated on the previous page, that a more effective exchanger would increase the efficiency of the system as a whole. Therefore, when designing new systems they should be made as effective as possible.

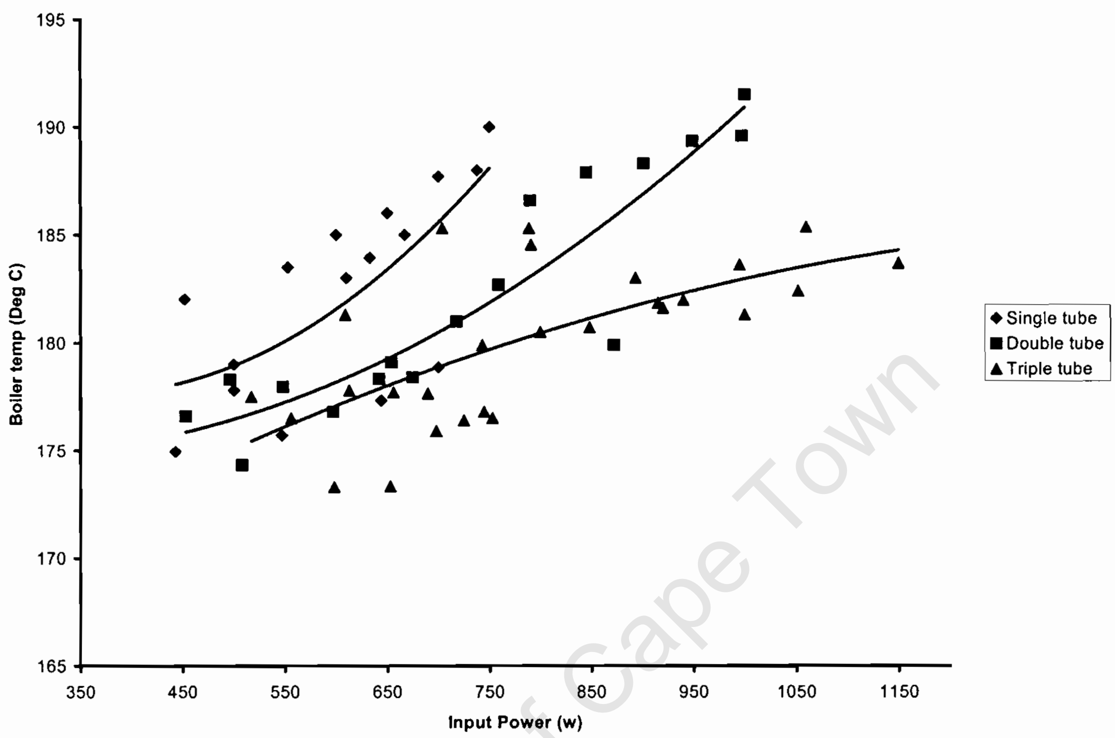

Figure 6.5) Boiler temperature response to increased heat input and lift tubes.

6.4) Boiler response to increased number of lift tubes

Referring to figure 6.5, it is noted that the boiler temperature is reduced with the addition of lift tubes. This is a result of a combined effect: the heat exchanger's performance, and a higher liquid flow rate as indicated by figures 6.3 a, b, c and figures 5.1-5.5.

With reference to the heat exchanger's performance, the author cannot comment on the outlook of the boiler's performance should a higher effectiveness heat exchanger was in use throughout the experimentation

The higher flow rates of solution circulation, indicates that it is supported by a small difference in concentration between the strong and weak solutions, Vicatos [23]. This means that the cooler weak solution is at equilibrium with a much purer vapour mixture and therefore requires less purification as shown in figure 6.6.

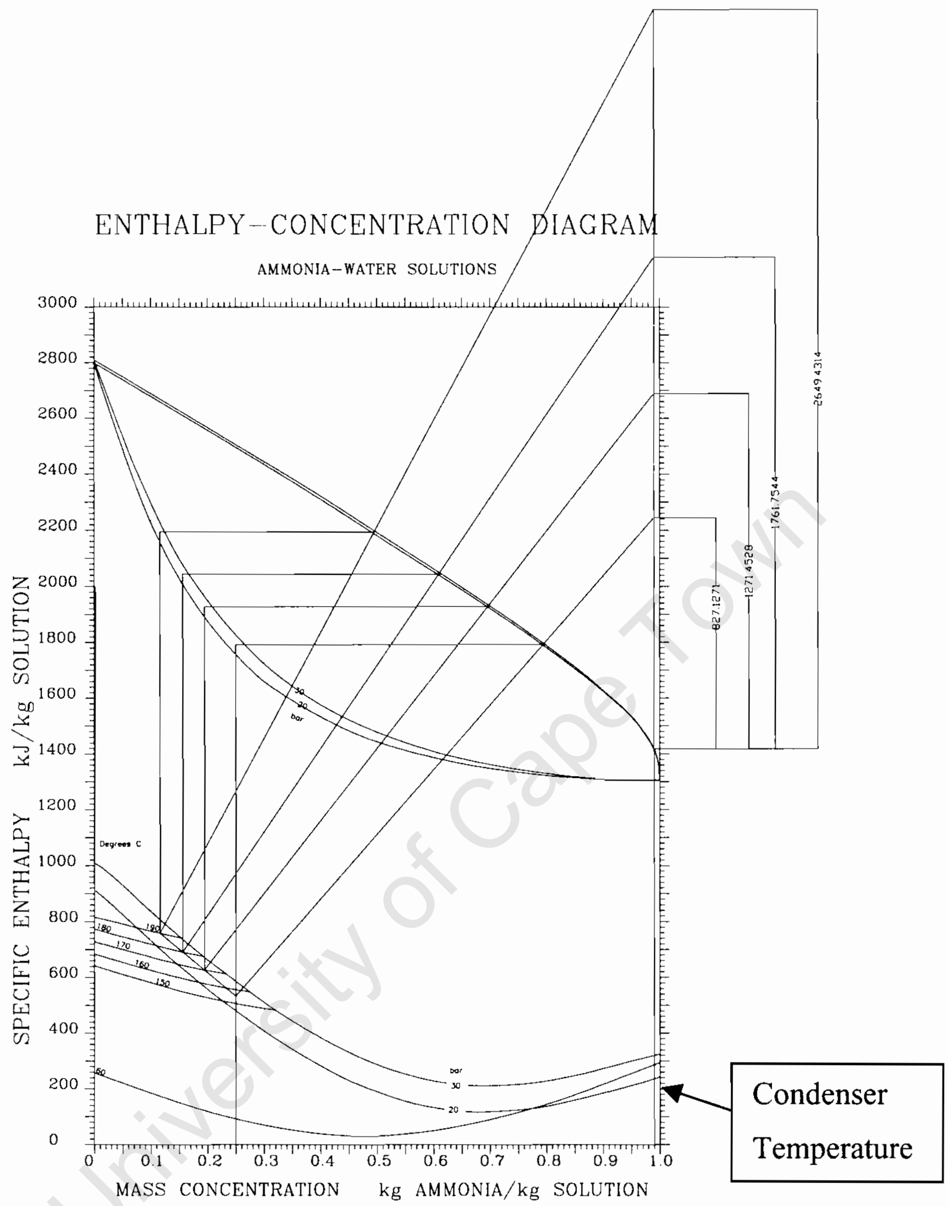

Figure 6.6) Construction in a H-x diagram showing that increased boiler temperature results in increased heat loss in the reflux condenser. The operating condenser temperature is also depicted.

With reference to figure 6.6 an increase in the boiler temperature indicates a dramatic increase of the rectification energy required to maintain vapour purity. This complies with the results and comments of paragraph 6.2. An additional factor, which would adversely affect the performance of the entire unit, is that at high boiler temperatures ammonia will dissociate (beginning at 180 °C) and form inert gasses that will negatively affect the system's performance. [28].

6.5) Performance of the absorber

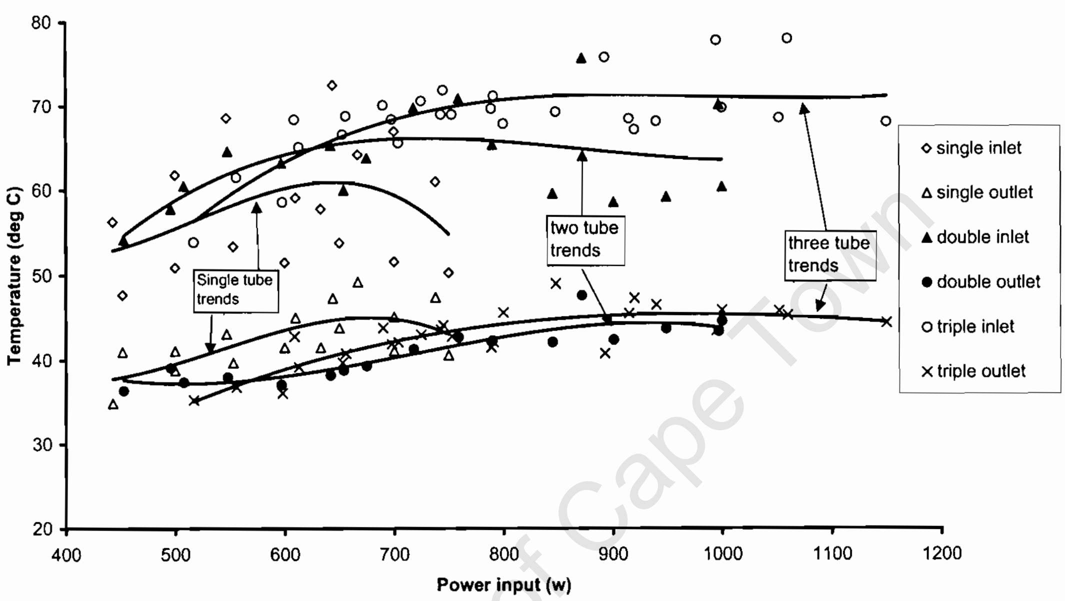

Figure 6.8) Response of the absorber

Having as a reference the problematic behaviour of the three fluid system due to the inadequate design of the absorber as describes by Vicatos [23], particular importance was paid to the external artificial cooling of the absorber (see chapter 3). With this external cooling, it was attempted to keep the strong solution temperatures at the exit of the absorber around 40°C, figure 6.8. However, it is seen in figure 6.9 that as the heat input increases in the boiler (either due to the actual heat added by the element or by the increased flow rate due to the number of tubes), the temperature of the fluid leaving the absorber continues to rise.

A temperature rise within the absorber would slow ammonia absorption rate, resulting in an ammonia build up in the evaporator, an increased ammonia partial pressure, an increased temperature, and a drop in refrigeration effect and capacity.

6.6) Performance of the evaporator

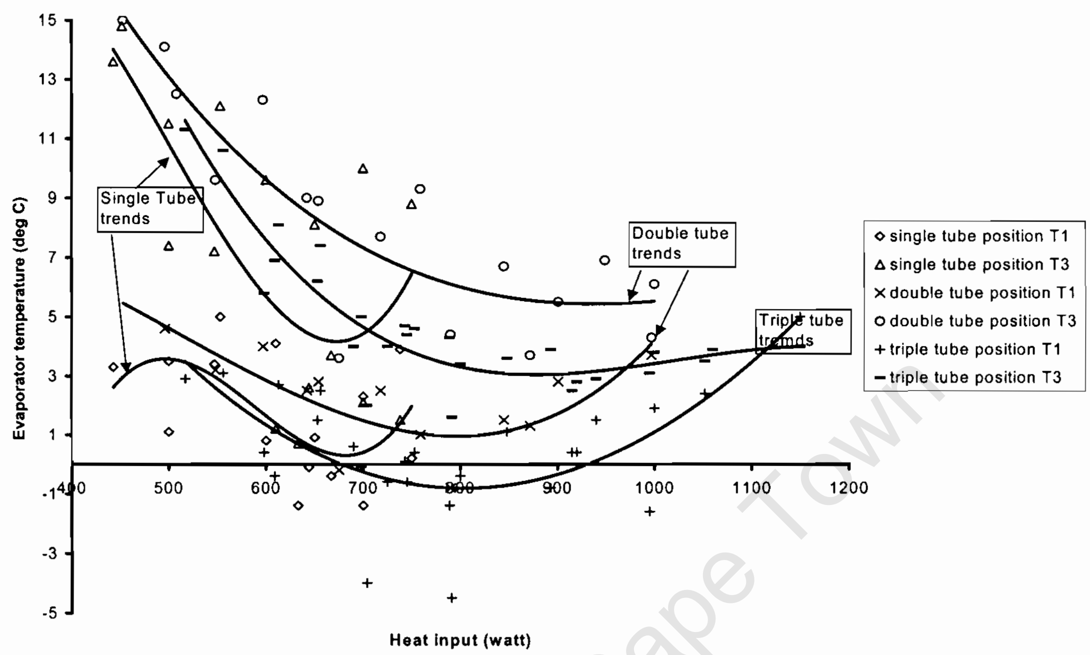

Figure 6.7) Evaporator response to different numbers of lift-tubes and heat inputs. The numbers 1 and 3 in the legend indicate thermocouple positions at evaporator's inlet and outlet respectively.

The evaporator performed satisfactorily, see figure 6.7. The temperatures achieved by the evaporator during testing, as mentioned in chapter 3, were established by keeping the load temperature above 0°C, and as constant as possible. Measurements were taken at steady state conditions and data was recorded only for inlet temperatures to the evaporator below 5°C.

6.6.1) Refrigeration performance

Figure 6.8 shows the cooling performance of the refrigerator as heat input is varied. The data shows a large amount of scatter within the results, but the general trend of the curves indicate that there is a steep increase in cooling capacity for the first part of every combination of heat input and number of tubes. It also shows a steep drop in performance after a certain power input has been reached, which is a culmination of:

- The bubble pump becoming choked and supplying vapour that is of a poor quality
- The rectifier is unable to cope with the increased water vapour and heat flux.
- The condenser supplying a poor refrigerant.
- The absorber's inability to cool the solutions
- The absorber's inability to absorb the vapours from the evaporator

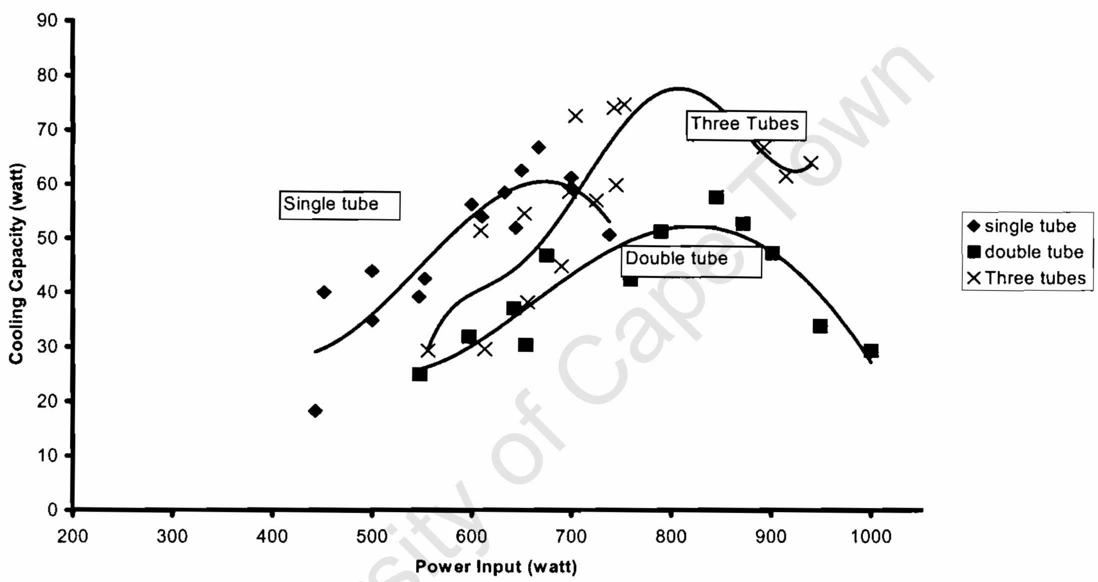

Figure 6.8) Performance of the refrigerator with respect to the heat energy input.

From the graph it can be deduced that the COP of the refrigerator is low, constantly less than 0.1.

When the cooling capacity performance of the unit is compared between all lift-tube configurations there appears to be no plausible explanation to the poor performance of the double lift tube configuration, except for the fact that all experimentation was forced to be on the same experimental rig.

Chapter 7

Conclusions and Recommendations

7.1) Conclusions

The aim of this dissertation is to determine whether the use of a multiple lift tube bubble pump is a viable option that could be used to increase the refrigeration capacity in diffusion absorption refrigeration plants. From the data and results obtained it can be seen that a multiple lift tube bubble pump is a viable and workable solution.

The addition of extra lift tubes supply a constant fluid/vapour ratio (as the single lift tube) but at increased heat inputs beyond the single lift tube capabilities. This was clearly indicated in this study, within the preliminary experiments using water as the working medium. The results helped in obtaining the mathematical prediction model for a number of lift tubes and heat input combinations. When the same pump set-up was transferred into an existing absorption plant this effect was not as clearly evident as for the water system.

Previous studies such as [23], [24] and [28] indicate that absorption machines are sensitive to their working environment and produce the desired performance only if the components operate within their designed parameters. Therefore, it can be stated that the poor performance of the experimental refrigeration unit was not due to experimental errors and construction inaccuracies, but due to taxing the performance of each component of the unit beyond its design characteristics.

Research has shown [23] that absorption systems could attain COP values as high as 0.6, while the experimental rig attained a maximum of 0.1. According to Vicatos [23], the maximum COP is estimated by knowing the evaporators and condensers temperatures. Therefore it is believed that regardless of the heat input /number of tubes combinations is immaterial to the maximum COP.

7.2) Recommendations

It is therefore recommended, for the verification of the above statement in the conclusion, to build three separate refrigeration units each one having a different combination of heat input and number of lift tubes. These units must be designed taking into consideration the temperature environments [23], as well as the refrigeration capacities, which eventually will size the components of the units. It is believed that all the units will obtain a COP in close proximity to each other.

Chapter 8

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Appendix A

The tubes used in the apparatus had to be safe, as they would be carrying hot fluids under pressure. Below is the basic design calculation for the safe tube wall thickness with respect to its diameter.

Using thin cylinder theory (thickness less than 1/20 internal diameter, *FOS* 1) we know that hoop stress is the larger stress component Hearn [8]. What is wanted is a formula that will give the wall thickness of the tube for a specific pressure and material.

$\sigma_h = \frac{P \times D}{2 \times t}$ Hoop stress in a thin walled tube.

Changing the subject of the formula

$$t = \frac{P \times D}{2 \times \sigma_h}$$

$P = 2.5 \times 10^6$ Pa.

$\sigma_h = 200$ MPa. Mild steel

$\sigma_h = 30$ MPa Glass

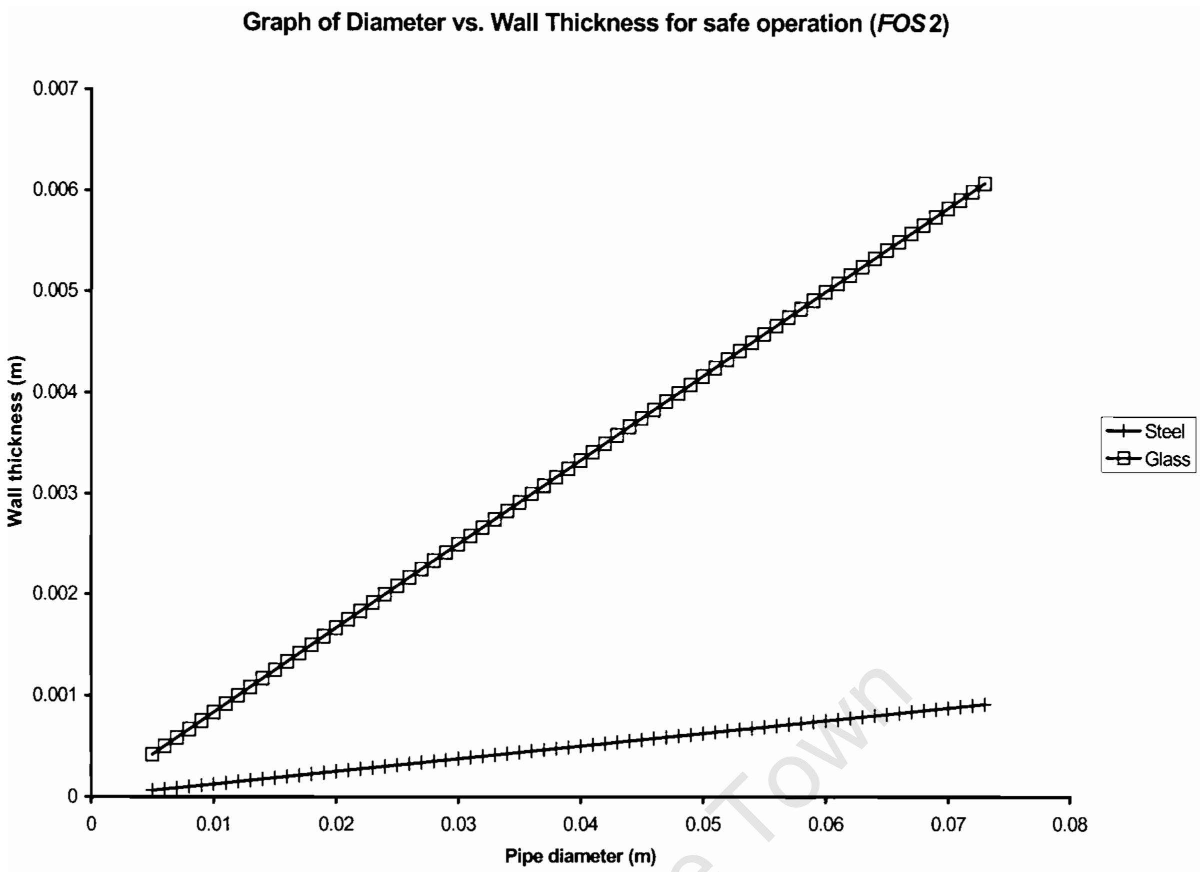
$FOS = 1$ Factor of safety

Resulting equations are

$t = 0.006 \times D$ Steel

$t = 0.042 \times D$ Glass

The graph following gives a quick reference for tube size vs. wall thickness. It can be seen that most standard steel and glass tubing will be safe to use and pose no strength problems.



Graph of Diameter vs. Wall Thickness for safe operation (FOS 2)
Wall thickness (m)
0.007
0.006
0.005
0.004
0.003
0.002
0.001
0
0
0.01
0.02
0.03
0.04
0.05
0.06
0.07
0.08
Pipe diameter (m)
Steel
Glass

Appendix B

This appendix is an evaluating calculation that will determine the necessity of adding the energy lost to the environment to the heat input calculations. The lift-tube need not be considered, as the energy lost from it should not affect the energy needed to boil the liquid, as the boiling has already occurred. It is used because it will give a result a worst-case scenario. The calculations below are for a vertical tube in air with natural convection. There are other faces from which heat is lost, but they are small and will have results that are insignificant compared to the losses from the boiler and lift-tube.

Assumptions and input data [9]

- Steady state
- Assume no radiation
- The inner walls of the boiler and lift-tubes are at liquid temperature
- Glass wool thickness 10 mm
- Ambient temperature T_{amb}=20 °C
- Liquid temperature T_{wall}=100-160 °C for water rig
- h_o=4.5 W/m^2 °C
- k_{steel}=51 W/m °C (at 125 °C)
- $k_{glass\ wool}$=0.038 W/m °C
- Boiler ID $_{steel}$ =66mm
- Boiler OD $_{steel}$ =76mm
- length of boiler 250mm
- lift-tube ID_{steel} =6mm
- Lift-tube OD_{steel} =8mm
- length of lift-tubes 450mm

$$q_{cond} = \frac{2 \times \pi \times L \times (T_{wall} - T_{surf})}{\frac{\ln\left(\frac{r_{so}}{r_{si}}\right)}{k_s} + \frac{\ln\left(\frac{r_{wo}}{r_{wi}}\right)}{k_w}} \tag{II,1}$$

$$q_{conv} = h \times A \times (T_{surf} - T_{amb}) \tag{II,2}$$

Combining the above equations

$$q = \frac{T_{wall} - T_{amb}}{\frac{\ln\left(\frac{r_{so}}{r_{si}}\right)}{k_s \times 2 \times \pi \times L} + \frac{\ln\left(\frac{r_{wo}}{r_{wi}}\right)}{k_w \times 2 \times \pi \times L} + \frac{1}{h_o \times A_o}} \tag{II,3}$$

For the worst-case scenario the heat loss is 20 watts for the boiler and 12 watts for the lift-tube, a total of 32 watts being lost. With the minimum wattage used being 400 watts, it results in a 7.5% of the smallest applied wattages. The discrepancy will then decrease as the applied wattage is increased. This inaccuracy has been left out of the calculations, as it is not an accurate calculation and will add other flaws to the results. For instance, it must be decided from which exact areas heat loss is happening. This calculation is only intended to give an indication of the losses and how much they will affect the systems performance.

Appendix C

The data recorded for the water tests at various pressures and lift-tube numbers.

Water test for 1 tube at 1bar abs.

Measure Volume	0.387	ltr	Tbase 20 °C
Cp	4200	J/kg.K	

Test no.	Energy in	time	Time	pump rate	T1	T2	T3	T4	T5	T6	T7	T8	heat water	total watts
	watt	min s	sec	ltr/s	°C	°C	°C	°C	°C	°C	°C	°C	watt	watt
1	399	3.51	231	0.00168	72.6	81.4	100.2	99.9	95.4	90.9	19.6	18.2	370.3	769.3
2	450	3.36	216	0.00179	72.4	78.7	100.5	100.1	95.7	91.3	19.5	17.6	395.0	845.0
3	507	3.22	202	0.00192	72.4	79.6	100.5	100.1	95.7	91.0	19.9	17.5	422.4	929.4
4	550	3.16	196	0.00198	72.4	78.9	100.5	100.1	95.7	91.1	19.6	17.5	435.1	985.1
5	601	3.15	195	0.00199	72.3	78.5	100.4	100.1	95.7	90.8	19.7	17.4	436.7	1037.7
6	660	3.16	196	0.00198	71.9	76.6	100.2	99.8	95.4	90.2	19.7	17.4	430.5	1090.5
7	703	3.15	195	0.00199	71.3	77.5	100.3	99.9	95.5	89.9	19.7	17.3	428.2	1131.2
8	757	3.23	203	0.00191	71.0	78.7	100.4	100.0	95.6	89.7	19.8	17.3	408.6	1165.6
9	762	3.22	202	0.00192	70.6	80.8	100.4	100.1	95.7	89.4	19.5	17.3	407.9	1169.9
10	810	3.27	207	0.00187	70.0	79.8	100.4	100.0	95.6	89.2	19.7	17.4	393.1	1203.1
11	850	3.32	212	0.00183	69.2	78.8	100.5	100.1	95.7	88.7	19.8	17.3	377.7	1227.7
12	908	3.37	217	0.00179	69.0	78.5	100.5	100.1	95.7	88.6	19.9	17.2	367.4	1275.4

Water test for 1 tube at 3 bar abs.

Measure Volume	0.387	ltr	Tbase 20 °C
Cp	4200	kJ/Kg.K	

Test no.	Power in	Time		Time	Pump rate	T1	T2	T3	T4	T5	T6	T7	T8	Heat water	Total watts
	watt	min	sec	sec	ltr/s	°C	°C	°C	°C	°C	°C	°C	°C	watt	watt total
1	251	18	44	1124	0.00034	81.7	112.5	135.0	134.6	127.6	98.6	18.3	21.2	89.3	340.3
2	303	10	23	623	0.00062	80.6	109.9	134.8	134.4	128.2	107.1	17.5	21.2	158.4	461.4
3	357	7	7	427	0.00091	80.0	107.5	134.5	134.1	128.1	112.0	17.5	20.5	228.6	585.6
4	406	6	23	383	0.00101	79.7	106.3	134.1	133.7	127.7	113.6	17.4	19.9	253.8	659.8
5	452	6	20	380	0.00102	79.7	106.1	134.3	133.9	127.9	114.1	17.5	19.6	255.8	707.8
6	498	6	10	370	0.00105	79.8	105.3	134.3	133.9	127.9	114.4	17.4	19.2	262.9	760.9
7	555	5	35	335	0.00116	79.6	102.4	134.4	134.0	128.0	115.7	17.5	19.1	289.4	844.4
8	605	5	19	319	0.00121	79.7	102.2	134.4	133.9	128.0	115.9	17.5	19.0	304.6	909.6
9	650	4	54	294	0.00132	80.0	101.8	134.6	134.2	128.2	116.6	17.5	19.0	332.1	982.1
10	704	4	31	271	0.00143	80.7	101.0	134.8	134.4	128.4	117.4	17.4	19.0	364.4	1068.4
11	752	4	23	263	0.00147	79.2	99.4	134.1	133.7	127.7	116.7	17.6	18.6	366.5	1118.5
12	800	3	58	238	0.00163	80.4	105.3	134.3	133.9	127.9	116.9	17.6	18.8	413.3	1213.3
13	853	3	52	232	0.00167	83.0	106.2	134.0	133.6	127.6	117.2	17.5	18.8	441.9	1294.9
14	900	3	49	229	0.00169	84.1	99.7	134.2	133.8	127.8	118.2	17.5	18.5	455.5	1355.5
15	954	3	25	205	0.00189	85.6	108.1	134.3	133.9	128.0	117.5	17.3	18.4	520.9	1474.9
16	1000	3	8	188	0.00206	87.6	109.9	134.0	133.7	127.6	117.6	17.3	18.4	584.9	1584.9
17	1046	2	41	161	0.00241	88.0	104.8	135.3	134.9	128.8	119.9	17.5	18.4	687.6	1733.6
18	1111	2	41	161	0.00241	90.7	115.7	135.4	135.0	129.0	120.6	17.5	18.8	714.2	1825.2

Water test for 1 tube at 5 bar abs.

Measure Volume	0.387	ltr	Tbase	20	°C
Cp	4200	kJ/kg.K			

Test no.	Power in	Time		Time	pump rate	T1	T2	T3	T4	T5	T6	T7	T8	heat water	Total watts
	watt	min	sec	sec	ltr/s	°C	°C	°C	°C	°C	°C	°C	°C	watt	watt
1	297	20	5	1205	0.00032	81.0	123.7	152.2	151.7	143.0	106.6	16.3	17.5	82.4	379.4
2	352	15	21	921	0.00042	83.2	123.9	152.1	151.6	144.1	111.3	16.3	17.6	111.6	463.6
3	400	11	3	663	0.00058	83.7	122.6	152.1	151.7	144.7	116.6	16.3	17.8	156.4	556.4
4	451	8	20	500	0.00077	84.3	121.5	152.2	151.8	144.9	121.9	16.3	18.0	209.1	660.1
5	507	6	56	416	0.00093	85.0	121.2	152.4	152.0	145.1	125.8	16.3	18.1	254.1	761.1
6	555	6	43	403	0.00096	85.0	120.5	152.5	152.1	145.3	127.2	16.3	18.3	262.5	817.5
7	610	7	30	450	0.00086	83.6	120.3	151.7	151.3	144.5	125.8	16.4	18.4	230.0	840.0
8	655	7	31	451	0.00086	82.2	119.6	151.8	151.4	144.6	125.5	16.4	18.6	224.6	879.6
9	704	7	19	439	0.00088	81.4	118.2	151.7	151.3	144.5	126.0	16.4	18.7	227.5	931.5
10	750	7	6	426	0.00091	81.0	117.5	151.8	151.3	144.6	126.6	16.5	18.9	233.0	983.0
11	804	6	35	395	0.00098	80.9	116.9	152.0	151.5	144.8	127.1	16.5	19.0	251.1	1055.1
12	854	6	14	374	0.00104	80.8	115.8	152.0	151.6	144.9	127.4	16.5	19.2	264.6	1118.6
13	904	5	32	332	0.00117	81.2	115.5	152.2	151.8	145.1	128.1	16.5	19.3	300.2	1204.2

Water test for 1 tube at 7 bar abs.

Measure Volume 0.387 ltr T0 20 °C

Cp 4200 kJ/kg.K

test no.	power in	time		Time	pump rate	T1	T2	T3	T4	T5	T6	T7	T8	heat water	total watts
	watt	min	sec	sec	ltr/s	°C	°C	°C	°C	°C	°C	°C	°C	watt	watt
1	300	26	7	1567	0.00025	91.9	134.2	162.4	161.9	152.1	106.8	16.7	17.3	74.7	374.7
2	351	20	28	1228	0.00032	92.6	133.5	162.0	161.4	153.0	113.5	16.7	17.3	96.2	447.2
3	400	14	6	846	0.00046	92.7	132.9	162.4	161.9	154.3	119.4	16.6	17.4	139.8	539.8
4	450	10	43	643	0.00060	92.4	131.9	162.8	162.3	154.8	124.8	16.6	17.5	183.3	633.3
5	500	8	13	493	0.00079	92.4	131.3	163.0	162.5	155.1	129.6	16.6	17.5	239.1	739.1
6	550	6	24	384	0.00101	91.9	130.1	162.0	161.5	154.2	134.2	16.6	17.6	304.5	854.5
7	605	7	19	439	0.00088	91.6	130.4	162.3	161.9	154.6	133.4	16.6	17.6	265.3	870.3
8	652	8	35	515	0.00075	89.8	130.2	162.3	161.8	154.6	131.9	16.5	17.7	220.7	872.7
9	702	8	30	510	0.00076	87.3	129.2	162.0	161.5	154.3	131.6	16.5	17.7	214.7	916.7
10	755	8	15	495	0.00078	85.3	128.1	161.8	161.4	154.1	132.1	16.5	17.8	214.7	969.7
11	800	8	3	483	0.00080	84.0	126.9	161.8	161.4	154.2	132.3	16.5	17.9	215.5	1015.5
12	852	7	47	467	0.00083	82.7	126.4	161.9	161.5	154.2	132.7	16.5	17.9	218.5	1070.5
13	906	7	23	443	0.00087	82.5	125.5	162.1	161.7	154.4	133.4	16.4	18.0	229.6	1135.6
14	1006	6	25	385	0.00101	82.6	124.9	162.8	162.3	155.0	134.6	16.4	18.0	264.6	1270.6

Water test for 2 tubes at 1 bar abs.

Measure Volume 0.387 ltr Tbase 20 °C

cp 4200 kJ/kg.K

test no.	Power in	time	time	Time	pump rate	T1	T2	T3	T4	T5	T6	T7	T8	heat water	total watts
	watt	min	s	sec	ltr/s	°C	°C	°C	°C	°C	°C	°C	°C	watt	Wt
1	305	3	38	218	0.0018	68.9	76.9	100.4	96.7	53.4	89.8	17.3	21.2	364.9	669.9
2	354	3	14	194	0.0020	69.5	73.0	100.2	93.5	57.2	93.4	17.1	20.9	415.4	769.4
3	402	2	41	161	0.0024	70.8	73.4	100.2	93.8	58.2	94.2	17.0	20.2	513.1	915.1
4	450	2	21	141	0.0027	72.1	74.0	100.4	95.7	58.6	94.3	1.0	19.8	601.8	1051.8
5	502	2	0	120	0.0032	73.4	73.8	100.3	94.9	59.1	95.6	17.3	19.8	724.4	1226.4
6	555	1	56	116	0.0033	75.1	73.3	100.7	97.1	59.9	96.1	17.2	19.5	747.8	1302.8
7	606	1	27	87	0.0045	76.3	80.0	100.4	96.9	60.3	97.1	17.2	19.0	1053.6	1659.6
8	651	1	37	97	0.0040	78.4	75.4	100.5	98.1	61.6	97.3	17.0	19.0	929.0	1580.0
9	710	1	14	74	0.0052	79.6	82.3	100.4	98.0	61.0	97.9	17.0	19.3	1311.3	2021.3
10	749	1	26	86	0.0045	81.1	78.6	100.2	98.9	60.0	98.2	17.1	18.9	1108.5	1857.5
11	804	1	18	78	0.0050	82.6	80.1	100.1	97.7	60.1	98.4	16.9	18.7	1254.6	2058.6
12	854	1	17	77	0.0050	84.1	80.6	100.6	99.7	60.1	98.9	17.0	18.9	1280.3	2134.3
13	898	1	17	77	0.0050	86.0	82.4	100.1	98.2	60.4	98.4	17.0	19.0	1318.0	2216.0
14	952	1	9	69	0.0056	87.8	84.7	100.2	98.1	60.1	98.5	17.1	18.9	1526.8	2478.8
15	1007	1	18	78	0.0050	89.9	84.5	100.8	100.1	60.1	99.0	17.0	19.0	1345.0	2352.0
16	1055	1	19	79	0.0049	92.2	86.4	100.1	98.9	60.2	98.4	17.0	19.0	1368.1	2423.1
17	1105	1	19	79	0.0049	93.4	87.3	100.7	100.1	60.0	98.9	16.9	19.1	1386.1	2491.1
18	1150	1	18	78	0.0050	94.7	89.1	100.1	98.7	60.0	98.3	17.0	19.4	1441.5	2591.5

Water test for 2 tubes at 3 bar abs.

Measure Volume	0.387	ltr	Tbase	20	°C
cp	4200	kJ/kg.K			

test no.	Power in	time	time	Time	pump rate	T1	T2	T3	T4	T5	T6	T7	T7	heat water	total watts
	watt	min	s	sec	ltr/s	°C	°C	°C	°C	°C	°C	°C	°C	watt	watt
1	302	12	54	774	0.00050	74.0	106.5	132.9	125.8	65.2	102.6	17.1	20.0	113.6	415.6
2	354	8	50	530	0.00073	76.0	105.5	133.5	123.2	66.7	105.8	17.2	20.2	171.8	525.8
3	406	6	50	410	0.00094	74.8	104.7	133.9	124.0	67.1	111.0	17.2	19.8	217.6	623.6
4	449	5	22	322	0.00120	75.3	103.4	134.2	123.4	68.1	114.2	17.0	19.2	279.5	728.5
5	500	4	12	252	0.00154	76.3	102.1	134.1	122.2	68.9	116.9	17.0	189.0	363.6	863.6
6	552	3	38	218	0.00178	77.7	100.1	134.0	121.6	69.3	120.1	17.0	19.0	430.7	982.7
7	599	3	24	204	0.00190	79.5	100.1	133.9	121.4	70.1	121.1	17.1	19.0	474.5	1073.5
8	658	3	17	197	0.00197	81.2	98.7	134.1	121.8	70.4	121.6	17.1	18.7	505.2	1163.2
9	701	3	7	187	0.00207	82.7	96.9	134.0	122.0	70.8	122.2	17.1	18.6	545.8	1246.8
10	758	2	54	174	0.00223	83.8	96.8	134.2	124.0	70.9	122.8	17.1	18.6	597.0	1355.0
11	800	2	44	164	0.00236	85.2	96.4	134.4	125.4	71.0	123.3	17.2	18.4	647.1	1447.1
12	860	2	25	145	0.00267	86.4	98.0	134.5	127.7	71.5	123.7	17.0	18.3	745.1	1605.1
13	909	2	24	144	0.00269	87.4	94.9	134.6	128.7	71.4	124.5	17.0	18.2	761.5	1670.5
14	955	2	13	133	0.00291	88.6	95.3	134.6	129.9	71.5	124.7	17.1	18.3	839.3	1794.3
15	1009	2	5	125	0.00310	89.9	95.6	134.9	130.9	71.7	125.4	17.0	18.2	910.2	1919.2
16	1054	2	2	122	0.00318	91.0	93.8	134.9	131.2	72.0	125.9	17.2	18.2	946.6	2000.6
17	1108	1	45	105	0.00369	92.1	98.0	135.1	132.9	72.3	125.8	17.2	18.2	1117.4	2225.4
18	1170	1	39	99	0.00391	93.0	99.5	135.1	133.1	71.9	126.2	17.1	18.2	1200.0	2370.0

Water test for 2 tubes at 5 bar abs.

Measure Volume	0.387	Ltr	Tbase	20 °C
cp	4200	kJ/kg.K		

test no.	power in	time	time	Time	pump rate	T1	T2	T3	T4	T5	T6	T7	T8	heat water	total watts
	watt	min	s	sec	ltr/s	°C	°C	°C	°C	°C	°C	°C	°C	watt	watt
1	301	22	3	1323	0.00029	85.5	123.6	151.6	140.9	67.9	105.7	16.5	19.1	80.5	381.5
2	350	11	56	716	0.00054	86.4	122.6	151.8	137.7	69.9	114.7	16.8	19.0	151.0	501.0
3	401	7	27	447	0.00087	85.5	121.4	151.6	135.7	71.8	121.6	16.5	19.0	238.3	639.3
4	450	6	30	390	0.00099	85.5	121.1	151.8	135.3	73.2	125.3	16.5	18.9	273.3	723.3
5	500	5	8	308	0.00126	85.7	119.6	151.7	135.3	74.3	128.9	16.4	18.4	347.0	847.0
6	608	4	19	259	0.00150	86.2	117.9	151.7	135.5	75.1	131.7	16.5	18.4	415.9	1023.9
7	655	3	37	217	0.00179	86.9	117.7	151.8	135.8	75.4	133.8	16.5	18.4	501.9	1156.9
8	701	3	33	213	0.00182	87.9	117.3	151.9	136.0	75.8	135.3	16.5	18.1	518.7	1219.7
9	750	3	55	235	0.00165	89.0	119.8	152.4	137.7	76.4	136.3	16.5	18.2	477.8	1227.8
10	799	4	18	258	0.00150	89.2	117.2	152.0	137.8	76.6	136.6	16.4	17.9	436.6	1235.6
11	860	4	9	249	0.00156	89.7	115.7	151.5	139.2	76.5	136.4	16.7	18.0	455.6	1315.6
12	909	3	47	227	0.00171	89.9	115.6	151.6	141.4	77.4	136.4	16.6	17.8	501.2	1410.2
13	951	3	52	232	0.00167	90.3	115.4	151.6	142.4	77.4	136.7	16.7	18.0	492.8	1443.8
14	999	3	45	225	0.00172	90.5	115.3	151.7	144.1	77.6	137.4	16.7	18.0	510.0	1509.0
15	1054	3	50	230	0.00168	90.4	114.1	151.8	145.9	77.9	138.0	16.5	17.8	498.4	1552.4
16	1109	3	23	203	0.00191	91.3	114.4	151.5	146.5	78.3	137.6	16.7	17.8	571.9	1680.9
17	1167	3	6	186	0.00208	91.6	113.7	151.5	147.7	78.6	138.3	16.6	17.5	626.7	1793.7

Water test for 2 tubes at 7 bar abs.

					Measure Volume		0.387	ltr	Tbase	20	°C				
					cp		4200	kJ/kg.K							
test no.	Power in	time	time	Time	Pump rate	T1	T2	T3	T4	T5	T6	T7	T8	heat water	total watts
	watt	min	s	sec	lt./s	°C	°C	°C	°C	°C	°C	°C	°C	watt	watt
1	349	24	68	1508	0.00026	88.6	132.3	163.2	152.4	101.0	108.6	16.3	19.0	74.0	423.0
2	401	13	27	807	0.00048	90.9	131.4	163.2	149.4	107.3	122.0	16.3	19.3	142.9	543.9
3	455	10	45	645	0.00060	91.1	130.6	162.9	147.9	111.2	125.0	16.3	18.8	179.3	634.3
4	501	8	39	519	0.00075	91.3	130.3	162.6	147.2	112.6	127.8	16.3	18.5	223.5	724.5
5	557	5	31	331	0.00117	90.7	128.9	162.9	147.9	115.2	135.0	16.2	18.3	347.6	904.6
6	608	4	34	274	0.00141	90.7	127.1	162.3	147.4	117.6	137.6	16.3	17.9	420.2	1028.2
7	661	3	51	231	0.00168	91.1	125.7	162.6	147.9	119.1	141.5	16.3	17.9	501.1	1162.1
8	704	4	5	245	0.00158	92.0	126.2	162.6	147.4	120.1	141.4	16.3	17.7	478.4	1182.4
9	750	3	36	216	0.00179	93.3	126.5	162.9	148.1	120.6	142.2	16.3	17.9	551.9	1301.9
10	799	3	11	191	0.00203	93.9	126.1	163.1	148.5	121.3	144.5	16.1	17.6	629.9	1428.9
11	860	3	56	236	0.00164	94.2	128.4	163.7	149.9	121.9	144.8	16.3	17.9	511.4	1371.4
12	907	4	12	252	0.00154	94.5	126.4	163.1	149.7	122.6	145.0	16.5	17.8	481.2	1388.2
13	955	4	23	263	0.00147	93.7	124.8	161.7	150.2	122.4	143.5	16.2	17.6	455.9	1410.9
14	1010	4	23	263	0.00147	93.2	124.3	161.9	150.9	122.3	144.1	16.5	17.5	452.8	1462.8
15	1120	4	6	246	0.00157	93.1	124.1	162.5	153.4	122.8	145.4	16.4	17.4	483.6	1603.6

Water test for 3 tubes at 1 bar abs.

Measure Volume 0.387 Tbase 20

Cp 4200

test no.	Power in	time		Time	pump rate	T1	T2	T3	T4	T5	T6	T7	T8	heat water	total watts
	watt	min	sec	sec	ltr/s	°C	°C	°C	°C	°C	°C	°C	°C	watt	watt
1	352	3	0	180	0.0022	67.2	73.4	100.4	100.0	96.6	92.3	17.1	17.9	427.1	779.1
2	402	2	34	154	0.0025	69.0	71.6	100.3	99.9	96.6	93.7	17.1	18.3	517.8	919.8
3	452	2	4	124	0.0031	70.6	71.1	100.4	100.0	96.7	94.2	17.1	18.7	663.6	1115.6
4	505	1	47	107	0.0036	73.0	70.8	100.5	100.1	96.8	95.1	17.1	19.1	771.8	1276.8
5	548	1	34	94	0.0041	75.6	72.6	100.3	99.8	96.7	95.6	17.1	19.4	911.2	1459.2
6	600	1	7	67	0.0058	86.4	82.5	100.2	99.8	96.6	97.0	17.1	19.8	1518.9	2118.9
7	654	1	2	62	0.0062	87.4	83.1	100.3	99.9	96.7	97.7	17.1	20.2	1657.0	2311.0
8	706	1	3	63	0.0061	88.0	83.8	100.4	100.1	96.8	97.8	17.1	20.6	1648.8	2354.8
9	745	0	57	57	0.0068	88.7	84.5	100.4	100.0	96.8	97.9	17.1	21.0	1840.1	2585.1
10	800	0	55	55	0.0070	89.2	85.0	100.3	99.9	96.7	98.3	17.2	21.4	1923.2	2723.2
11	848	0	52	52	0.0075	89.8	85.8	100.6	100.2	97.0	98.6	17.2	21.8	2057.7	2905.7
12	908	0	49	49	0.0079	90.4	86.3	100.5	100.1	96.9	98.7	17.2	22.1	2202.4	3110.4
13	958	0	48	48	0.0081	91.3	87.4	100.4	100.1	96.9	98.6	17.2	22.5	2284.3	3242.3
14	1015	0	48	48	0.0081	92.1	87.5	100.4	100.0	96.8	98.6	17.2	22.9	2289.6	3304.6
15	1061	0	48	48	0.0081	93.0	88.3	100.4	100.0	96.8	98.6	17.2	23.3	2315.1	3376.1
16	1112	0	46	46	0.0084	93.9	89.2	100.5	100.1	96.9	98.8	17.2	23.7	2449.5	3561.5
17	1167	0	47	47	0.0082	94.4	89.2	100.1	99.6	96.5	98.3	17.3	24.1	2395.9	3562.9
18	1230	0	46	46	0.0084	91.4	87.0	100.4	100.0	96.8	98.5	17.3	24.5	2370.9	3600.9

Water test for 3 tubes at 3 bar abs.

Measure Volume	0.387	Tbase	20
Cp	4200		

test no.	Power in	time		Time	pump rate	T1	T2	T3	T4	T5	T6	T7	T8	heat water	total watts
	watt	min	sec	sec	ltr/s	°C	°C	°C	°C	°C	°C	°C	°C	watt	watt
1	308	15	13	913	0.00042	71.3	109.0	134.0	133.5	128.2	101.3	16.3	17.2	91.5	399.5
2	350	10	5	605	0.00064	72.8	107.9	134.5	134.0	128.7	106.4	16.3	17.4	141.9	491.9
3	401	6	49	409	0.00095	73.6	106.1	134.6	134.2	129.4	111.9	16.3	17.5	213.2	614.2
4	450	5	4	304	0.00127	74.6	103.9	134.4	133.9	129.5	116.9	16.3	17.6	292.3	742.3
5	504	4	22	262	0.00148	76.2	102.4	134.1	133.7	129.4	116.3	16.3	17.8	349.0	853.0
6	554	3	44	224	0.00173	77.5	101.2	134.2	133.8	129.5	119.0	16.3	17.9	417.7	971.7
7	602	3	19	199	0.00195	79.0	100.3	134.2	133.8	129.5	119.3	16.3	18.0	482.6	1084.6
8	655	2	49	169	0.00229	80.9	98.7	134.4	134.0	129.7	122.7	16.3	18.2	586.5	1241.5
9	704	2	23	143	0.00271	83.1	97.3	134.1	133.7	129.4	122.2	16.3	18.3	717.7	1421.7
10	748	2	10	130	0.00298	84.8	97.0	133.9	133.4	129.2	122.7	16.2	18.4	810.8	1558.8
11	805	2	0	120	0.00323	86.9	96.7	133.1	132.7	128.5	123.8	16.2	18.6	907.1	1712.1
12	850	1	54	114	0.00340	89.3	98.2	133.8	133.4	129.1	124.5	16.3	18.7	988.7	1838.7
13	906	1	55	115	0.00337	90.9	96.8	134.4	134.0	129.7	125.3	16.3	18.8	1003.3	1909.3
14	957	1	44	104	0.00373	92.6	95.2	134.8	134.3	130.1	125.9	16.3	18.9	1136.5	2093.5
15	1020	1	40	100	0.00387	94.4	94.8	135.1	134.6	130.4	126.5	16.1	19.1	1211.1	2231.1
16	1060	1	33	93	0.00417	99.2	95.7	132.7	132.6	128.8	120.8	16.1	19.2	1384.9	2444.9
17	1116	1	30	90	0.00430	97.3	94.6	131.4	130.9	126.9	124.9	16.1	19.3	1397.3	2513.3
18	1180	1	15	75	0.00517	99.0	99.0	132.5	132.1	127.9	125.3	16.1	19.5	1714.7	2894.7
19	1223	1	14	74	0.00524	100.1	100.0	132.8	132.4	128.2	125.7	16.1	19.6	1762.3	2985.3

Water test for 3 tubes at 5 bar abs.

		Measure Volume	0.387	Tbase	20		
		Cp	4200				

test no.	Power in	time		Time	pump rate	T1	T2	T3	T4	T5	T6	T7	T8	heat water	total watts
	watt	min	sec	sec	ltr/s	°C	°C	°C	°C	°C	°C	°C	°C	watt	watt
1	350	14	11	851	0.00046	90.9	127.1	151.6	151.2	146.1	114.7	17.1	18.3	135.5	485.5
2	404	9	24	564	0.00069	89.8	125.5	151.4	150.9	146.0	120.6	17.1	18.4	201.3	605.3
3	459	6	50	410	0.00094	88.7	124.6	151.5	151.0	146.2	125.6	17.1	18.6	272.6	731.6
4	499	6	1	361	0.00107	88.5	124.3	151.7	151.2	146.3	127.6	17.1	18.7	308.8	807.8
5	559	4	39	279	0.00139	88.7	122.2	151.7	151.2	146.3	129.7	17.1	18.8	400.6	959.6
6	600	3	55	235	0.00165	88.9	121.0	151.8	151.3	146.4	133.9	17.1	18.9	476.8	1076.8
7	649	3	35	215	0.00180	89.5	119.9	151.9	151.5	146.5	135.9	17.1	19.1	526.1	1175.1
8	704	3	25	205	0.00189	91.0	122.0	152.3	151.8	146.9	133.3	17.1	19.2	563.9	1267.9
9	750	3	17	197	0.00197	91.1	120.6	152.0	151.6	146.7	135.7	17.1	19.3	587.6	1337.6
10	806	2	41	161	0.00241	92.8	118.4	151.2	150.8	146.0	137.4	17.1	19.5	735.7	1541.7
11	850	2	50	170	0.00228	93.9	112.1	150.3	149.9	144.7	134.9	17.1	19.6	707.5	1557.5
12	905	2	47	167	0.00232	95.3	121.6	152.2	151.7	146.9	139.0	17.1	19.7	733.5	1638.5
13	956	2	51	171	0.00227	96.4	121.4	152.0	151.6	146.8	138.8	17.2	19.8	726.6	1682.6
14	1006	2	48	168	0.00231	97.0	120.3	152.3	151.9	147.1	141.1	17.3	20.0	745.8	1751.8

Water test for 3 tubes at 7 bar

Measure Volume 0.387 Tbase 20

Cp 4200

test no.	Power in	time		Time	pump rate	T1	T2	T3	T4	T5	T6	T7	T8	heat water	total watts
	watt	min	sec	sec	ltr/s	°C	°C	°C	°C	°C	°C	°C	°C	watt	watt
1	402	16	0	960	0.00040	94.3	136.4	163.5	162.9	157.4	113.1	17.5	18.5	126.0	528.0
2	454	12	35	755	0.00051	93.2	134.9	162.9	162.4	157.0	122.6	17.6	18.5	157.8	611.8
3	505	8	7	487	0.00080	92.7	134.1	163.3	162.8	157.4	128.2	17.5	19.0	242.8	747.8
4	551	6	35	395	0.00098	92.3	132.2	163.3	162.8	157.3	130.8	17.5	19.2	297.8	848.8
5	599	5	44	344	0.00113	91.6	133.2	163.4	162.9	157.6	134.9	17.5	19.5	338.9	937.9
6	648	4	56	296	0.00131	91.6	131.4	163.1	162.6	157.3	136.4	17.5	19.8	393.5	1041.5
7	705	4	16	256	0.00151	91.7	129.9	163.2	162.7	157.5	140.9	17.5	19.8	456.0	1161.0
8	750	4	0	240	0.00161	93.2	126.2	162.5	162.0	155.8	136.5	17.5	19.9	496.6	1246.6
9	805	3	29	209	0.00185	92.9	128.1	163.1	162.6	157.5	142.1	17.5	20.1	567.4	1372.4
10	851	3	17	197	0.00197	93.9	127.1	163.4	162.9	157.7	146.3	17.4	20.2	610.0	1461.0
11	900	3	25	205	0.00189	95.0	129.6	163.2	162.7	157.5	147.2	17.6	20.4	595.1	1495.1
12	960	3	30	210	0.00184	95.8	126.0	162.6	162.2	156.9	147.1	17.5	20.5	587.3	1547.3
13	1001	2	21	141	0.00275	97.1	122.0	162.4	162.0	156.8	148.0	17.5	20.6	889.8	1890.8
14	1060	2	19	139	0.00279	99.1	125.9	163.2	162.7	157.6	148.4	17.5	20.8	925.4	1985.4
15	1150	2	9	129	0.00300	99.8	123.3	163.1	162.6	157.5	149.8	17.5	20.9	1007.0	2157.0

Appendix D

Refrigeration tests for a single lift-tube and increasing wattage

Test No.	1	2	3	4	5	6
T1	3.3	5.2	3.5	1.1	3.4	5
T2	16.7	18.6	15.2	11.7	11.7	16.3
T3	13.6	14.8	11.5	7.4	7.2	12.1
T4	14	18.1	18.6	9.3	10.4	16.8
T5	11	9.5	5.2	5.6	5.2	7.1
T6	34.9	41	38.8	41.1	43.1	39.7
T7	38.7	35.6	34.8	42.7	46.9	35.3
T8	56.3	47.7	50.9	61.8	68.6	53.4
T9	22.1	38.8	30.9	24.2	26	34
T10	28.5	50.5	43.2	35	37.2	47.4
T11	32.6	36	35.3	38	40	36
T12	41.6	66.7	61.3	48.2	49.6	69.7
P1	128.46	135.76	130	126.7	101.6	133
P2	174.96	182.02	179	177.8	175.7	183.49
P3	173.93	180.88	178	177.01	174.55	182
P4	164.5	176.27	174	168.45	170.85	178.5
P5	161.92	168.04	168	166.18	165.3	171.8
P6	125.2	129.23	135	129.12	125.84	134
P7	33.33	36.85	36.88	39.33	41.48	38
P8	60.01	52.32	57.24	66.97	74.26	60.6
Watt	443	452	500	500	547	553
time (90ml)	1.02	1.21	1.55	40	50	1.26
time sec	62	81	115	40	50	86
flow rate (ltr/s)	0.001452	0.001111	0.000783	0.00225	0.0018	0.001047
flow rate (m^3/s)	1.45E-06	1.11E-06	7.83E-07	2.25E-06	1.8E-06	1.05E-06
warmer temp diff	3	8.6	13.4	3.7	5.2	9.7
specific heat J/kg.K	4186	4186	4186	4186	4186	4186
conc m^3/kg	0.001001	0.001001	0.001001	0.001001	0.001001	0.001001
mass flow rate kg/s	0.00145	0.00111	0.000782	0.002248	0.001798	0.001046
cooling capacity watt	18.21295	39.96356	43.85889	34.81709	39.1457	42.45452
effectiveness NTU Hot	79.25189	88.20794	84.47224	78.21048	73.52609	83.10912

Refrigeration tests for a single lift-tube and increasing wattage

Test No.	7	8	9	10	11	12
T1	0.8	4.1	-1.4	-0.1	0.9	-0.4
T2	13.4	4	3.2	5.2	11.5	6.7
T3	9.6	1.2	0.7	2.6	8.1	3.7
T4	16.2	9.3	7.6	7.4	14.2	10.8
T5	5.6	2	1.7	2.3	5.4	5.3
T6	41.5	45	41.5	47.3	43.8	49.2
T7	35.8	41.6	38.2	49.2	37.5	43.3
T8	51.5	59.1	57.8	72.5	53.8	64.2
T9	32.9	30.1	28.5	28.1	46.7	33.8
T10	47.7	45.5	46.1	41.1	58.2	47.9
T11	36.3	41	36.6	42.8	37.2	42.7
T12	66.2	68.8	73	54.9	89.1	79.4
P1	133.48	113	129.22	101.2	135	131
P2	185	183	183.94	177.33	186	185
P3	184	181.6	182.84	176.13	184.9	183.9
P4	180.1	178.32	174.4	172.85	181.5	175.67
P5	172.24	170.88	172.59	167.33	174	174.03
P6	129.63	125.31	131.86	124.65	127.16	133.55
P7	38.59	44.6	39.14	44.08	40.34	44.13
P8	58.3	64.9	65.61	78.83	60.1	70.77
Watt	600	610	633	644	650	667
time (90ml)	1.11	51	38	37	53	31
time sec	71	51	38	37	53	31
flow rate (ltr/s)	0.001268	0.001765	0.002368	0.002432	0.001698	0.002903
flow rate (m^3/s)	1.27E-06	1.76E-06	2.37E-06	2.43E-06	1.7E-06	2.9E-06
warmer temp diff	10.6	7.3	5.9	5.1	8.8	5.5
specific heat J/kg.K	4186	4186	4186	4186	4186	4186
conc m^3/kg	0.001001	0.001001	0.001001	0.001001	0.001001	0.001001
mass flow rate kg/s	0.001266	0.001763	0.002366	0.00243	0.001697	0.002901
cooling capacity watt	56.19507	53.877	58.4412	51.88229	62.49676	66.78081
effectiveness NTU Hot	85.25253	83.92461	80.16486	71.80527	85.21622	79.49192

Refrigeration tests for a single lift-tube and increasing wattage

Test No.	13	14	15	16
T1	2.3	-1.4	3.9	0.2
T2	12.9	4.2	3.8	11.4
T3	10	2.1	1.5	8.8
T4	15.6	8	9.5	13.9
T5	7.1	1.5	1.7	7.1
T6	45.1	41.1	47.4	40.6
T7	34	42.9	44.1	34
T8	51.6	66.98	61	50.3
T9	46.5	31.3	31.2	33.4
T10	60	46.5	49.3	50.3
T11	37.1	36	42.8	35.7
T12	106.4	67.8	78.9	98.5
P1	135	101.65	118	137
P2	187.7	178.88	188	190
P3	186.5	177.94	186.7	188.3
P4	183.28	174.57	183.8	185.6
P5	174.85	169.14	175.36	175.6
P6	130.79	122.51	127.8	127.1
P7	40.5	38.61	49.62	39.8
P8	59.22	74.68	67.12	58.7
Watt	700	700	738	750
time (90ml)	54	40	58	35
time sec	54	40	58	35
flow rate (ltr/s)	0.001667	0.00225	0.001552	0.002571
flow rate (m^3/s)	1.67E-06	2.25E-06	1.55E-06	2.57E-06
warmer temp diff	8.5	6.5	7.8	6.8
specific heat J/kg.K	4186	4186	4186	4186
conc m^3/kg	0.001001	0.001001	0.001001	0.001001
mass flow rate kg/s	0.001665	0.002248	0.00155	0.002569
cooling capacity watt	59.2483	61.16515	50.61944	73.12932
effectiveness NTU Hot	86.06624	72.36651	86.08239	86.08247

Refrigeration tests for double lift-tubes and increasing wattage

Test No.	1	2	3	4	5	6
T1	5.7	4.6	5.4	3.2	4.0	2.5
T2	19.3	19.1	16.0	15.3	17.0	14.7
T3	15.0	14.1	12.5	9.6	12.3	9.0
T4	22.3	23.1	14.3	20.6	19.3	20.9
T5	7.8	6.7	6.2	3.3	6.2	3.2
T6	36.4	39.1	37.4	38.0	37.1	38.2
T7	36.2	39.3	40.9	41.8	41.0	42.2
T8	54.2	57.8	60.5	64.6	63.2	65.3
T9	29.4	29.6	23.3	32.5	31.1	31.9
T10	39.3	40.4	30.6	43.7	42.7	43.6
T11	33.5	34.9	35.1	33.6	32.7	33.6
T12	49.2	50.9	43.2	54.9	52.1	55.4
P1	129.0	129.4	99.5	125.7	123.5	125.3
P2	176.6	178.3	174.3	178.0	176.8	178.3
P3	164.4	175.7	170.0	176.7	175.5	177.1
P4	170.8	172.2	167.9	173.4	172.3	173.8
P5	165.2	167.8	160.3	169.8	167.5	169.2
P6	125.9	129.7	123.1	128.6	125.5	126.6
P7	34.8	36.6	36.2	36.7	35.4	36.7
P8	60.5	62.6	64.7	72.3	70.6	73.3
Watt	453	496	508	548	597	642
time (90ml)	3.1	3.23	1.29	4.21	2.35	
time sec	190	203	89	261	155	180
flow rate (ltr/s)	0.000474	0.000443	0.001011	0.000345	0.000581	0.0005
flow rate (m^3/s)	4.74E-07	4.43E-07	1.01E-06	3.45E-07	5.81E-07	5E-07
warmer temp diff	14.5	16.4	8.1	17.3	13.1	17.7
specific heat J/kg.K	4186	4186	4186	4186	4186	4186
conc m^3/kg	0.001001	0.001001	0.001001	0.001001	0.001001	0.001001
mass flow rate kg/s	0.000473	0.000443	0.00101	0.000345	0.00058	0.0005
cooling capacity watt	28.73	30.41	34.26	24.95	31.81	37.01
effectiveness NTU Hot	80.29	80.20	77.06	73.23	73.35	72.37

Refrigeration tests for double lift-tubes and increasing wattage

Test No.	7	8	9	10	11	12
T1	2.8	-0.2	2.5	1.0	-0.8	1.5
T2	13.7	8.4	14.2	14.8	8.2	9.3
T3	8.9	3.6	7.7	9.3	4.4	6.7
T4	18.5	10.4	21.3	18.4	16.6	14.5
T5	5.6	1.7	2.5	4.8	1.1	4.1
T6	38.8	39.3	41.3	42.7	42.3	42.1
T7	38.7	41.1	45.2	46.1	41.7	37.7
T8	60.0	63.8	69.8	70.9	65.4	59.6
T9	32.7	27.0	37.1	39.8	39.7	46.7
T10	46.5	40.0	49.9	52.6	55.9	62.7
T11	32.4	35.3	35.6	37.1	36.7	34.8
T12	63.8	53.9	66.6	72.0	85.6	110.5
P1	124.6	103.7	126.5	128.0	130.1	130.6
P2	179.1	178.4	181.0	182.7	186.6	187.9
P3	177.3	175.7	180.0	181.5	185.5	186.7
P4	174.3	173.3	177.2	178.6	182.9	184.6
P5	169.3	168.4	172.8	173.6	177.1	177.4
P6	128.4	129.6	130.3	130.7	133.0	134.0
P7	35.4	36.6	39.2	41.8	40.4	38.6
P8	69.4	70.7	78.1	79.2	74.0	67.9
Watt	654	675	718	759	790	845
time (90ml)	2.4	1.1	4.17	2.01	1.54	1.08
time sec	160	70	257	121	114	68
flow rate (ltr/s)	0.000563	0.001286	0.00035	0.000744	0.000789	0.001324
flow rate (m^3/s)	5.63E-07	1.29E-06	3.5E-07	7.44E-07	7.89E-07	1.32E-06
warmer temp diff	12.9	8.7	18.8	13.6	15.5	10.4
specific heat J/kg.K	4186	4186	4186	4186	4186	4186
conc m^3/kg	0.001001	0.001001	0.001001	0.001001	0.001001	0.001001
mass flow rate kg/s	0.000562	0.001285	0.00035	0.000743	0.000789	0.001322
cooling capacity watt	30.35	46.78	27.53	42.31	51.18	57.57
effectiveness NTU Hot	74.62	74.10	70.91	71.64	75.39	78.87

Refrigeration tests for double lift-tubes and increasing wattage

Test No.	13	14	15	16	17
T1	1.3	2.8	6.7	3.7	7.8
T2	6.3	8.5	9.9	6.1	8.7
T3	3.7	5.5	6.9	4.3	6.1
T4	10.6	16.8	18.8	9.4	18.8
T5	2.9	3.0	4.8	4.0	4.8
T6	47.6	42.4	43.7	43.4	44.6
T7	51.6	36.5	36.7	44.4	38.2
T8	75.7	58.6	59.2	70.2	60.4
T9	32.5	49.7	54.3	47.6	48.3
T10	45.8	68.0	78.1	63.6	74.1
T11	41.9	34.7	35.5	36.9	37.1
T12	67.9	122.5	130.0	103.5	132.5
P1	111.6	128.0	136.5	111.8	131.0
P2	179.9	188.3	189.4	189.6	191.5
P3	178.6	187.2	188.2	184.7	190.4
P4	175.5	184.6	185.7	182.2	187.5
P5	171.2	177.4	178.7	176.2	180.6
P6	126.6	125.4	129.6	127.2	126.3
P7	43.8	38.8	39.9	39.9	40.6
P8	82.6	67.0	67.6	79.4	67.4
Watt	872	901	949	997	1000
time (90ml)	55	1.5	2.36	35.9	3
time sec	55	110	156	36	180
flow rate (ltr/s)	0.001636	0.000818	0.000577	0.0025	0.0005
flow rate (m^3/s)	1.64E-06	8.18E-07	5.77E-07	2.5E-06	5E-07
warmer temp diff	7.7	13.8	14	5.4	14
specific heat J/kg.K	4186	4186	4186	4186	4186
conc m^3/kg	0.001001	0.001001	0.001001	0.001001	0.001001
mass flow rate kg/s	0.001635	0.000817	0.000576	0.002498	0.0005
cooling capacity watt	52.70	47.22	33.78	56.46	29.28
effectiveness NTU Hot	69.54	79.67	80.03	71.04	80.84

Refrigeration tests for triple lift-tubes and increasing wattage

Test No.	1	2	3	4	5	6
T1	2.9	3.1	0.4	-0.4	2.7	1.5
T2	15.4	15.4	10.6	12.5	13.7	10.9
T3	11.3	10.6	5.8	6.9	8.1	6.2
T4	16.9	19.1	12.2	11.8	20.6	13.3
T5	5.5	4.1	2.9	5.8	2.5	3.6
T6	35.3	36.8	36.1	42.8	39.2	39.7
T7	34.4	38.5	37.2	44.2	42.8	42.7
T8	53.9	61.5	58.6	68.4	65.1	66.6
T9	25.8	29.9	32.3	28.3	30.7	32.1
T10	36.9	48.1	45.4	41.7	41.8	45.8
T11	34.1	32.7	33.2	38.9	34.3	36.3
T12	48.8	49.9	61.3	54.6	51.6	64.2
P1	125.6	123.9	96.2	124.47	122.6	102
P2	177.5	176.5	173.3	181.3	177.8	173.34
P3	171.75	174.3	172.3	173	176.8	172.25
P4	171.18	171.9	168.1	172.92	171.6	168.81
P5	165.65	166.22	163.7	170.32	167.38	164.63
P6	125.77	131.07	122.5	127.61	125.24	123.1
P7	33.16	34.06	36.6	40.87	37.02	39.33
P8	59.45	64.5	67.5	75.2	73.48	74.06
Watt	517	556	598	609	613	653
time (90ml)	1.54	3.13	62	44	3.51	67
time sec	114	193	62	44	231	67
flow rate (ltr/s)	0.000789	0.000466	0.001452	0.002045	0.00039	0.001343
flow rate (m^3/s)	7.89E-07	4.66E-07	1.45E-06	2.05E-06	3.9E-07	1.34E-06
warmer temp diff	11.4	15	9.3	6	18.1	9.7
specific heat J/kg.K	4186	4186	4186	4186	4186	4186
conc m^3/kg	0.001001	0.001001	0.001001	0.001001	0.001001	0.001001
mass flow rate kg/s	0.000789	0.000466	0.00145	0.002044	0.000389	0.001342
cooling capacity watt	37.64009	29.25396	56.46014	51.3274	29.49289	54.49387
effectiveness NTU Hot	80.15699	76.96731	75.68843	73.48011	72.0313	72.28252

Refrigeration tests for triple lift-tubes and increasing wattage

Test No.	7	8	9	10	11	12
T1	2.5	0.6	-0.1	-4.0	-0.6	0.1
T2	13.4	9.8	8.2	6.4	7.4	8.3
T3	7.4	4.0	5.0	2.0	4.0	4.7
T4	19.3	17.5	11.9	10.4	13.7	13.6
T5	2.7	1.8	3.2	2.5	1.9	1.6
T6	40.8	43.8	41.9	42.1	43.0	43.6
T7	44.5	45.1	44.5	41.5	44.0	44.2
T8	68.8	70.1	68.4	65.6	70.6	69.0
T9	28.6	37.3	31.3	32.5	31.7	34.0
T10	40.2	48.4	45.6	47.5	45.9	48.6
T11	35.7	37.2	37.0	37.7	37.6	37.3
T12	51.2	62.5	66.3	72.5	68.3	67.3
P1	123.0	123.7	99.2	126.1	102.3	126.1
P2	177.7	177.7	175.9	185.3	176.4	179.9
P3	176.8	176.8	175.0	175.0	175.5	179.0
P4	172.5	172.6	171.1	177.6	171.7	173.0
P5	168.5	168.1	167.0	174.8	167.8	170.0
P6	125.8	126.2	125.4	131.3	125.1	126.0
P7	39.2	41.7	41.2	39.8	42.0	42.2
P8	76.4	78.6	75.5	72.2	78.3	77.1
Watt	656	690	698	704	725	743
time (90ml)	2.44	2.12	56	41	78	1.01
time sec	164	132	56	41	78	61
flow rate (ltr/s)	0.000549	0.000682	0.001607	0.002195	0.001154	0.001475
flow rate (m^3/s)	5.49E-07	6.82E-07	1.61E-06	2.2E-06	1.15E-06	1.48E-06
warmer temp diff	16.6	15.7	8.7	7.9	11.8	12
specific heat J/kg.K	4186	4186	4186	4186	4186	4186
conc m^3/kg	0.00100	0.00100	0.00100	0.00100	0.00100	0.00100
mass flow rate kg/s	0.00055	0.00068	0.00161	0.00219	0.00115	0.00147
cooling capacity watt	38.10	44.77	58.48	72.53	56.94	74.05
effectiveness NTU Hot	71.22	70.82	72.71	76.00	71.12	72.64

Refrigeration tests for triple lift-tubes and increasing wattage

Test No.	13	14	15	16	17	18
T1	-0.6	0.4	-1.4	-4.5	-0.4	1.1
T2	7.6	7.6	9.0	5.5	5.6	6.0
T3	4.4	4.6	4.3	1.6	3.4	3.6
T4	13.8	13.4	12.2	10.0	11.3	12.5
T5	2.2	1.9	5.2	2.7	2.3	2.1
T6	44.1	42.8	41.5	42.1	45.6	49.0
T7	46.9	45.2	41.6	45.3	45.5	44.1
T8	71.9	69.0	69.7	71.2	67.9	69.3
T9	32.9	37.0	34.0	38.4	45.0	50.1
T10	47.9	51.5	49.8	52.9	58.4	62.2
T11	38.6	38.2	37.2	37.9	39.2	41.0
T12	71.3	71.4	69.0	76.2	83.0	93.4
P1	102.3	104.1	124.6	120.9	102.5	100.6
P2	176.8	176.5	185.3	184.5	180.5	180.7
P3	176.0	175.7	176.3	177.0	179.6	179.8
P4	172.3	171.9	177.1	177.9	175.8	176.3
P5	168.3	168.1	174.0	176.3	170.8	171.3
P6	125.9	126.3	128.6	131.2	128.6	127.0
P7	43.8	42.8	40.2	39.2	43.6	45.0
P8	79.8	77.1	75.2	82.3	75.5	77.2
Watt	745	753	789	791	800	848
time (90ml)	73	58	38	40	58	75
time sec	73	58	38	40	58	75
flow rate (ltr/s)	0.001233	0.001552	0.002368	0.00225	0.001552	0.0012
flow rate (m^3/s)	1.23E-06	1.55E-06	2.37E-06	2.25E-06	1.55E-06	1.2E-06
warmer temp diff	11.6	11.5	7	7.3	9	10.4
specific heat J/kg.K	4186	4186	4186	4186	4186	4186
conc m^3/kg	0.001001	0.001001	0.001001	0.001001	0.001001	0.001001
mass flow rate kg/s	0.001232	0.00155	0.002366	0.002248	0.00155	0.001199
cooling capacity watt	59.81	74.63	69.34	68.69	58.41	52.19
effectiveness NTU Hot	71.09	72.63	73.81	68.59	74.89	74.53

Refrigeration tests for triple lift-tubes and increasing wattage

Test No.	19	20	21	22	23	24
T1	-0.8	0.4	0.4	1.5	-1.6	1.9
T2	7.9	5.0	4.4	4.3	6.6	6.0
T3	3.9	2.8	2.5	2.9	3.1	3.8
T4	12.1	10.4	8.1	9.4	12.1	11.5
T5	4.9	1.5	1.9	2.1	4.2	3.0
T6	40.8	47.3	45.5	46.5	43.5	45.9
T7	46.6	42.3	42.1	42.2	47.1	43.5
T8	75.8	67.2	68.5	68.2	77.8	69.8
T9	30.6	39.4	36.7	39.4	27.4	34.4
T10	45.0	57.1	54.7	57.1	42.4	52.2
T11	36.5	39.3	38.0	39.0	37.2	38.2
T12	75.1	90.7	86.2	88.6	68.2	82.2
P1	119.0	101.1	95.5	96.8	119.4	101.9
P2	183.0	181.6	181.8	182.0	183.6	181.3
P3	178.9	180.6	180.9	181.0	180.5	180.3
P4	176.3	177.2	177.5	177.5	176.6	177.1
P5	174.4	171.8	172.3	172.3	175.8	171.5
P6	128.4	124.7	124.4	123.9	124.3	121.7
P7	40.2	44.2	43.0	43.8	41.6	44.2
P8	89.6	76.3	78.4	77.9	90.1	84.0
Watt	893	920	915	940	995	1000
time (90ml)	40.6	58	38	43	43.35	48
time sec	40.6	58	38	43	43.35	48
flow rate (ltr/s)	0.00222	0.00155	0.00237	0.00209	0.00208	0.00188
flow rate (m^3/s)	2.22E-06	1.55E-06	2.37E-06	2.09E-06	2.08E-06	1.88E-06
warmer temp diff	7.2	8.9	6.2	7.3	7.9	8.5
specific heat J/kg.K	4186	4186	4186	4186	4186	4186
conc m^3/kg	0.001001	0.001001	0.001001	0.001001	0.001001	0.001001
mass flow rate kg/s	0.002215	0.00155	0.002366	0.002091	0.002074	0.001873
cooling capacity watt	66.75	57.76	61.41	63.90	68.59	66.65
effectiveness NTU Hot	63.18	74.88	72.65	73.41	63.85	68.79

Refrigeration tests for triple lift-tubes and increasing wattage

Test No.	25	26	27
T1	2.4	6.7	5.0
T2	6.0	1.1	5.9
T3	3.5	3.9	4.0
T4	12.4	11.7	9.7
T5	2.4	5.5	3.9
T6	45.8	45.3	44.4
T7	42.9	43.0	41.5
T8	68.6	78.0	68.1
T9	36.3	37.7	41.9
T10	55.0	51.9	63.0
T11	39.2	38.3	37.0
T12	90.1	88.9	105.6
P1	97.7	120.2	96.7
P2	182.4	185.4	183.7
P3	181.5	181.6	182.8
P4	178.1	178.7	179.8
P5	171.9	177.1	175.6
P6	125.7	126.3	128.9
P7	44.1	42.1	43.7
P8	82.9	91.8	80.9
Watt	1052	1060	1150
time (90ml)	61	36	42
time sec	61	36	42
flow rate (ltr/s)	0.001475	0.0025	0.002143
flow rate (m^3/s)	1.48E-06	2.5E-06	2.14E-06
warmer temp diff	10	6.2	5.8
specific heat J/kg.K	4186	4186	4186
conc m^3/kg	0.001001	0.001001	0.001001
mass flow rate kg/s	0.001474	0.002498	0.002141
cooling capacity watt	61.71	64.82	51.98
effectiveness NTU Hot	69.64	63.16	71.79

Appendix E

Example of the spreadsheet layout used during the prediction model calculation

Stenning and Martin, and Lister, adapted for increased system pressure

Constants and known's						
inch	1		m	0.0254		volumes
cf	1		m^3	0.028316	100 kg liquid	0.1108
cfm	1		m^3/s	0.000471933	3 kg vapour	0.8187
H	0.0654		initial head (m)		vol ratio	7.38898917
g	9.81		gravitational acceleration (m/s^2)			
L	0.445		Pump length (m)			
f	0.008		friction factor			
s	4		can go up to 2.5 for efficient running			
D	0.006		diameter of the pipe (m)			
Ps	700000		abs. system pressure (pa)			
Tb	164.97		Boiling temperature (deg C)			
ro water	902.5270758		density of water at saturated conditions (kg/m^3)			
vf	0.001108		specific liquid volume at saturation(m^3/kg)			
vg	0.2729		specific vapour volume at saturation(m^3/kg)			
hfg	2066.3		enthalpy for vapourisation (kJ/kg)			
CpDeltaT	624.0633354		(kJ/kg)			
T1	20		inlet Temperature in (Deg C)			
surf tension	0.05					
D boil	0.066	(m)				
D elm	0.02	(m)				
Effected height	0.1	(m)				
pre calcs						
n	3		desired number of tubes			
K	9.493333333		loss constant			
A	8.4823E-05		Area of pipe (m^2)			
P1	700579.0379		total static pressure at the inlet (pa)			
P1/2	700289.519		Mid tube pressure (pa)			
ro vapour	3.664345914		density of vapour (kg/m^3)			
bubble vel	0.233369667	gas bubble velocity in the boiler (m/s)				
area boiler	3.11E-03	m^2				
Actual math						
water vapour middle	volume water exiting	desired H/l ratio	H/L ratio	vapour released	heat input boiling	heating input
(m^3/s)	(m^3/s)			kg/s	watt	watt
0.000001	4.30841E-08	1.47E-01	0.14701549	3.66435E-06	7.571637963	26.5532209
0.000005	2.15544E-07	1.47E-01	0.14730237	1.83217E-05	37.85818981	132.8354444
0.00002	8.58614E-07	1.49E-01	0.14928986	7.32869E-05	151.4327593	529.336053

Appendix F

Dimensions of the water rig