รายงานวิจัยฉบับสมบูรณ์

A study of a Diffusion Absorption Refrigerator

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กุมภาพันธ์ 2545

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งานวิจัยนี้เป็นการศึกษาและปรับปรุงระบบทำความเย็นแบบ Diffusion Absorption ที่ได้จัดสร้างขึ้นและสามารถทำงานได้แล้ว โดยทำการศึกษาถึงตัวแปรหลักที่มีผลกระทบต่อการทำงานของระบบ เพื่อเป็นพื้นฐานในการออกแบบสำหรับการใช้งานจริงต่อไปในอนาคต โดยทำการแปรค่าของส่วนที่น่าจะส่งผลต่อการทำงาน คือ ค่าปริมาณความร้อน input ที่ generator ค่าปริมาณของก๊าซเฉื่อยที่อัดเข้าไปในระบบ และค่าของอุณหภูมิทำงานที่ rectifier และในการศึกษานี้ได้มีการสร้างแบบจำลองทางคณิตศาสตร์อย่างง่ายขึ้นเพื่อใช้ประโยชน์ในการคำนวณหาค่าสมรรถนะสูงสุดของระบบขึ้นด้วย เพื่อใช้ประโยชน์ในการปรับปรุงระบบให้มีสมรรถนะที่ดีขึ้น

จากผลการทดลองที่ได้พบว่า การแปรค่าปริมาณความร้อนที่ generator ส่งผลกระทบโดยตรงต่อการทำงานของระบบเนื่องจากเป็นตัวแปรสำคัญที่กำหนดอัตราการกำเนิดไอสารทำงานอันเป็นส่วนสำคัญต่อการหมุนเวียนสารทำงานโดยbubble pump นอกจากนี้การแปรค่าของปริมาณก๊าซเฉื่อยก็ส่งผลกระทบโดยตรงต่ออุณหภูมิใน generator เพราะค่าอุณหภูมิ (saturation temperature) จะแปรไปตามความดันที่เปลี่ยนแปลงไป และยังพบว่าการอัดก๊าซเฉื่อยควรกระทำที่ค่าซึ่งเหมาะสม มิเช่นนั้นแล้วระบบอาจจะไม่ทำงาน หรือทำงานด้วยอุณหภูมิที่สูงเกินไป ส่วนการแปรค่าอุณหภูมิ rectifier นั้นไม่พบการเปลี่ยนแปลงของการทำงานที่เด่นชัด ซึ่งอาจเกิดจากขีดจำกัดในการทำงานของ evaporator หรือ absorber แต่ยังไม่สามารถระบุลงไปได้อย่างชัดเจน ผลที่ได้จากการคำนวณโดยแบบจำลองทางคณิตศาสตร์ สามารถช่วยแสดงให้เห็นว่า ค่าสมรรถนะของระบบที่ค่อนข้างต่ำน่าจะมีสาเหตุมาจาก ความไม่เหมาะสมของปริมาณไอสารทำความเย็นที่ใช้ขับbubble pump กับการดูดซึมไอสารทำความเย็นโดย absorber นอกจากนี้ผลจากการคำนวณยังสามารถใช้ช่วยในการระบุแหล่งของความร้อนทิ้งที่อาจจะนำมาใช้เพื่อช่วยเพิ่มสมรรถนะของระบบได้อีก เช่น ความร้อนทิ้งจาก rectifier หรือการเพิ่มปริมาณความร้อนแลกเปลี่ยนที่ solution heat exchanger เป็นต้น

จากผลการศึกษานี้พบว่าระบบ Diffusion Absorption สามารถที่จะพัฒนาให้มีสมรรถนะในการทำงานที่ดีขึ้นได้ โดยปรับปรุงอุปกรณ์บางส่วนของระบบเช่น ออกแบบ bubble pump ใหม่ให้สามารถทำงานได้สัมพันธ์กับabsorber ซึ่งจะส่งผลให้เกิดการใช้ความร้อนเพื่อการกำเนิดไอสารทำความเย็นอย่างมีประสิทธิภาพ ปรับปรุง absorber ให้สามารถดูดซึมไอสารทำความเย็นได้มากขึ้น หรือปรับปรุง evaporator ให้มีความสามารถในการระเหยไอสารทำความเย็นได้ในปริมาณที่สูงขึ้นแต่ต้องทำงานให้สัมพันธ์กับabsorber ที่จะออกแบบขึ้นใหม่นี้ด้วย

Keywords : Diffusion, Absorption, Refrigeration, Bubble pump

Output จากโครงการวิจัยที่ได้รับทุนจาก สกว.

1. ผลงานตีพิมพ์ในวารสารวิชาการระดับนานาชาติ
   Srikhirin, P. and Aphornratana, S, An Experimental investigation of a Diffusion Absorption Refrigerator, Applied Thermal Engineering
   (กำลังรอผลตอบรับอย่างเป็นทางการจากบรรณาธิการ)

2. ได้ช่วยสร้างนักวิจัยใหม่ในระดับปริญญาเอก ซึ่งปัจจุบันสำเร็จการศึกษาแล้ว
   (นายพงษ์สิทธิ์ ศรีคิรินทร์)

ABSTRACT

A study of a Diffusion Absorption Refrigerator

A diffusion absorption refrigeration (DAR) system is a type of vapor absorption refrigeration system. It is a truly heat operated refrigeration system which requires only heat for its operation. Its working fluid contains a working fluid pair; ammonia as refrigerant, water as absorbent and an auxiliary gas. Hydrogen is normally used as an auxiliary gas. However, in this study, helium is used as the auxiliary gas for a reason of safety. Its operating performance is quite low compared with a simple absorption system. It can be easily used and requires less maintenance since no moving part is contained in the system. Its operation is quiet too.

In this study, the DAR was studied both in theoretical and experimental aspects. An experimental refrigerator was designed and fabricated. Its configuration was arranged in a manner that the operating parameters of the system could be separated. Heat input at the generator, rectification temperature, condensation temperature, and auxiliary gas charge pressure could be varied individually. A mathematical model was also constructed as a primary tool for system analysis. To calculate the model, some operating conditions are required. The model is simple which could be calculated manually. It can be used to identify location of heat loss occurring in the system components. It helps to improve the system performance more pertinent. Comparisons of the actual tested results and the calculation results are presented. Operating characteristic of the DAR was dependent on the bubble pump characteristics. However, its performance relied on the absorption and evaporation capability of the absorber and the evaporator respectively. System improvement options were recommended for implementation of a modified system in the future.

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Chapter	Title	Page

NOMENCLATURE

aux	auxiliary gas
c	ammonia concentration (kmol m^{-3})
c_p	specific heat capacity (kJ $kg^{-1}K^{-1}$)
COP	coefficient of performance
D	molecular diffusivity (m^2 s^{-1})
DAR	diffusion absorption refrigeration or diffusion absorption refrigerator
g	acceleration due to gravity (m s^{-2})
h	specific enthalpy (kJ kg^{-1})
H_2	hydrogen
H_2O	water
He	helium
HR	head ratio
ID	inside diameter (in., mm.)
$\dot{m}$	mass flow rate (kg s^{-1})
NH_3	ammonia
OD	outside diameter (in., mm.)
P	pressure (bar, MPa)
Q_H	high temperature heat (kJ)
Q_L	low temperature heat (kJ)
Q_s	surrounding temperature heat (kJ)
$\dot{Q}_{in}$	input power (kW)
Re	Reynolds number
T	temperature (°C)
x,X	concentration
Z	elevation (m)

GREEK LETTERS

δ	film thickness (m)
ρ	density (kg m^{-3})
Δ	difference
Γ	mass flow rate per unit width (kg $m^{-1}s^{-1}$)
μ	viscosity (kg $m^{-1}s^{-1}$)
ε	combined evaporator absorber effectiveness

SUBSCRIPTS

amm	ammonia
con	condenser
evap	evaporator
liq	liquid phase
rec	rectifier
vap	vapor phase

CHAPTER I

Introduction

Most industrial processes use a lot of thermal energy by burning fossil fuel to produce steam or heat for various purposes. After the processes, an amount of heat is rejected to the surrounding as waste. The waste heat can be converted to useful refrigeration by using a heat operated refrigeration system, such as an absorption refrigeration cycle. As a result electricity purchased from utility companies for conventional vapor compression refrigerators can be reduced. Moreover, the use of heat operated refrigeration systems helps reduce problems related to the global environment, such as the so called greenhouse effect from CO_2 emission from the combustion of fossil fuels in utility power plants.

Another difference between absorption systems and conventional vapor compression systems is the working fluid used. Most vapor compression systems commonly use chlorofluorocarbon refrigerants (CFCs). The use of CFCs is restricted throughout the globe due to depletion of the ozone layer. This makes absorption systems more prominent. Although absorption systems seem to provide many advantages, vapor compression systems still dominate all market sectors. In order to promote the use of absorption systems, further development is required to improve their performance and reduce cost.

The early development of an absorption cycle dated back to the 1700's. It was known that ice could be produced by evaporation of pure water from a vessel contained within an evacuated container in the presence of sulfuric acid [Herold and Radermacher, 1989]. As the acid absorbed water vapor, causing a reduction of temperature, layers of ice were formed on the water surface. The major problems of this system were corrosion and leakage of air into the vacuum vessel. In 1859, Ferdinand Carre' introduced a novel machine using water/ammonia as the working fluid. This machine shown in Figure 1.1 was granted a US patent in 1860 [Gosney, 1982]. Machines based on this patent were used to make ice and store food. It was used as a basic design in the early age of refrigeration development.

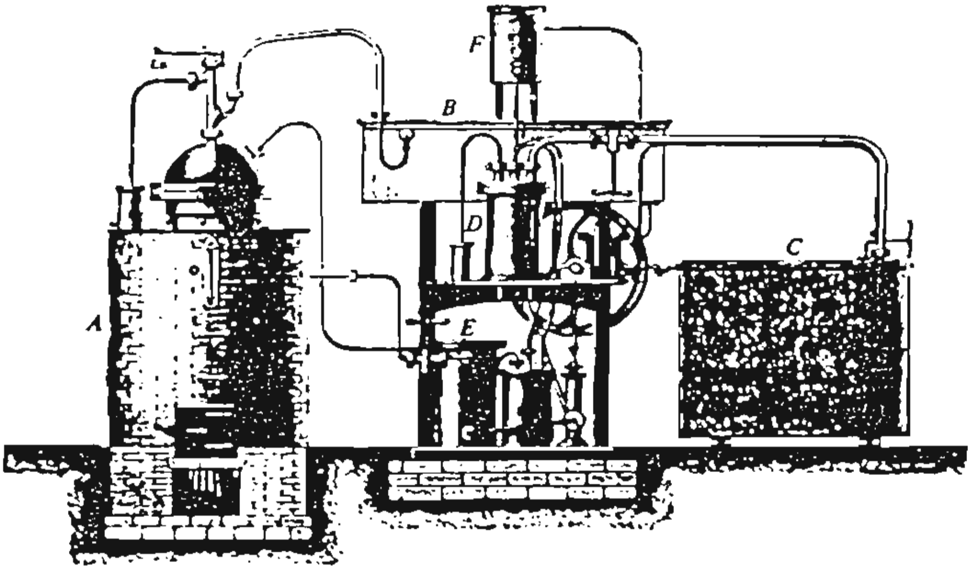

Figure 1.1 Carre' continuous absorption machine. *A* : boiler, *B* : condenser *C* : evaporator *D* : absorber *E* : solution heat exchanger and *F* : header tank for cooling water (Gosney, 1982).

In 1950's, a system using lithium-bromide/water as the working fluid was introduced for industrial applications. A few years later, a double-effect absorption system was introduced and has been used as an industrial standard for a high performance heat-operated refrigeration cycle. Various types of absorption refrigeration cycles have been discussed in literature [Srikhirin et al., 2001; Eames and Aphornratana, 1993].

Considering a system containing two vessels connected to each other as shown in Figure 1.2. Inside the left vessel, there is liquid refrigerant. The other vessel contains a binary solution of absorbent/refrigerant. The solution in the right vessel will absorb refrigerant vapor from the left vessel. While the refrigerant vapor is being absorbed, the temperature of the remaining refrigerant will reduce as a result of its vaporization.

Refrigeration effect occurs inside the left vessel. At the same time the solution inside the right vessel becomes more diluted because of the higher content of absorbed refrigerant. This is called "absorption process".

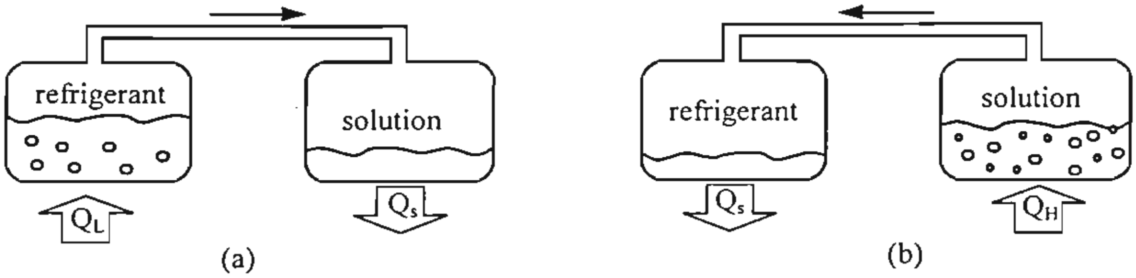

Figure 1.2 a) absorption process and b) refrigerant separation process.

On the other hand, when the refrigerant is absorbed until the solution cannot continue the absorption process, it must be separated out from the diluted solution. Heat is the key of the separation process. It is applied to the right vessel in order to expel the refrigerant from the solution. Transferring heat to the surroundings causes condensation of the refrigerant vapor. With these processes, the refrigeration effect can be produced using thermal energy. However, this cannot be done continuously as the process cannot be done simultaneously. Therefore, an absorption refrigeration cycle is a combination of these two processes as shown in Figure 1.3. As the separation process occurs at a higher pressure than the absorption process, a circulation pump is required to circulate the solution. Coefficient of Performance of an absorption refrigeration system is obtained from;

$$COP = \frac{\text{cooling capacity obtained at evaporator}}{\text{heat input for the generator} + \text{work input for the pump}} \tag{1.1}$$

The work input for the pump is negligible relative to the heat input at the generator, therefore, the pump work is often neglected in the analysis.

Using heat as primary energy input, a simple vapor absorption cycle has a relatively low COP comparing with the vapor compression cycle. Even the prime energy for the absorption system is in a form of heat, electricity still being required to drive a circulation pump. However, there is a type of absorption cycle that does not require any circulation pump. In such a system, working fluid is circulated by the thermo-siphon effect known as a bubble pump. The noted examples are water chillers manufactured by Yazaki and domestic refrigerators manufactured by Electrolux.

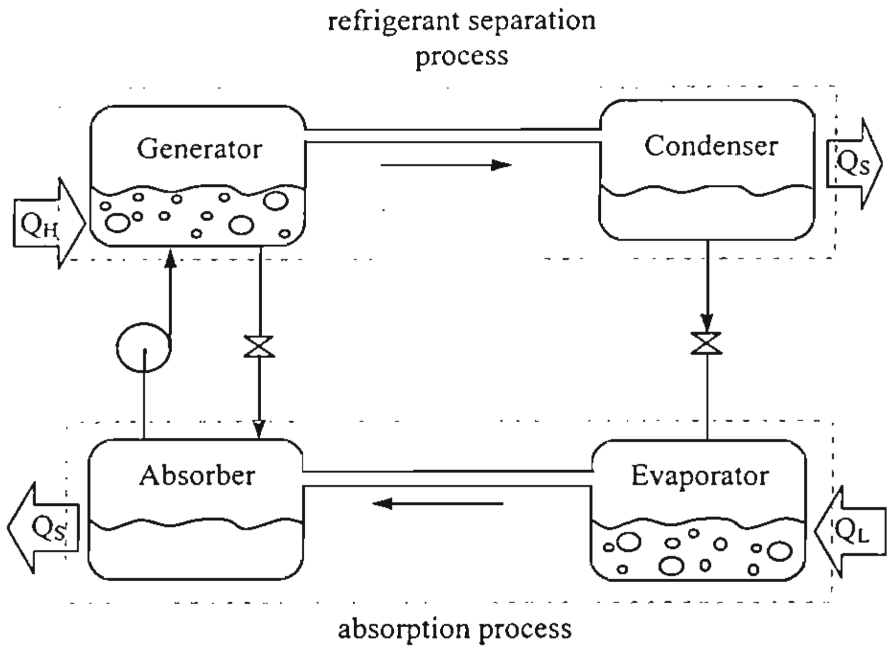

Figure 1.3 Vapor absorption refrigeration cycle.

Yazaki Inc. of Japan introduced a self-circulating absorption refrigeration system based on a single-effect system using lithium-bromide/water. Using water as a refrigerant, differential pressure between the condenser and the evaporator is very low and can be maintained by using the principle of hydrostatic head. Solution from the absorber can be circulated through the generator by a bubble pump. The weak refrigerant solution returns gravitationally back to absorber. A schematic diagram of this system is shown in Figure 1.4. With the effect of bubble pump, the solution is boiled and pumped simultaneously.

As water is the refrigerant, the cooling temperature is limited to be above freezing point (0°C). The entire system is operated under vacuum condition, therefore leak of air into the system is not easy to avoid. The system requires cooling water for its condenser and absorber in order to prevent crystallization of lithium-bromide.

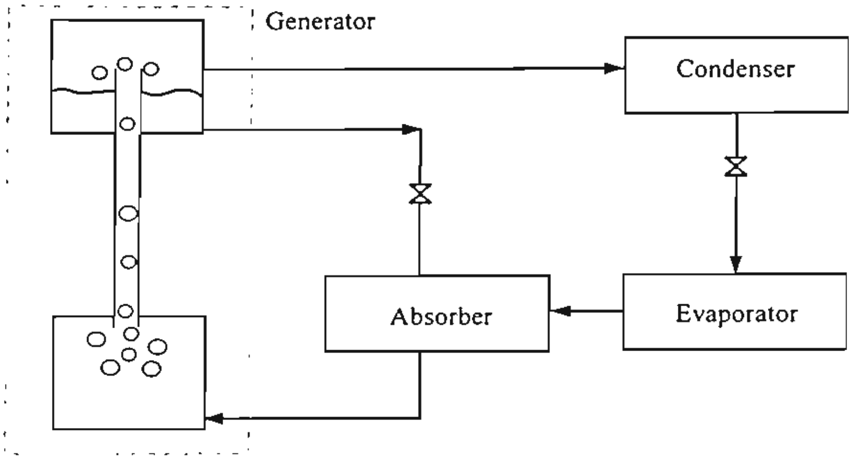

Figure 1.4 A schematic diagram of the Yazaki system vapor absorption cycle.

The Electrolux system is generally known as a Diffusion Absorption Refrigeration (DAR) system. Figure 1.5 shows a schematic diagram of a domestic refrigerator. The DAR is a type of self-circulating absorption system using water/ammonia. As ammonia is the working fluid, differential pressure between the condenser and the evaporator is too large to be overcome by a bubble pump. An auxiliary gas is charged into the evaporator and the absorber. There is no pressure differential in the system making use of the bubble pump possible. The obtained cooling effect is based on Dalton's principle of partial pressure. Because the auxiliary gas is charged into the evaporator and the absorber, the partial pressure of ammonia in both evaporator and absorber is kept low enough to correspond with the temperature required inside the evaporator. The auxiliary gas should be noncondensable such as hydrogen or helium.

The DAR can operate without any use of electrical and mechanical energy. As there is no moving part, system maintenance, noise, and vibration are at minimum. This system has been used for more than 70 years. Millions of such refrigerators have been built and are used mainly in domestic applications and for use in camping and caravans. They can be powered by kerosene or liquid petroleum gas. Electrically powered units are also available, which are suitable for quiet places such as hotel rooms.

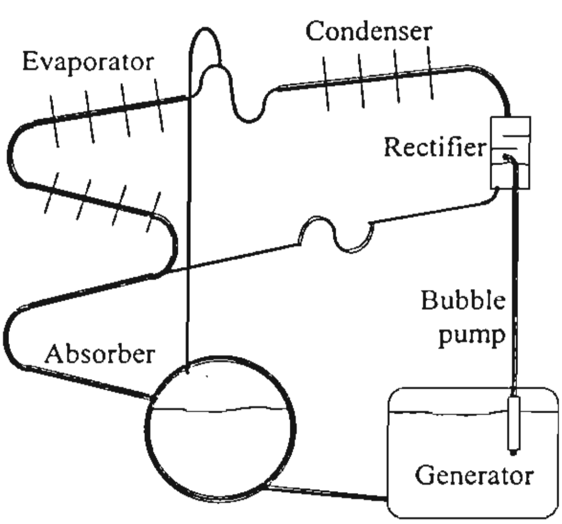

Figure 1.5 A schematic diagram of DAR.

Even though this system has been used for a long time, its application is limited to small refrigerators only. Its efficiency is relatively poor. Normally, a refrigerator based on this system provides cooling capacity up to 200 W with COP of 0.2. There have been many attempts to enhance the system performance. However, those researches were carried out with commercially designed refrigerators that were specifically designed as air-cooled systems.

The operation of the DAR is simple but its characteristics are not. To study the DAR characteristics, an experimental refrigerator based on the Platen-Munters cycle was designed and fabricated. It was designed so that later modifications could be done conveniently. Knowing that hydrogen can cause explosion if leaks, in this study helium was selected as the auxiliary gas instead of hydrogen for safety reason. The experimental refrigerator was tested under various operating conditions. A simple mathematical model was also developed. The experimental and calculated results were compared.

This thesis describes and evaluates both theoretical and experimental studies of a diffusion absorption refrigerator (DAR). Reviews of study in the field of vapor absorption system and DAR in the past are included in chapter 2. An experimental set-up was constructed with the details of construction, which are described in chapter 3. The bubble pump is considered as fundamental to the DAR, it was studied and is presented in chapter 4. A simple set-up was constructed for studying some characteristics of the bubble pump. A curve fitted equation of a bubble pump having similar dimensions as that used in the experimental DAR was obtained. A mathematical model was developed and used as a primary tool for analysis of the experimental DAR. The mathematical model concepts are explained in chapter 5. Actual performances of the experimental DAR are presented in chapter 6. The experimental and the calculated results are also compared and discussed. Discussions of system improvement are included in chapter 7. The system was analyzed theoretically. Some options are proposed as alternatives for system improvements. The last chapter, chapter 8, is conclusions of this study and recommendations for future study.

CHAPTER II

Background of a Diffusion Absorption Refrigeration Cycle

This chapter provides a literature survey of past research on the diffusion absorption refrigeration cycle. Even though this system was invented and used for almost 70 years, very few literatures concerning this system were found.

2.1 System used as a domestic refrigerator

Figure 2.1 shows a concept of natural circulation system based on a thermo-siphon effect. Liquid in the pump tube is heated in order to generate vapor. Bubbles are formed, which float up and push the liquid up to the top. This effect is known as a concept of 'bubble pump'.

For an absorption system using ammonia/water, pressure difference between the generator and the absorber is normally high. Therefore, a bubble pump cannot directly replace a mechanical pump. To overcome this problem an inert gas is charged into the absorber and the evaporator. Then, during the operation, the pressure is equalized throughout the system. In a DAR, the entire system has a single pressure, the evaporator can produce low temperature and cooling effect by principle of the Dalton's law of partial pressure.

Figure 2.2 shows a schematic view of a diffusion absorption cycle, which is used as a domestic refrigerator. It uses ammonia as refrigerant, water as absorbent and hydrogen as auxiliary gas. At the storage tank, there is solution with concentration of 35% (ammonia 35% and water 65% by mass). This solution flows to the generator where it is heated to 180°C and some ammonia is evaporated out. The vapor forms bubbles that push columns of solution in the pump-tube up to the liquid-vapor separator. Weak solution with concentration of 10% collected at the separator returns back to the absorber via a solution heat exchanger (SHX).

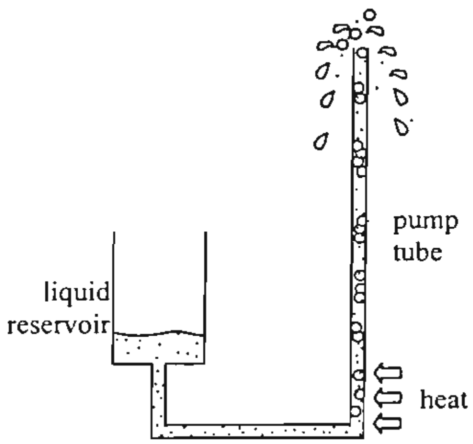

Figure 2.1 A simple set-up of a bubble-pump

The SHX allows the strong solution entering the generator to be preheated by hot solution flowing to the absorber. Using the SHX helps reduce heat input at the generator and thus increases the COP. At the separator, the vapor leaving the pump tube usually contains some quantity of water. Then, the vapor is purified at a rectifier where it is cooled down to 70°C. Water is condensed and drops back as condensate to the separator.

The pure ammonia vapor is then liquefied in the air-cooled condenser. For normal operating conditions, the system pressure is approximately 25 bar. The liquid passes to the evaporator, which is divided into two sections, a freezer and a food chiller. As the evaporator is charged with hydrogen, in the freezer the partial pressures of ammonia and hydrogen are approximately 1 and 24 bar respectively. At this condition, ammonia evaporates at -30 to -18°C. Since ammonia continues to evaporate, its partial pressure rises to 3 bar in the food chiller. This corresponds to an evaporating temperature at -5°C. The vapor is absorbed by the strong solution in the air-cooled absorber. Weak solution is then transferred to the storage tank to complete the cycle.

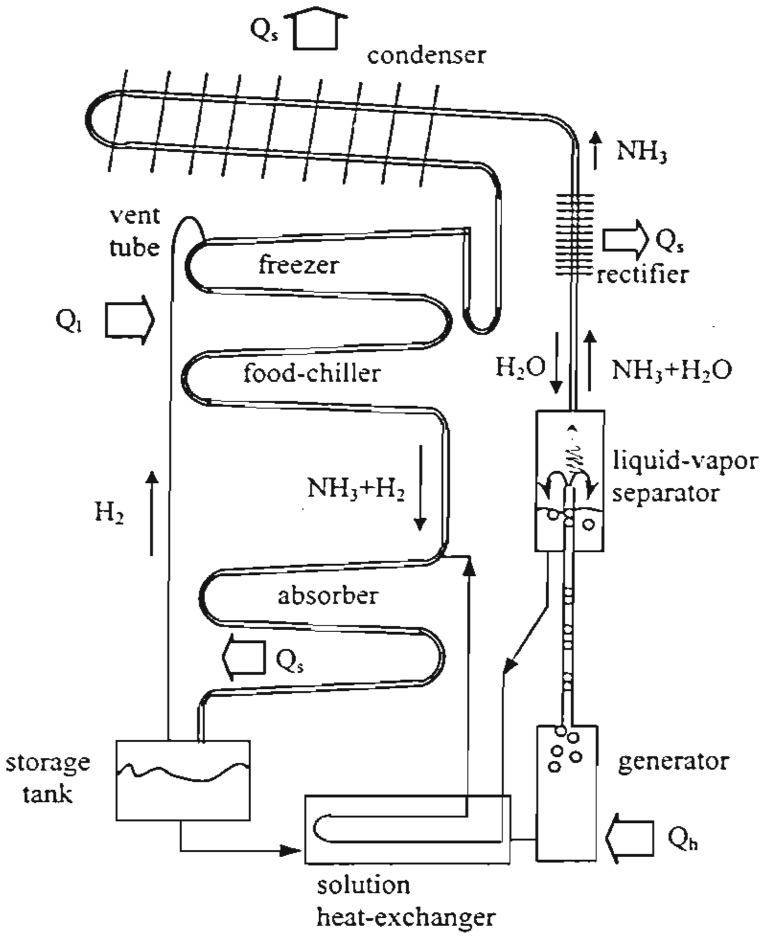

Figure 2.2 A schematic diagram of a diffusion absorption refrigeration cycle.

The density of ammonia is considerably greater than that of hydrogen. The vapor (ammonia and hydrogen) becomes heavier as ammonia is still being evaporated. Therefore, it drops from the freezer to the food chiller and enters the absorber. In the absorber, ammonia is absorbed into the solution, the vapor become lighter and thus it rises up to the top. This causes a circulation of hydrogen in the evaporator-absorber part. Hydrogen circulation affects rate of evaporation in the evaporator and absorption rate in the absorber. Hydrogen not only affects the mass transfer rate, its circulation also reduces the cooling capacity as the gas is warm when it leaves the absorber. A gas heat exchanger may be used to exchange heat between cold (evaporator exit) and warm (evaporator inlet) gases.

2.2 Literatures concerning a diffusion absorption system

The DAR was firstly invented by Platen and Munters in 1920, students at the Royal Institute of Technology, Stockholm in Sweden. However, most of the literatures found concerns improvements of the small domestic refrigerator such as redesign of the boiler, modification of traditional bubble-pump, or inclusion of the auxiliary gas heat exchanger.

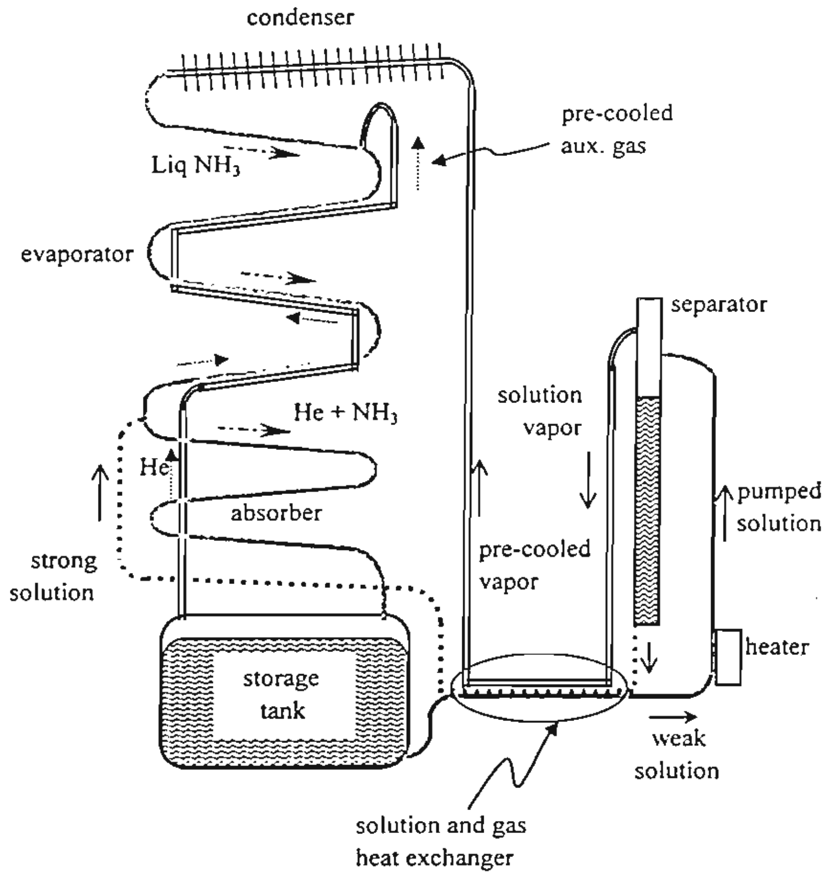

Figure 2.3 A schematic diagram of the integrated 3X-boiler DAR.

2.2.1 Development of the boiler

Normally one-third of heat input to the generator is rejected to the surroundings at the rectifier. This amount of heat could be recovered to improve the system performance. Stierlin and Ferguson [1990], designed a new generator. A solution heat exchanger and a rectifier were integrated as a single unit called 3X-boiler. The schematic diagram of the system integrated with 3X-boiler is shown in figure 2.3. It lets the three fluid streams (weak solution entering the generator, strong solution leaving the separator, and vapor leaving the separator) exchange heat with each other simultaneously. As a consequence of counter-flow heat exchange between these three fluids, the rectification heat was recovered usefully. It helps reduce the heat input to the generator. The system COP could be improved by not less than 50%. It was claimed that, a domestic refrigerator based on this design had a COP of 0.51. At surrounding temperature of 32°C, it operated at freezer and food chiller temperatures of -20°C and 5°C respectively.

2.2.2 Alteration of auxiliary gas and inclusion of gas heat exchanger

Circulation of the auxiliary gas inside the evaporator and the absorber results from density change due to the evaporation of ammonia. Hydrogen, the lowest density vapor, seems to be the most suitable auxiliary gas for a diffusion absorption system. However, hydrogen is extremely flammable. In order to exclude any possibility of explosions, hydrogen may be replaced with helium [Narayankhedkar and Maiya, 1985]. Helium is heavier than hydrogen, thus it affects circulation rate of gas circuit and mass transfer performance. However, helium has lower thermal conductivity and specific heat capacity than hydrogen, internal heat load caused by circulation of gas is reduced. Therefore, the cooling capacity and COP remains unaffected.

The auxiliary gas circulation rate was proposed to cause effect on the system performance [Watts and Gulland, 1958]. An auxiliary gas heat exchanger, GHX, was included to transfer heat from the auxiliary gas after absorption process to that before being absorbed. Some amount of heat from absorption process accumulated in the auxiliary gas was considered as internal load. This amount of heat is normally transferred to the evaporator as an internal load. Higher gas circulation rate normally increases mass transfer in the evaporator-absorber. It accelerates evaporation rate of ammonia in the evaporator.

However, it increases the internal load to the evaporator, which reduces useful refrigerating effect. It is recommended that good mass transfer surfaces and highly efficient GHX are required for enhancement of the net refrigerating effect. Eber [1975] introduced new compact heat exchangers for absorption cooling units. The heat exchanger was designed to have a compact size with cylindrical fins. It caused the GHX to have better heat transfer performance while flow resistance was minimized.

Chen et al. [1996] conducted experiments on a small commercial refrigerator. Performance was obtained based on measured heat transfer through the cabinet and electrical energy required by the generator heater. When the heater power, at the generator, was increased from a certain value, cooling capacity and COP increased rapidly due to the start up of the bubble pump. Then the COP remained constant while the cooling capacity increased as more ammonia was produced. In this range, ammonia was completely evaporated in the evaporator. If the heater power continued to increase, more ammonia was produced. However, it could not completely evaporate due to the capacity of the evaporator or the absorber. As the heat input increased while the cooling capacity remained constant, COP dropped. Similar to Stierlin and Ferguson [1990], Chen et al designed a generator for a small refrigerator. Released heat from rectification process was used to preheat the weak solution before entering the generator. It was claimed that, COP was increased from 0.2 to 0.3, a 50% increased.

2.2.3 Improvement of the bubble pump

The DAR was optimally operated within a specific operating range. Variation of heat input at the generator would stop operation of the unit intermittently. This normally happened whenever variation of the refrigerating load was required. Various designs of hydraulic trap were installed and tested with the domestic refrigerators [Sellerio, 1951].

Some designs were proposed to enhance pumping performance. Attenuation of intermittence in liquid pumping was obtained. Moreover, it could be operated with higher pumping effect. Lucus [1967] proposed a specially design boiler. It was designed to serve the system operation in both low and high input power. Therefore, the system could be operated continuously with a wider range of input power. This concept was similar to the boiler designed by Stierlin [1967], which uses the evaporator partially as the gas heat exchanger. Heat could be exchanged internally in the system. This boiler design was intended to be used in the deep-freezer absorption unit.

Performance of a bubble-pump may be defined as the pumping ratio, which is given as a ratio between volume flow rate of liquid and vapor through the pump-tube. Experimental studies using water/lithium bromide as fluid were conducted [Pfaff et al., 1998]. Pump tubes with diameter of 10, 14 an 18 mm were used. For fixed level of liquid in a reservoir and length of pump-tube, liquid flow rate increased linearly with the power input to the generator and pumping ratio was constant. The pumping ratio increased when the level of liquid in the reservoir was increased or the tube length was reduced. Pumping action was not found for 18mm tube as vapor formed small bubbles and caused the liquid to sparkle. This caused the liquid to oscillate in the tube without being lifted up. For smaller tubes, bubbles covered the whole cross section of the tube. The bubbles acted like a piston and lifted the liquid up.

2.3 Conclusion

From the literature, many researchers tried to improve performance of the diffusion absorption systems, which were used as domestic refrigerators. Most of them tried to improve the COP by reducing heat input to the generator. One notable approach was to integrate a rectifier and a solution heat exchanger together. Thus, the rectification heat could be recovered to the generator.

It must be noted that there was no attempt to design or construct a system totally different from those used as domestic refrigerators. Therefore, the effect of operating temperatures, charging of working fluid, and component design could not be clearly understood.

Although a diffusion absorption system has relatively poor COP compared with other refrigeration systems, it provides many advantages. Before a more advanced system can be developed, extensive research concerning a diffusion absorption refrigeration system is needed.

CHAPTER III

The Experimental Refrigerator

An experimental refrigerator was designed to test a diffusion absorption refrigeration cycle. The capacity of each component was designed based on data listed in table 3.1. They were used as a guideline. It was expected that the actual operating conditions would be different from the calculated values. Therefore, all the components were over-sized. Figure 3.1 and 3.2 shows a photograph and a schematic diagram of the entire system respectively.

Table 3.1 Design data for the experimental refrigerator

generator temperature (°C)	150-200
rectifier exit temperature (°C)	50-90
absorber temperature (°C)	30-50
condenser temperature (°C)	30-50
generator heat input (W)	3000
cooling capacity (W)	600

3.1 Construction material

All the material used must be able to withstand pure ammonia and its aqueous solution. The major disadvantages of ammonia are its high condensing pressure, toxicity and corrosive to copper and copper alloy. The system maximum pressure and temperature were 20 bar and 200°C respectively.

All vessels were constructed from stainless steel 304 schedule 10s pipes. Fittings, valves, and tubes used were made from 316 stainless steel. Sealing materials (o-ring) were EPDM for temperature below 150°C and PTFE for temperature above 150°C. Liquid Teflon was used as thread sealant for all thread fittings.

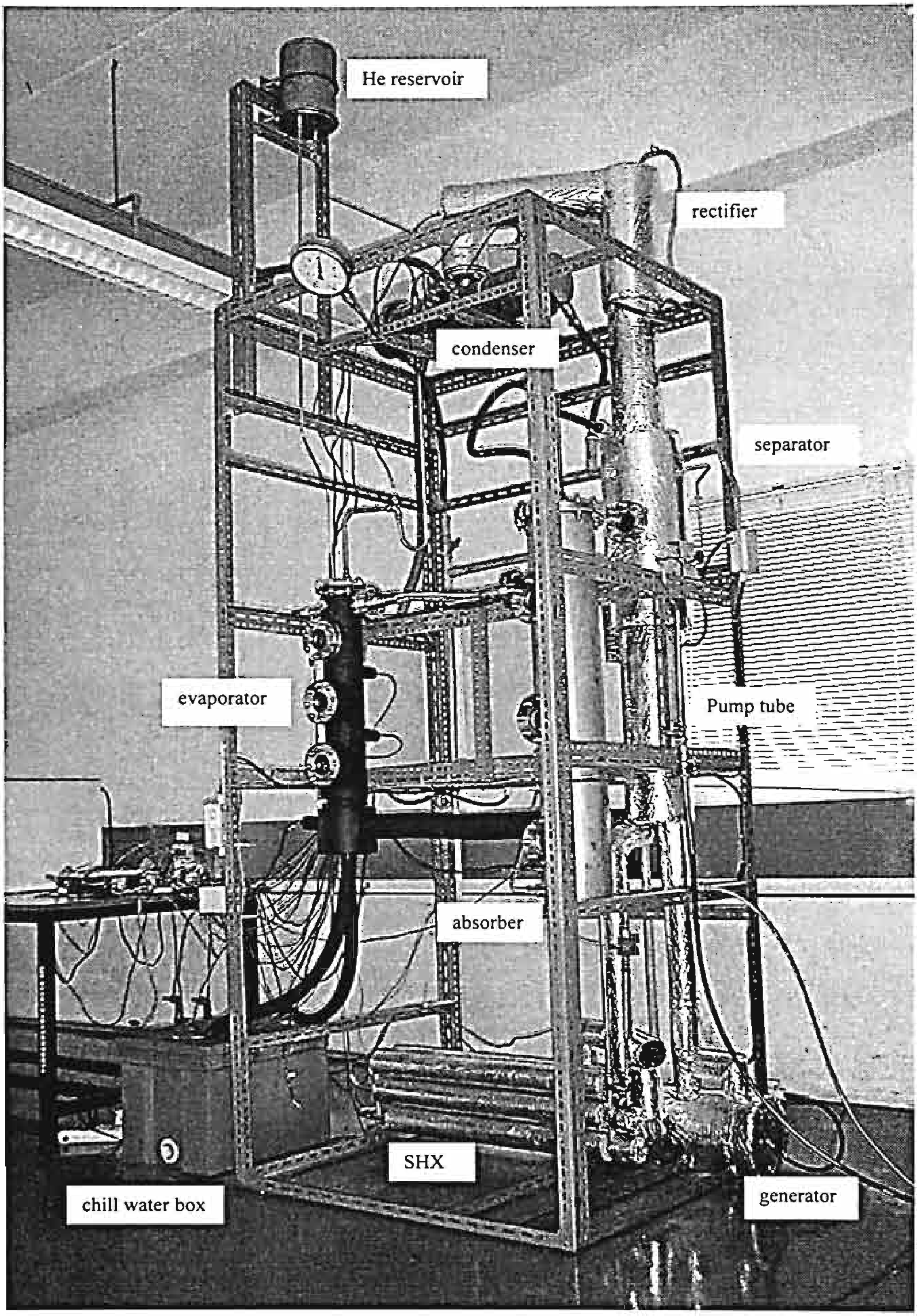

Figure 3.1 A photograph of the experimental set-up

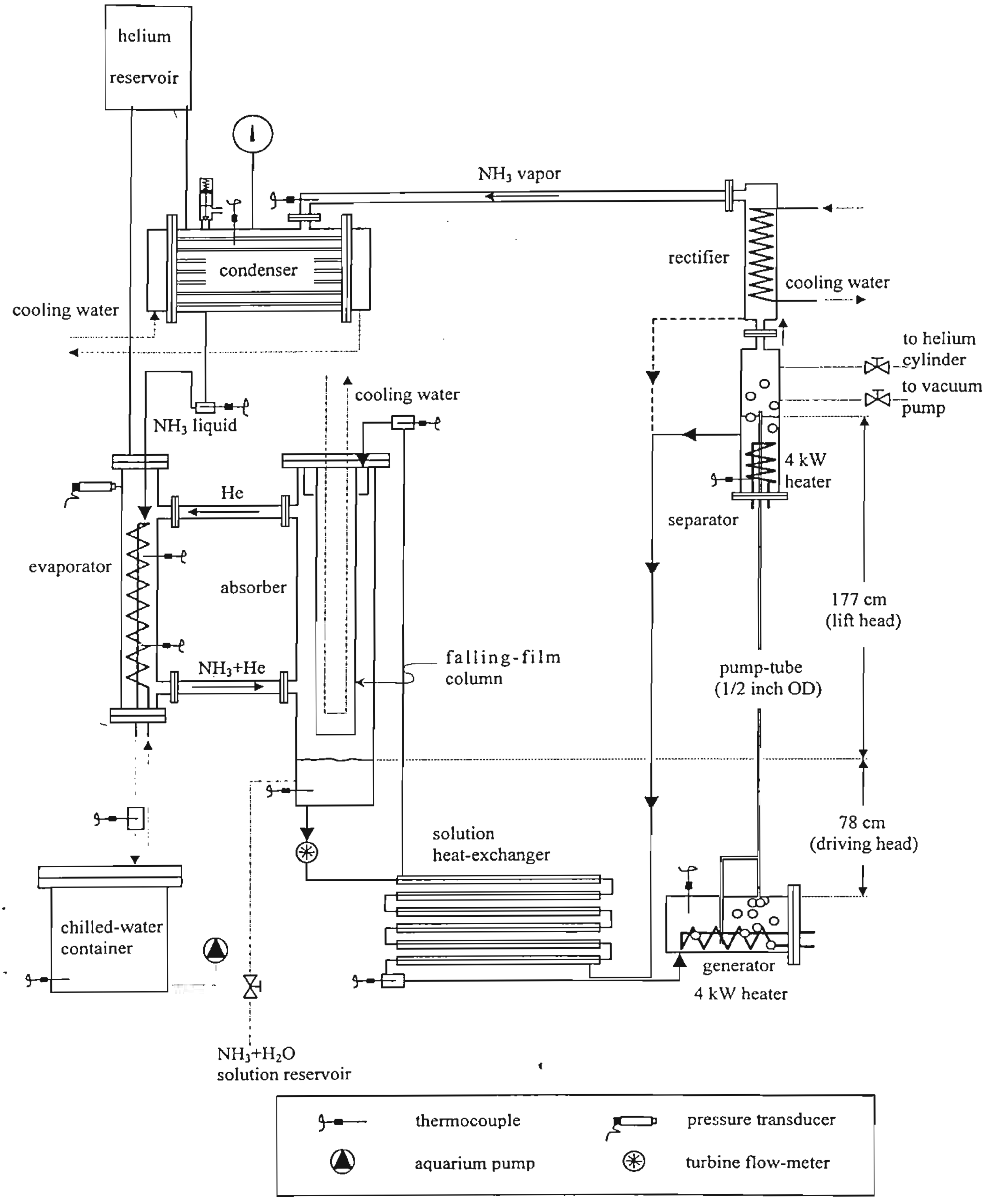

Figure 3.2 Schematic view of the experimental refrigerator.

3.2 The condenser

The condenser was designed based on a shell and tubes heat exchanger, cooling water was arranged to flow through the tubes. It was designed for 8 kW of heat load. It consisted of sixty 10 mm OD tubes, which provided total surface area of 1.2 m^2. The shell was constructed from a 0.50 m long 6-inch pipe. The condenser was equipped with a sight glass. The sight glass was 12 mm thick tempered-glass with 80 mm diameter.

3.3 The solution heat exchanger (SHX)

The solution heat exchanger consisted of six double pipe heat exchangers connected in series. The hot solution from the separator and the warm solution from the absorber flowed counter currently through the annular duct and inner tube respectively. The inner tube was a 12 mm OD seamless tube while the outer was a half-inch pipe. The total length was 6 m, which provided a total heat transfer area of 0.25m^2.

3.4 The evaporator

In the evaporator of conventional refrigeration systems, when pressure is abruptly dropped, liquid refrigerant evaporates to vapor at low temperature as it absorbs heat from a cooling space. The refrigerant is evaporated as it is heated in the same way as water in a kettle. The evaporation rate is controlled by an amount of heat input.

In an evaporator of a diffusion absorption system, liquid ammonia evaporates in an environment of an auxiliary gas (hydrogen or helium). This process is similar to how water evaporates from a wetted cloth. The evaporation rate is increased if the cloth is exposed to blown air or its evaporation rate can be enhanced more with reduced humidity of the air. Thus, the evaporator must be designed with enough wetted surface area. The pressure dropped must be minimized, in order to allow the inert gas to circulate.

The constructed evaporator consisted of a vertical coiled tube installed inside a shell pipe. Liquid ammonia was allowed to drop on the top to create a liquid film and evaporate in an environment of helium on the outer surface of the coil. The cooling load was obtained as circulation of chilled water through the coil.

The coil has a pitch diameter of 55 mm and 0.6 m long. It was made from 6 m of 10 mm OD seamless tube. This provided a total area of 0.19 m^2. The shell was constructed from a 3-inch pipe, 0.75 m long. At the top coil, droplets of liquid ammonia were created by a 6 mm OD tube, which was pointed to the top coil. The shell was equipped with three sight glasses. Thus, evaporation process could be observed.

There were two 1-inch pipes connected at the top and bottom of the shell. The bottom pipe allowed the evaporated ammonia and the auxiliary gas to be drawn into the absorber. This pipe also allowed the unevaporated water, which was carried in with liquid ammonia, to return back to absorber. The other pipe allowed the warm auxiliary gas from the absorber to return back to the evaporator.

3.5 The absorber

There are several types of absorber e.g. packed column, spray column, and falling film column. A falling film column was chosen because of its low-pressure drop and simplicity in construction. The need of an external heat exchanger and a circulation pump were eliminated as well. Theories of the falling film absorption process are given in several textbooks. A design procedure of a falling film absorber taken from the text book is as follows [Treybal 1968].

The change in concentration over a given length of falling film is,

$$\frac{\delta V_{avg}}{L}\left(C_L - C_O\right) = k_{l,avg}\left(C_i - C\right)_{avg} \tag{3.1}$$

The average concentration change in the right hand side could be obtained from a logarithmic average value as,

$$(C_i - C)_{avg} = \frac{(C_L - C_O)}{\ln\left[\frac{(C_i - C_O)}{(C_i - C_L)}\right]} \tag{3.2}$$

where $k_{l,avg}$ average mass transfer coefficient ($m \cdot s^{-1}$)
V_{avg} average film velocity ($m \cdot s^{-1}$)
L length of wet wall (m)
δ film thickness (m)
C ammonia concentration ($kmol \cdot m^{-3}$)
C_O ammonia concentration at the beginning of the wall ($kmol \cdot m^{-3}$)
C_i ammonia concentration at liquid vapor interface ($kmol \cdot m^{-3}$)
C_L ammonia concentration at the end of the wall ($kmol \cdot m^{-3}$)

The average film velocity could be calculated from:

$$V_{avg} = \frac{\Gamma}{\rho \delta} \tag{3.3}$$

The film thickness,

$$\delta = \left[\frac{3\mu\Gamma}{\rho^2 g}\right]^{1/3} \tag{3.4}$$

The mass transfer coefficient depends on contact time of liquid and vapor. The contact time is long when the flow rate is small or the film Reynolds number ($Re=4\Gamma/\mu$) less than 100. For long contact time,

$$k_{l,avg} = 3.41\frac{D}{\delta} \tag{3.5}$$

For short contact time,

$$k_{l,avg} = \left[\frac{6D\Gamma}{\pi\rho\delta L}\right]^{1/2} \tag{3.6}$$

where Γ mass flow rate per unit width ($kg \cdot m^{-1} \cdot s^{-1}$)
μ viscosity ($kg \cdot m^{-1} \cdot s^{-1}$)
D molecular diffusivity ($m^2 \cdot s^{-1}$)
ρ liquid density ($kg \cdot m^{-3}$)

Height of the wall could be calculated by dividing it into a series of incremental lengths and assuming isothermal condition over each increment. The concentration entering and leaving each increment could be determined by using equations (3.1) to (3.6).

The constructed absorber consists of an internal vertical column on which solution formed a liquid film. The film would be created as the solution passed through a small gap at the top of the column. The surface of the column was cooled by the circulation of cooling water inside the internal column. The internal column was a 0.7 m long 4-inch pipe. This provided an external surface area of 0.25 m^2. The absorber shell was a 1 m long of a 6 inch pipe. Three sight glasses were used for observation of the absorption process. Two 1 inch pipes were used to connect the absorber's shell to the evaporator.

3.6 The bubble-pump system

The bubble pump system consisted of a pump tube, a generator, and a liquid vapor separator. The vapor generator was a horizontal shell (0.35 m long, 4-inch pipe). Heat input was supplied by a 4 kW electric heater. The pump-tube was attached to the top of the shell. The vapor, which was driven off from the solution, entered the pump tube and formed series of bubbles. The bubbles pushed the liquid solution, which entered via the liquid-inlet-tube, as shown in the schematic diagram, up through the pump tube. The liquid-inlet-tube was inserted into the generator, so that only liquid was allowed to enter. At the top of the pump tube, liquid solution dropped and accumulated in the separator before returning back to the absorber while the vapor rose up to the rectifier. The separator was constructed from a 0.35 m long of 4-inch pipe. It was equipped with a 4 kW immersion heater, which could be used to generate additional ammonia vapor. The use of two heaters (one in the generator and the other in the separator) provided more flexible operation as it allowed the liquid and vapor flows rate to be controlled separately. However, in this study, the second heater was not used.

3.7 The rectifier

Normally, ammonia vapor exiting the pump tube is not pure. It contains some amount of water, which is evaporated with ammonia. If this water is allowed to liquefy in the condenser, it will enter the evaporator and cause the evaporator temperature to increase (at a given pressure, ammonia evaporates at a higher temperature with increased amount of water). In order to maximize the system performance, the ammonia vapor must be purified before entering the condenser. To purify the vapor, it must be cooled down to a certain temperature in a rectifier. Thus, only pure ammonia can enter the condenser as the water is condensed and dropped back to the separator. The rectifier used here was designed based on a shell and coil type heat exchanger. The shell was a 0.4 m long 3inch pipe. The internal cooling coil, through which cooling water was circulated, was made from 6m of 10mm OD tube and provided total surface area of $0.19m^2$.

3.8 Helium reservoir

The helium reservoir was constructed from a 0.25 m long 6 inch pipe. It was located at the highest position and was connected to the evaporator and the condenser by two vent tubes (6mm OD). The helium reservoir allowed the helium to expand when the condenser temperature and pressure increased. The vent tubes also allowed the evaporator pressure and the condenser pressure to be equalized.

3.9 Thermal insulation

To reduce unwanted heat losses from the system, hot components (generator, pump-tube, rectifier, solution heat exchanger, and pipe line) were insulated with 30 mm thickness of glass fiber wool with aluminum foil backing. To prevent unwanted heat gains to the system, the evaporator was covered with rubber foam insulation. The condenser was also insulated with rubber foam, so that its condensation temperature could be kept as required. The absorber, the helium reservoir, and the rectifier were left uncovered.

3.10 Hermetic seal test

The feature of this experimental refrigerator was that it could be easily dismantled for modification. This resulted in the use of many removable joints and gaskets. The system was tested hydrostatically to 40 bar. To ensure leak proof, the system was tested with compressed nitrogen to 20 bar.

3.11 Working fluids

Diffusion absorption system uses three working fluids. Ammonia (NH_3) and water are used as refrigerant and absorbent respectively. For a commercially available refrigerator, hydrogen (H_2) is used as auxiliary gas. However, in this experiment, helium was used instead of hydrogen for reasons of safety.

Ammonia and water solution with concentration of 0.25 was prepared in a separated vessel as shown in figure 3.3. The vessel was initially filled with 7 liters of distilled water. Before the vessel was charged with ammonia, the air inside was evacuated by using a vacuum pump. The concentration was calculated by weighing the vessel containing solution.

The experimental refrigerator was evacuated before charging the prepared-solution. Then the solution was charged to the absorber. After the system was charged with ammonia-water solution, the system pressure was recovered. Then it was charged with helium until the required pressure was obtained.

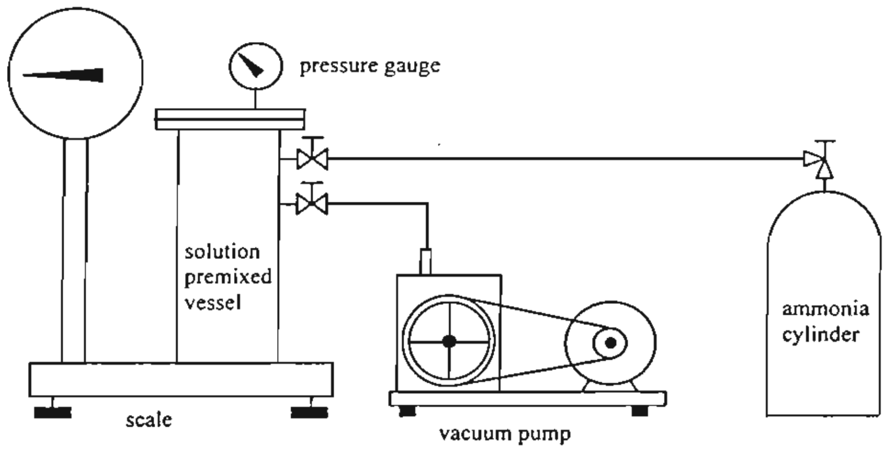

Figure 3.3 A set up used to prepare ammonia-water solution.

3.12 Instrumentation and control

Variables to be measured and controlled were temperature, pressure, flow rate, and electric power input. Control and data logging were provided by a personal computer with data acquisition system (ADAC 5525MF). It was arranged so that the operating temperature and pressure could be pre-set and controlled.

Type K thermocouple probes were used for measuring fluid temperature at various locations as shown in Figure 3.2. Each probe was connected to a thermocouple amplifier with an output of 10 mV/°C. Each was calibrated using a precision thermometer.

System pressure could be measured by using a dial compound gauge (-1 to 24 bar). A pressure transducer (0 to 60 bar) with signal output of 0 to 10 V was also used for more precise measurement. It was calibrated using a standard dead-weight tester. The signal output was sent to the computer.

The generator's and the separator's temperatures were controlled by adjusting their power of respective heaters (using computer controlled solid-state relays). The electrical power input to each heater was measured using a power transducer. Each transducer was calibrated using a digital power meter.

Temperature of the cooling water was maintained between 30 and 32°C throughout the tests. Temperature of vapor ammonia exiting from the rectifier was controlled using solenoid valve to on/off the cooling water.

3.13 Conclusion

The details of design and construction of the experimental refrigerator are described. Heat input to the system was provided by means of an electric heater. This allowed the power input to be controlled and recorded conveniently. The system was constructed so that each component could be analyzed individually.

CHAPTER IV

An Experimental Study of Bubble Pumps

In the DAR, circulation of the working solution employs natural circulation. A mechanical pump is not required which is an outstanding feature of the DAR. It is found that performance of a DAR system depends strongly on characteristics of a bubble-pump. Therefore, it is necessary to study the bubble pump characteristics. It is a key parameter for the mathematical model, which is going to be presented in the following chapter.

From the literature [Maiya, 1985], it is known that the important parameters of the bubble pump are pump tube diameter, driving head, lift head and pump heat input. In this study a bubble pump was tested using water instead of aqueous ammonia solution and compressed air instead of vaporized solution. This eliminated the need of heat input for vaporization of working solution. Moreover, it was more convenient to quantify the actual amount of air supply. The pumping performance was found to depend on four parameters i.e. tube length, tube diameter, head-ratio, and airflow rate. Two sizes of pump-tube were tested. A curve-fitted equation of tube having same size as that used in the DAR set-up was obtained.

4.1 The experimental set-up

The experimental set-up was constructed with a simple design. A photograph of the set-up and a schematic diagram are shown in Figures 4.1 and 4.2 respectively. The set-up was tested with four parameters that varied independently. The tests were conducted under atmospheric pressure. Pure water was used instead of ammonia-water solution and air was used instead of vaporized solution. Therefore, the working condition could be varied conveniently as required. Airflow rate was easily controlled by adjusting a valve installed at the airflow-meter inlet.

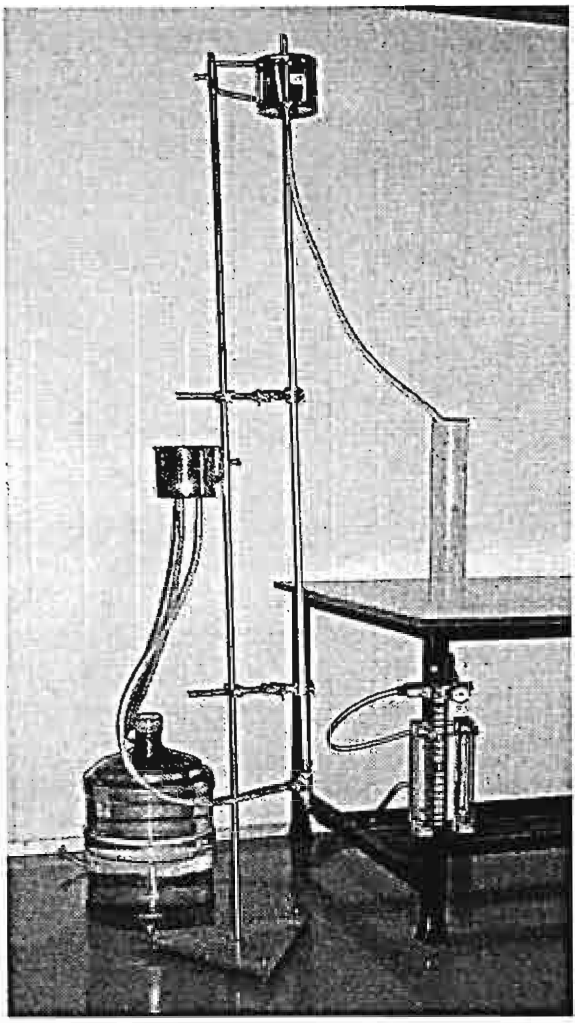

Figure 4.1 A photograph of the bubble pump set-up.

Two sizes of pump tube were tested. One was 3/8 inch outside diameter (OD 9.52 mm, ID 7.7 mm) and another was a 1/2 inch OD (OD 12.7 mm, ID 10.75 mm). Each size was tested with three different lengths, 1.00 m, 1.65 m, and 2.00 m. Effects of airflow rate and head ratio were studied. The head ratio is defined as:

$$HR = \frac{\text{driving head}}{\text{lift head}} \tag{4.1}$$

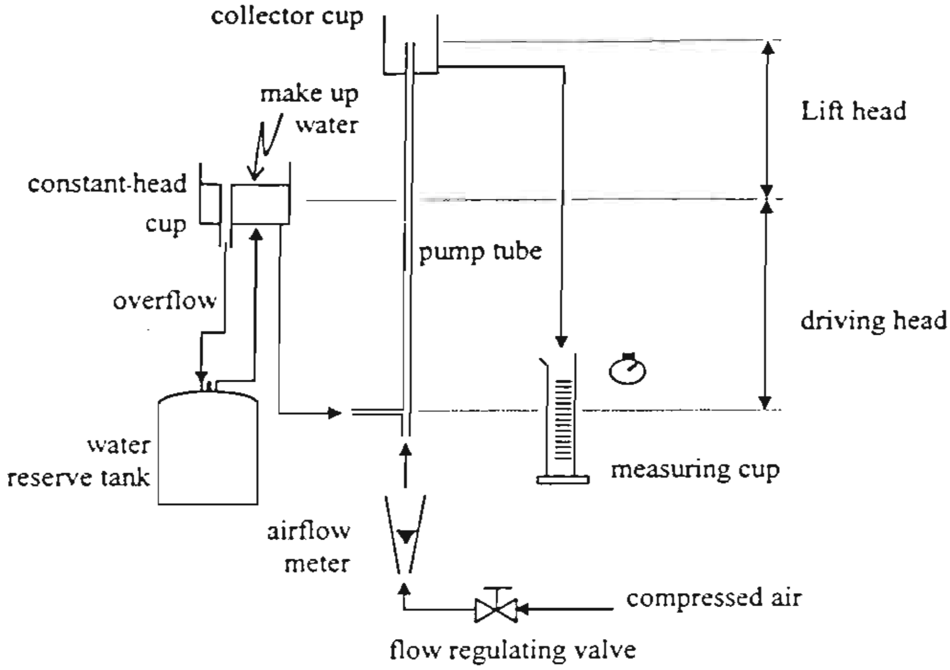

Figure 4.2 A schematic diagram of the bubble-pump test rig.

Referring to the schematic diagram, the head ratio could be adjusted by leveling the constant-head cup until the lift head and the driving head correspond to the required head ratio. Airflow rate could be adjusted by a metering valve installed at the airflow meter inlet. Pumped water flow rate could be determined from the obtained amount of water within a period of recorded time. The constant-head cup, pump tube, and collector cup were put along the set-up column by fixtures. All fixtures could be adjusted along the column so that levels of all equipment attached could be adjusted as required.

4.2 Experimental procedures

Adjust the head ratio to a required level by sliding the constant-head cup along the column. The airflow rate was adjusted by regulating pressure of compressed air to a level somewhat higher the atmospheric pressure, 1.5 bar. Then, it could be finely adjusted to the required value by adjusting the needle valve at the airflow meter inlet. Water was pumped to the collector cup and flowed downward through a plastic hose. During each test, the water and airflow rates were recorded. The pumped water was collected and timed. Then, the pumped water flow rate could be determined. Air was supplied with variation step of 0.2 $l\cdot min^{-1}$ from 1 $l\cdot min^{-1}$ to 5 $l\cdot min^{-1}$. Then, the head-ratio was altered and the experiments were repeated. Variation of head-ratio was done with 7 values i.e. 80:20, 70:30, ..., 20:80. Then, the pump tube was changed with another tube size and the experiments were repeated. Comparison of the pumped water flow rate with variation of tube size, tube length, and head ratios of selected cases are presented as in Figures 4.3 to 4.5 respectively.

4.3 Effect of tube diameter

Figure 4.3 shows effect of tube diameter to the pump performance at a fixed head ratio. For each tube size, the obtained results showed similar trend of pumped water with variation of airflow rate. However, the amount of pumped water was different. It should be noted that with the small tube, 3/8 inch, the amount of pumped water was increased with increased air supply until a certain point. At this point, the pumped water flow rate was maximum. Increment of airflow rate caused reduction of pumped water flow rate. It was clearly shown that the maximum water flow rate was reached with 3/8inch pump tubes. For pump tube of 1/2inch, the pumped water flow rate was found to increase with increased airflow rate throughout the tested range. No maximum water flow rate appeared. If a higher airflow rate was available, it would be expected that the results would be similar to the case of 3/8inch tube. From the Figure, it can be seen that at low airflow rate, a smaller tube performed better pumping effect than a larger one. However, the operating range of 1/2inch tube was wider.

It should be noted that with the small tube, the start of pumping effect occurred at a lower airflow rate than the larger one. It is easier for slug flow to occur in the smaller tube due to smaller cross sectional area of the tube. However, the operating range of the smaller tube was less. Air in the small tube flows faster than that in the large tube for the same airflow rate. It is clearly shown that to start the pumping effect for the large tube, more airflow is required comparing with that of the small tube. This should result from larger cross sectional area of the tube, which is harder for slug flow to occur. Similar trends were obtained for different tube length but with different values of flow rate.

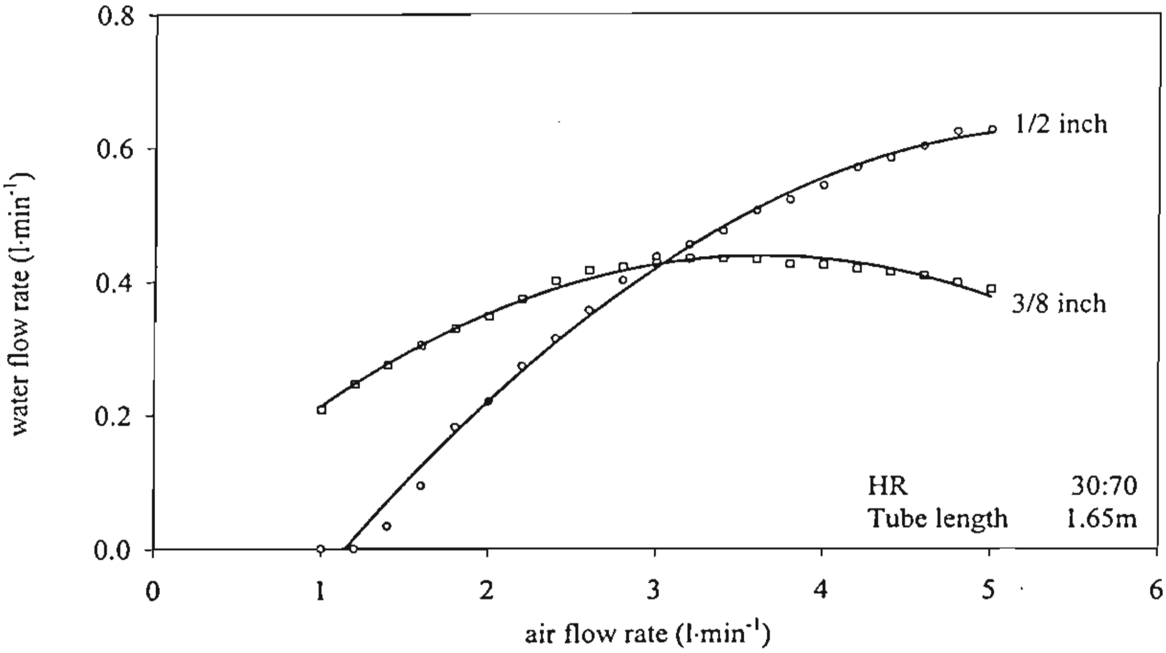

Figure 4.3 Effect of tube diameter on pumping performance of bubble pump

4.4 Effect of tube length

Figure 4.4 shows effect of tube length to the pump performance. Performance of the pump was found to improve with the tube length. However, increments of the pumping effect decreased with the length. The pumped water flow rates at the same supply airflow rate of each tube length are differed. For example with half inch tube at airflow rate of 5 $l\cdot min^{-1}$, pumped water was 0.4 $l\cdot min^{-1}$ with 1m long tube, 0.63 $l\cdot min^{-1}$ with 1.65 m long tube, and 0.68 $l\cdot min^{-1}$ with 2.00 m long tube. It is clearly shown that pumped water flow rate is not directly proportional to the length of tube. Comparing between 1.00 m tube and a 1.65 m tube, the pumped water flow rate increased around 50%. For the 1.65 m tube and 2.00 m tubes, the pumped water flow rate increased around 8%.

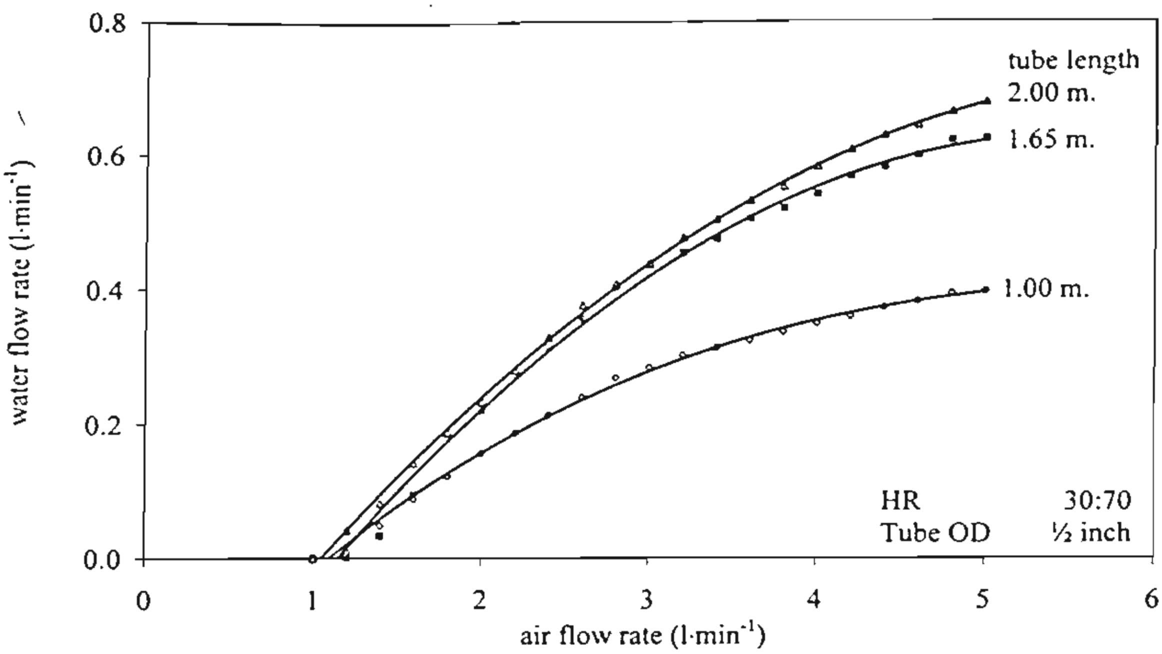

Figure 4.4 Effect of tube length on performance of bubble-pump.

The increment of pumped water flow rate might be considered as a result of greater volume occupied by the water in the pump tube during each pumping cycle. Greater volume of water could be accumulated in a longer pump tube with a similar head ratio. This amount of water would be pumped in each pumping cycle. However, the longer distance that water must flow through the tube causes greater friction loss during the flow. Therefore, the pumped water flow rate was found to increase with longer pump tube. While the incremental rate with longer pump tube was reduced as a result of higher pressure drop occurring from friction loss of water in the tube.

4.5 Effect of head ratio

Figure 4.5 shows effect of head ratio on the pump performance. It was found that the pump performance improved with increased head ratio. At the same airflow rate, the water flow rate increased almost linearly with the head ratio.

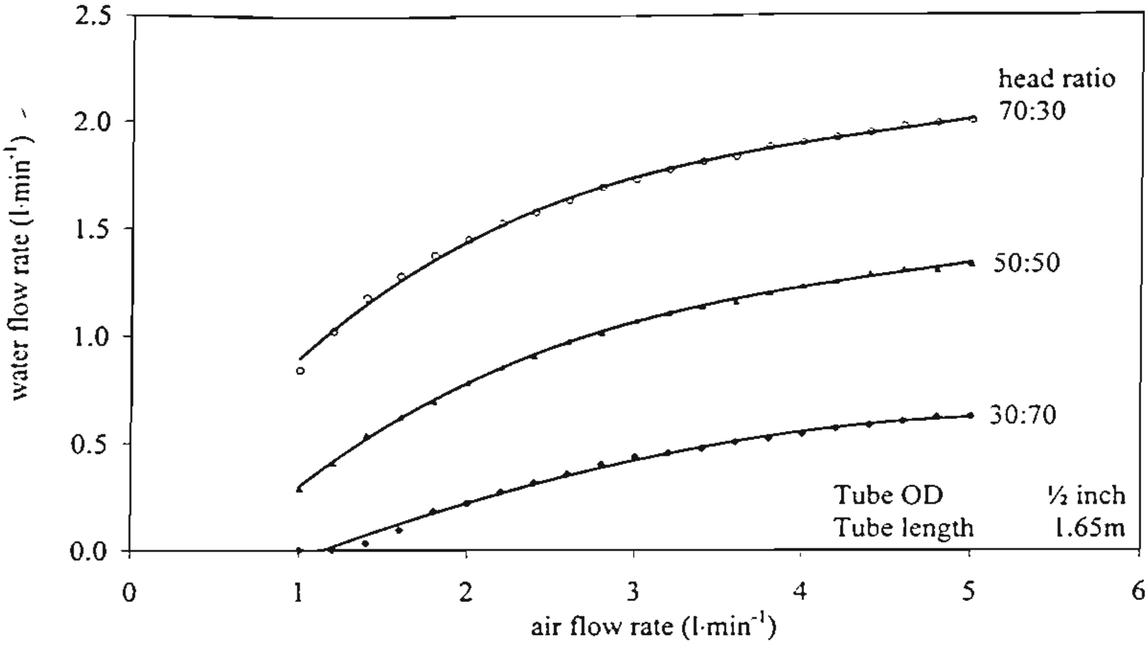

Figure 4.5 Effect of head ratio on performance of bubble pump.

Higher pumped water flow rate was obtained with increased head ratio. According to the definition of head ratio, it could be considered as a driving pressure. The amount of water accumulated in the tube should be more at a higher head ratio as shown in figure 4.5. The greater depth causes higher pressure difference, which resulted in higher pressure to push water into the pump tube. It could be imagined that with higher head ratio, water could be pushed into the tube easier. A new batch of water could be pushed to occupy the volume of water just being pumped in the prior pumping cycle. Therefore, the pumping performance should be improved with head ratio.

4.6 The pump tube used in the experimental refrigerator

In this study,1/2 inch tube (12.7 mm OD and 10.9 mm ID) was arbitrarily chosen as a pump tube for the experimental refrigerator. The lift head was 1.0 m and the driving head is 0.65 m. The operating range was extended to be 1-15 l·min^{-1} so as to be fitted as an equation for using in the calculation model (in Chapter 5).

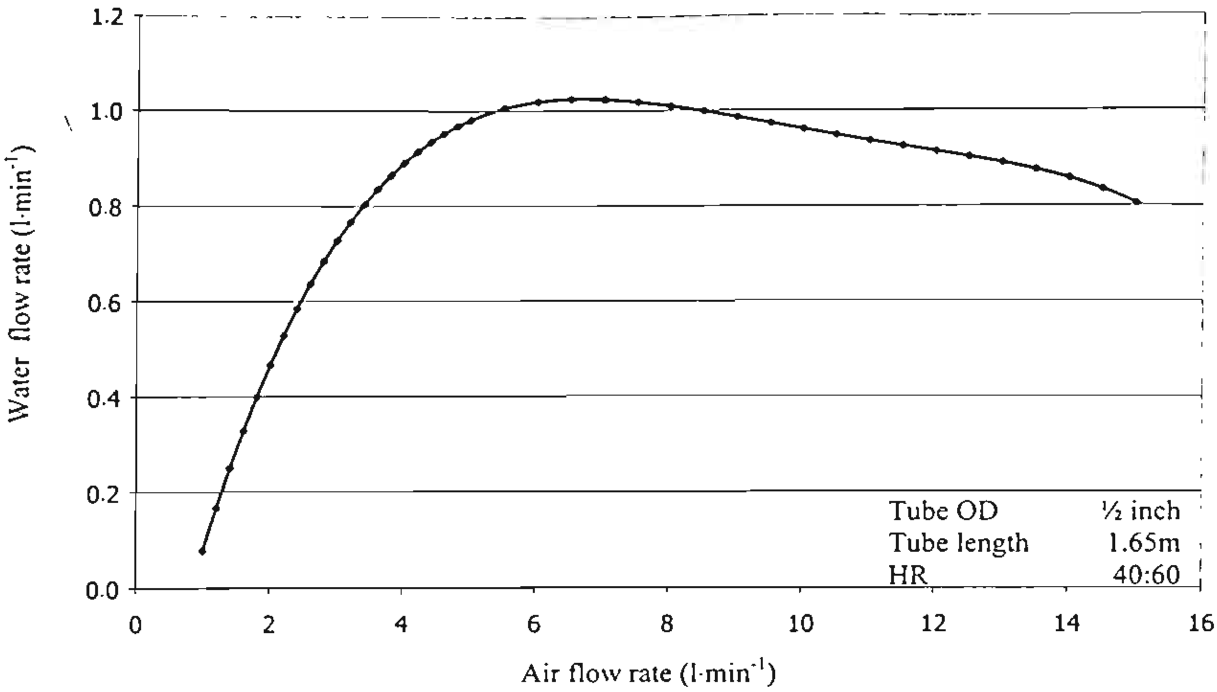

Figure 4.6 Performance of the bubble-pump used in the experimental refrigerator.

The performance curve is provided in figure 4.6. By using least square method, the relation between air and water flow rate through the pump tube is:

$$\dot{V}_{wat} = -0.00014\dot{V}_{air}^4 + 0.00625\dot{V}_{air}^3 - 0.09706\dot{V}_{air}^2 + 0.63772\dot{V}_{air} - 0.46802 \qquad (4.2)$$

This fitted equation is validated to the experimental data with the R-squared value of 0.997.

4.7 Conclusion

It can be concluded that performance of a bubble pump is strongly dependent on the airflow rate and tube dimensions (diameter, length, and head ratio). For a diffusion absorption refrigeration system, performance of a bubble pump is important. It can be used to determine the refrigerant circulating in the condenser and evaporator, and the solution circulation rate between the generator and the absorber.

In this chapter experimental studies of a bubble-pump were conducted. A simple rig was designed and constructed. Air was used as driving vapor and water was used as pumped liquid. The tests showed that the pump performance was depended on the tube dimensions, airflow rate, and head-ratio. The pump-tube with the same dimensions with that used in the experimental refrigerator was also tested. A curve-fitted equation was obtained.

CHAPTER V

Mathematical Model

A simple mathematical model for calculating the performance of the experimental DAR is presented in this chapter. The model is intended for use as a primary tool for analysis of the system. It is used to determine the maximum performance for given operating conditions. When comparing the calculated results with actual values, it shows how losses occurred in different devices. It helps in modifications of the system. To calculate the model, some operating conditions are required. They can be obtained from the experimental results. This chapter provides concepts of the calculation model and calculation procedures.

Figure 5.1 shows a schematic diagram of the experimental refrigerator used in the analysis. Knowing that the DAR is a complex system, which requires specific information in various areas. It is complicated for studying in all details. To simplify the calculation model, some assumptions must be included. They are listed as the followings:

- Temperatures and pressure of working fluid were obtained, based on actual values, from the experimental refrigerator.
- The system was operated in a steady state at the specified working conditions.
- There was no heat loss, no condensation of the vaporized working solution while it was flowing through the pump tube. Therefore, all vaporized liquid would cause a pumping effect in the bubble pump.
- The entire system had an equalized pressure. Pressure drops occurring in tubes, heat exchanger, and system components were negligible.
- The internal heat load as well as mass transfer due to the auxiliary gas was not considered.

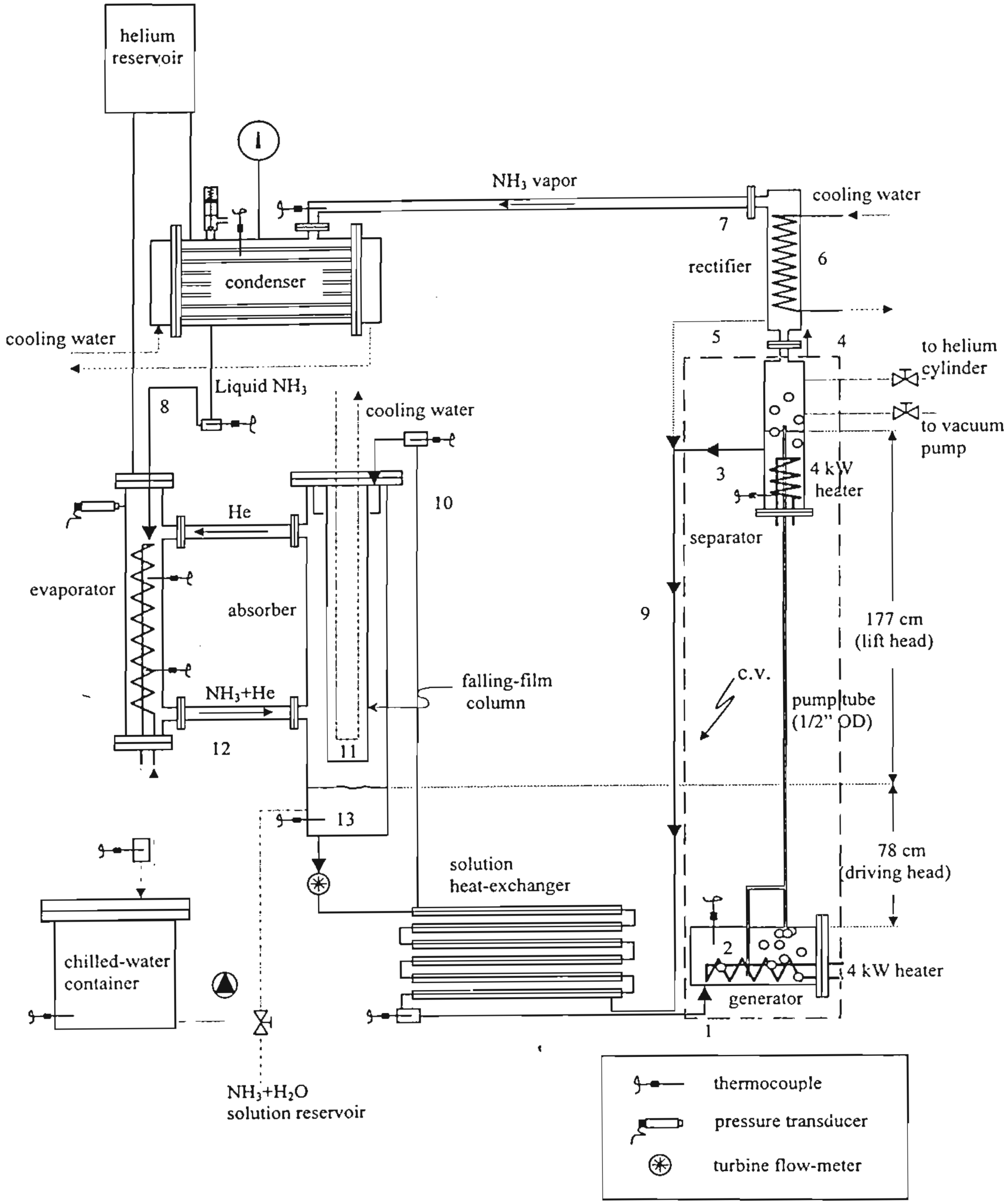

Figure 5.1 Schematic view of the experimental refrigerator.

- For each system component, energy and mass transfer were calculated based on the first law of thermodynamics.
- Based on the assumptions used, the model would provide maximum COP and cooling capacity for given operating conditions.
- The bubble-pump characteristic was obtained as a curve-fitted equation based on the experiment described in chapter 4.
- Properties of aqueous ammonia solution were obtained from correlations provided by Patex and Klomfar [1995] and ASHRAE [1993].

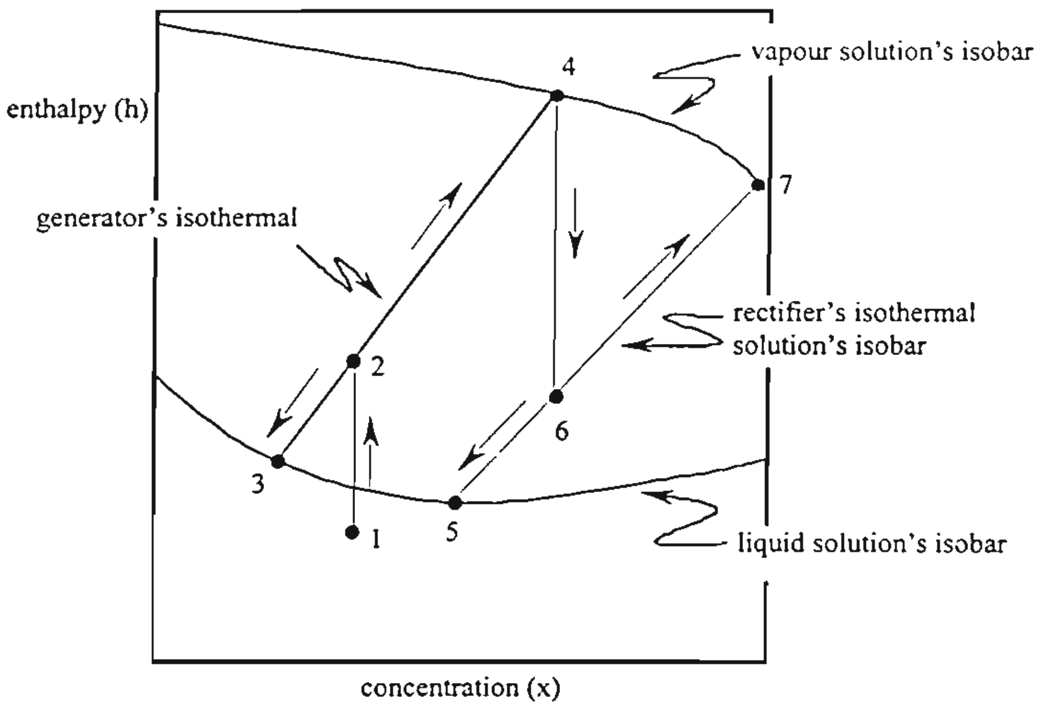

Figure 5.2 P-T-X-h diagram of aqueous ammonia solution for the processes related to the model.

5.1 Generator, Bubble pump, and Separator

According to Figures 5.1 and 5.2, heat input at the generator, $\dot{Q}_{in}$ is used for two purposes. Firstly, it is used for vaporization of weak solution. The solution entering the generator is heated up and vaporized to form bubbles. They agglomerate to become a larger gas volume pushing columns of liquid above itself in the pump tube. A slug flow occurring in the pump tube is known as a bubble pump. The saturated liquid solution is lifted up to the separator by the vaporized solution flowing upward due to density difference between liquid and vapor. A correlation of vaporized solution and liquid solution flow rate was obtained from the experiment of a simple bubble pump set-up as described in chapter 4. To determine the vaporized solution and the liquid solution flow rate and amount of heat input, an experimental correlation of a bubble pump, and the working solution properties must be considered simultaneously.

Refer to Figure 5.1, an energy balance is applied to a control volume, containing generator, pump tube, and separator,

$$\dot{Q}_{in} = \dot{m}_1 h_1 - \dot{m}_3 h_3 - \dot{m}_4 h_4 \tag{5.1}$$

Then, a mass continuity relation is applied to the total mass,

$$\dot{m}_1 = \dot{m}_3 + \dot{m}_4 \tag{5.2}$$

The experimental correlation of a simple bubble pump which was obtained from chapter 4,

$$\dot{V}_3 = -0.00014\dot{V}_4^4 + 0.00625\dot{V}_4^3 - 0.09706\dot{V}_4^2 + 0.63772\dot{V}_4 - 0.46802 \tag{5.3}$$

As the bubble pump correlation is presented in terms of volumetric flow rate, mass flow rate of both vaporized and liquid solution must be transformed from the volumetric flow rate.

$$\dot{m}_3 = \frac{\dot{V}_3}{\upsilon_3} \tag{5.4}$$

$$\dot{m}_4 = \frac{\dot{V}_4}{\upsilon_4} \tag{5.5}$$

Specific volume of the liquid solution, υ_3, and the vapor part, υ_4 , can be calculated from [Threlkeld, 1970]:

$$\upsilon_3 = (1 - X_3)\ \upsilon_{water-liq} + 0.85\ X_3\ \upsilon_{amm-liq} \tag{5.6}$$

$$\upsilon_4 = (1-X_4)\,\upsilon_{water-vap} + X_4\,\upsilon_{amm-vap} \tag{5.7}$$

In the calculation procedure, equation (5.1) to (5.7) are simultaneously solved. To determine mass flow rate of fluid leaving and entering the separator and the generator, the following are required:

1. Concentration of the liquid solution at generator inlet, 1, must be initially assumed. Then, the solution enthalpy (1) needed for calculation in equation 5.1 can be calculated from the correlations [Patek and Klomfar 1998].
2. It is reasonable to assume that liquid and vapor leaving the separator are in equilibrium at their temperatures and pressures. Then their concentration, enthalpy and specific volume could be determined.
3. Mass flows ($\dot{m}_1, \dot{m}_3$ and $\dot{m}_4$) can be determined by solving equations 5.1, 5.2 and 5.3 simultaneously.
4. Concentration of weak solution at the generator inlet (1) must be verified. Conservation of mass of ammonia is recalled for the verification,

$$\dot{m}_1 X_1 = \dot{m}_3 X_3 + \dot{m}_4 X_4 \tag{5.8}$$

Unless it is valid, it is used as a corrected concentration for iteration of the enthalpy used in equation 5.1. All calculation steps are repeated until the obtained value is valid by equation 5.8. Then, the obtained results could be used for further calculations.

These calculation steps are shown as a flow chart in Figure 5.3.

5.2 Rectifier

Vaporized solution is separated from the liquid solution in the separator. The vapor flows upward to be purified in the rectifier while the liquid flows downward through the solution heat exchanger (SHX) to the absorber.

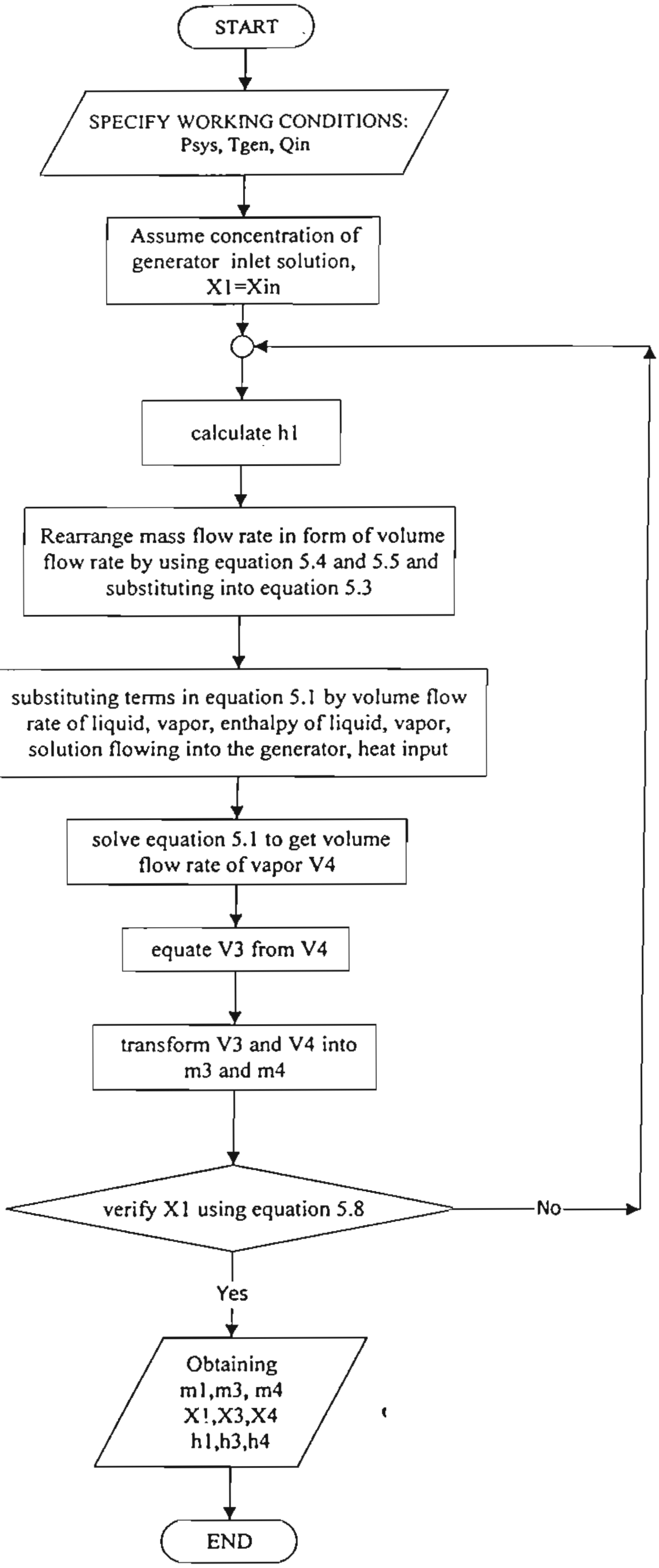

Figure 5.3 A calculation flow chart of the mathematical model.

Water vapor in the vaporized solution is partially condensed in the rectifier. The rectification process is graphically explained as shown in Figure 5.2.

Saturated vapor (4), flowing upward from the separator, is cooled down as two-phase mixture (6) in the rectifier. Condensate obtained from partial condensation of water vapor in the rectifier (5) flows back into the separator. The rectified vapor (7) enters the condenser. Mass flow rate of the purified vapor can be determined from,

$$\dot{m}_7 = \frac{(X_6 - X_5)}{(X_7 - X_5)}\dot{m}_4 \tag{5.9}$$

This amount of ammonia vapor with concentration of X_7 is liquefied in the condenser. There is an amount of heat rejected by partial condensation of water vapor in the rectification process. It is determined from,

$$\dot{Q}_{rec} = \dot{m}_7 h_7 + \dot{m}_5 h_5 - \dot{m}_4 h_4 \tag{5.10}$$

Condensate (5) and liquid solution (3) are aggregated as the strong solution (9). It flows via the SHX to the absorber. Therefore, mass flow rate of the strong solution can be determined from continuity of mass,

$$\dot{m}_9 = \dot{m}_3 + \dot{m}_5 \tag{5.11},$$

with concentration of,

$$X_9 = \frac{\dot{m}_3 X_3 + \dot{m}_5 X_5}{\dot{m}_9} \tag{5.12}$$

5.3 Condenser

The rectified ammonia vapor from the rectifier (7) enters the condenser. It is liquefied by rejecting heat to the cooling water. The amount of rejected heat is calculated from,

$$\dot{Q}_{con} = \dot{m}_7 h_7 - \dot{m}_8 h_8 \tag{5.13}$$