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Sweeping Gas Membrane Evaporative Cooling for the Enhanced Performance of Vapour Compression Refrigeration

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SWEEPING GAS MEMBRANE EVAPORATIVE COOLING FOR THE ENHANCED PERFORMANCE OF VAPOUR COMPRESSION REFRIGERATION

by

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A thesis submitted to the Faculty of Graduate and Postdoctoral Studies as part of the fulfillment of the requirements for the degree of:

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Abstract

A sweeping gas membrane evaporative (SGME) cooling system was developed to remove heat from the condenser of a vapour compression chiller. The chiller was designed for hot environments and could only achieve a coefficient of performance (COP) of 1.0 using an air-cooled condenser. In this work, the condenser was modified to have circulating water absorb heat, which was then cooled in membrane modules by evaporation across a hydrophobic membrane into an air stream. Steady-state operation of the chiller was achieved with the SGME system. Increasing either air flow or water circulation rate increased the COP of the chiller, which reached a value of 1.83 with an optimal heat load of 435 W. The SGME system was then compared to cooling towers, a conventional method of evaporative heat removal. It was found that the SGME system provided more heat removal per unit volume and per unit of air flow than any investigated cooling tower. 89% of the heat removal was attributed to evaporation, and only 1.5% of the water entering the module was lost to evaporation, making the system more compact and efficient than towers. The sweeping gas membrane evaporation system was superior to both air-cooled condensers and to cooling towers.

Résumé

Un système de refroidissement à evaporation utilisant l'entraînement de gaz à travers des membranes (SGME) a été développé pour extraire la chaleur du condenseur d'un refroidisseur à compression de vapeur. Le refroidisseur a été conçu pour des environnements chauds et pouvait atteindre un coefficient de performance (COP) de seulement 1,0 à l'aide d'un condenseur refroidi par l'air. Dans cet étude, le condenseur a été modifié pour faire circuler l'eau afin qu'elle puisse absorber la chaleur. L'eau a été ensuite refroidi par évaporation dans des modules de membrane à travers une membrane hydrophobe dans un courant d'air. Le système SGME a été utilisé pour réaliser le fonctionnement en état stationnaire. L'augmentation du débit d'air ou du taux de circulation de l'eau a augmenté le COP du refroidisseur: on a atteint un COP de 1,83 avec une charge calorifique optimale de 435 W. Le système SGME a ensuite été comparé aux tours de refroidissement, qui sont conventionnellement utilisées pour faire dissiper la chaleur par évaporation. Il a été constaté que le système SGME a fait extraire plus de chaleur par unité de volume et par unité de débit d'air que toute tour de refrodissement étudiée. On peut attribuer 89% de l'extraction de la chaleur à l'évaporation. De plus, seulement 1,5% de l'eau entrant dans le module a été perdue à l'évaporation, ce qui rend le système plus compact et plus efficace que des tours. Le système SGME a été supérieur aux condenseurs refroidis à l'air et aux tours de refroidissement.

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Chapter 1: Introduction and overview

Introduction

Cooling is a process necessary for many applications such as food storage and processing, enclosed space air conditioning, and industrial process cooling. While water-based cooling can take the form of melting ice (Briley, 2004) or transferring heat to liquid water (Kharrufa and Adil, 2008), a third option is evaporative cooling, which makes use of the large heat of evaporation for applications such as refrigeration (Afonso, 2006) and cooling towers (Marques *et al.*, 2009).

Evaporative cooling has many advantages over ice- or liquid-based cooling. Evaporation requires approximately 2400 kJ/kg, which is a much larger amount of energy per unit mass than melting (approximately 330 kJ/kg) or raising the temperature of a water reservoir (4.2 kJ/kg per degree of temperature increase). In hot environments, ice or cooling water requires energy to be produced for a cooling process, but evaporation is favourable at high temperature due to the exponential increase in water saturation vapour pressure with respect to temperature. It has been found that using water evaporation to remove heat from a refrigerator condenser is more efficient than cooling with a stream of liquid water or air (Hosoz and Kilicarslan, 2004), and evaporative cooling towers are an effective method of cooling large quantities of hot water down to as low as the wet bulb temperature of ambient air (Heidarinejad *et al.*, 2009).

Evaporative cooling does have its drawbacks. When liquid water and air are in direct contact with each other, such as in a cooling tower or some air conditioning systems, it is possible for water droplets to be entrained in the passing air, which causes heat transfer between the water and air to occur by conduction rather than evaporation of liquid water into vapour. In addition, liquid water can be a breeding ground for microorganisms that cause ailments such as Legionnaire's disease, which are spread by air passing over the surface of the water (Charles and Johnson, 2008). Commercial cooling towers also tend to be quite large because of the need to spray water into fine droplets that can fall a long distance in contact with air. The corresponding large volume of blown air for a tower leads to noise pollution from fans. A technique that harnesses the large energy transfer of evaporation without the negative effects of air-liquid contact and large volume is desirable.

Membrane evaporation can solve the problems of evaporative cooling. Like the similar technology of membrane distillation, membrane evaporation is the transfer of water vapour – but not liquid water – across a membrane driven by a vapour pressure difference (Curcio and Drioli, 2005). In the case of sweeping gas membrane evaporation, the low humidity of unsaturated air on the permeate side of the membrane provides the low water vapour pressure necessary to force feed water to evaporate and cross the membrane (Khayet *et al.*, 2004). As some feed water evaporates and absorbs heat in the process, the remaining liquid water cools. While this is the basis of cooling towers as well, the addition of a membrane barrier between the liquid water and the air forces most heat transfer to occur through evaporation rather than conduction across the membrane, and prevents material in the water from entering the air stream. Additionally, if a hollow fiber membrane module is used, water does not need to be sprayed into fine droplets, but can remain as streams within the fibers, providing a more compact cooling arrangement. Interestingly, feed cooling has not been seriously investigated in membrane distillation, and the feed temperature is often considered constant, usually because there is a large quantity of pre-heated water or because the feed is a concentrated solution with relatively low vapour pressure (Sudoh *et al.*, 1997). A patent describes a technique of connecting the permeate to vacuum, making use of the extremely low pressure to reach a liquid temperature of close to the freezing point (Lomax and Moskito, 1999). However, operating a vacuum pump requires a significant electrical demand and needs a condenser to prevent liquid water from entering the pump, whereas a sweeping gas evaporative system requires only a blower, and the resulting air/vapour mixture can be rejected safely to the environment if necessary. Membrane evaporative cooling is a new way to improve upon conventional evaporative technology.

Hypotheses

In this thesis, it was proposed that a sweeping gas membrane evaporation system should be coupled with a vapour compression refrigeration cycle. Two hypotheses were proposed based on this arrangement:

- A small membrane cooling device should provide the cooling capacity required for the steady-state operation of a commercial chiller, and do so with higher performance than a conventional air-cooled condenser.
- Sweeping gas membrane evaporation removes more heat per unit volume and is more efficient in the use of air than conventional evaporative cooling technology, such as cooling towers.

Overview of thesis

This thesis demonstrates the effectiveness of a membrane evaporative cooling system coupled with a chiller, and how it improves upon conventional cooling technology.

Chapter 2 is a review of the literature related to membrane and cooling technology. There are a variety of methods for lowering temperatures, but evaporative techniques will be explored because of their effectiveness over ice- and liquid-based methods, with particular emphasis on refrigeration and cooling towers. Refrigeration is investigated because of the use of a vapor-compression cycle in a chiller, whose performance can be compared to other systems through the use of a coefficient of performance (COP). Since our membrane cooling system replaces an air-cooled condenser, condensers will also be investigated. Cooling towers are the conventional method of using air to cool water via evaporation, so they are explored to provide a background for this new evaporative cooling method. Finally, membrane evaporation and the related technique of membrane distillation are reviewed as a foundation for explaining the heat and mass transfer within a membrane-based system.

Chapter 3 provides detail about the apparatus and experimental method. A chiller was used to cool down a water bath heated by immersion heaters. Though the chiller was originally designed with an air-cooled condenser, the condenser was modified so that a stream of water could absorb the heat. The heated water then entered three hollow fiber membrane modules. Air passed on the permeate side to sweep away vapour, while the remaining water was cooled and returned to the condenser. Though the amount of water in circulation decreased over time, runs continued for approximately two hours until recorded temperatures and the rate of evaporative water removal became constant. The chiller provided steady-state cooling under these conditions.

Chapter 4 is a paper focused on the chiller and condenser of the cooling system, which are related to the first hypothesis. It demonstrates that steady-state cooling can be achieved, and uses the coefficient of performance to determine the effectiveness of this system compared to others. Parameters such as heat load, air flow, and water circulation rate were also adjusted to plot their respective effects on the chiller's performance. The modified condenser improved the COP to roughly double what could be achieved with an air-cooled system, with less air in total. Chapter 4 also explains the reasoning for how the three modules were arranged to optimize performance, and explains the importance of condenser heat removal for the performance of a chiller.

Chapter 5 is a paper focused on the membrane evaporation process, which is connected to the second hypothesis. Like the previous chapter, it demonstrates steady-state evaporation and shows the effects of parameters such as heat load, air flow, and circulation rate on the quantity of cooling (rate of evaporation) and the quality of cooling (cooled membrane outlet temperature). It is demonstrated that the membrane evaporation system provides enough cooling to allow the chiller to operate at steady state. Since cooling towers are the conventional evaporative cooling technology with wide commercial applications, the cooling performance of this membrane system is compared to different types of cooling towers. When normalized for volume and feed temperature, membrane modules provide a more compact method of evaporative cooling with the additional benefit of not having liquid water and air come in direct contact with each other.

Chapters 6 and 7 link the previous two chapters and give conclusions and recommendations. Chapter 4 (refrigeration) and Chapter 5 (membranes and evaporation) cover different fields of study; the final chapters explain how these concepts are linked together for the combined effort of producing an efficient chiller that can provide cooling below the temperatures that can be achieved with a cooling tower, which is itself less compact than our new membrane-based system. It concludes that the new membrane-based system is the most effective method of combining the best aspects of refrigeration technology and evaporative cooling. Nevertheless, there are new avenues to explore in this novel technique, so some recommendations for future studies are provided.

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Chapter 2: Literature review

Evaporative cooling

Cooling is necessary in a variety of domestic and industrial applications. Air conditioning (Afonso, 2006; Izquierdo *et al.*, 2008), refrigeration (Pearson, 2008; Hosoz and Kilicarslan, 2004), electronics cooling (Baummer *et al.*, 2008; Sauciuc, 2005; Kim *et al.*, 2008), and personal cooling garments (Webster *et al.*, 2005; Cadarette *et al.*, 2006) are all examples of applications where cooling below room temperature is needed in an environment at higher temperature. Water is a useful cooling medium in these situations because it is ubiquitous, non-toxic, and has excellent physical properties for heat transfer.

Water-based cooling can be summarized by the methods by which water absorbs heat. Melting ice (Briley, 2004) or snow (Hamada *et al.*, 2007) allows an object's temperature to be cooled to the freezing point of water, and the heat of fusion, the heat absorbed by the melting process, is approximately 330 kJ/kg. It is also possible to use solutions or other liquids to lower the freezing point and drive the cooling temperature lower. While this is an effective cooling method, it requires energy to produce the ice initially, which may not be effective in warmer climates. When cooling to temperatures that are above the freezing point of water, heat can be transferred to a reservoir or stream of cold liquid water, such as a stream in a heat exchanger (Incropera and deWitt, 2002) or a reservoir moderating the temperature of a building (Kharrufa and Adil, 2008). Water has a high specific heat capacity (4.2 kJ/kg/K) compared to other cooling media such as air, and a flow of water along a heat exchange surface provides a high heat transfer coefficient. This method is effective when large quantities of cooling water are available, but it is less practical when a limited amount of water is available, where heat transfer would significantly raise the temperature of the small reservoir. Evaporation is a third method of water-based cooling (Alizadeh, 2008), making use of the large heat of evaporation of water (approximately 2400 kJ/kg). The large energy requirement of phase change makes evaporative cooling effective in processes such as refrigeration or cooling towers.

There are advantages and disadvantages to using evaporative cooling compared to other physical transformations. Evaporation takes in much more heat per unit mass than melting or sensible heat transfer, and evaporation can occur at all temperatures between the triple point (0.01°C) and boiling point (100°C at atmospheric pressure) of pure water. However, despite the wide range of temperatures, evaporation only becomes effective at higher temperatures, such as above 40°C, which is reflected in the exponential increase in vapour pressure relative to temperature as shown in the Clausius-Clapeyron equation (Smith and Van Ness, 2005):

$$\ln P = -\frac{\Delta H}{RT} + C \tag{1}$$

As such, evaporation is minimal at the low temperatures often desired by cooling. Another drawback is the increase in volume or pressure of the produced vapour relative to the liquid; for the same mass of water, an evaporation-based heat exchanger would take up more space than an exchanger that only raised the temperature of liquid water. Finally, evaporative processes often humidify the air (Johnson *et al.*, 2003; Charles and Johnson, 2008), which can be uncomfortable in hot environments and can encourage the growth of microorganisms.

Based on these benefits and drawbacks, evaporative cooling is most effective for removing heat from hot sources, such as in a cooling tower.

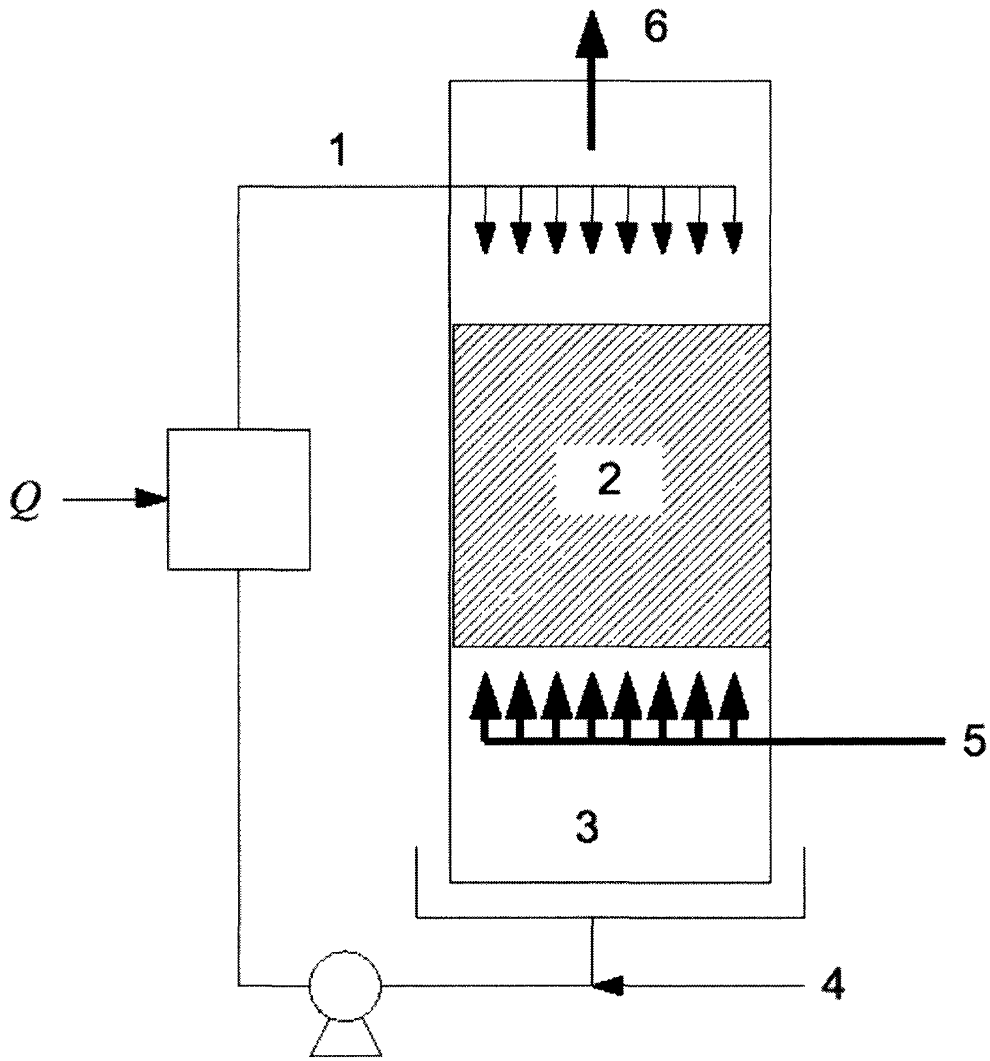

Figure 2.1: Schematic of a cooling tower (adapted from Marques *et al.*, 2009). 1: Hot inlet water; 2: packing; 3: tower basin; 4: makeup water; 5: inlet air; 6: heated outlet air

Cooling towers

In industry, a common method for cooling hot water to near room temperature is with a cooling tower. As shown in Figure 2.1 (Marques *et al.*, 2009), a cooling tower consists of a stream of hot water that is sprayed downward through the tower to a collection basin. As the water droplets descend, they come into contact with unsaturated air, which causes the water to lose heat by evaporation as the air becomes saturated and heated (Khan *et al.*, 2003). A packed bed or baffles can be used within the tower to increase contact time between the water and air.

Cooling towers are designed in a variety of sizes and arrangements. While cooling towers used in industry have volumes that are typically larger than 1000 m^3 (Marques *et al.*, 2009; Kairouani *et al.*, 2004), bench-scale towers have been studied with volumes less than 1 m^3 (Khan *et al.*, 2003; Naphon, 2005). Nearly all towers spray water downward into fine droplets, but there are variations in how the air is supplied to the tower. Air can enter the tower by natural draft from the surrounding environment (Pearlmutter *et al.*, 2008) or it can be blown in with fans (Naphon, 2005). Air can flow upwardly counter-current to the falling water droplets (Heidarinejad *et al.*, 2009) or it can make use of a cross-flow arrangement by having air enter at several points throughout the height of the tower in a direction perpendicular to the falling water (Hajidavaloo *et al.*, 2010).

Various studies have investigated the performance of these different sizes of cooling towers. A sampling of results from recent studies into cooling towers is shown in Table 2.1.

Table 2.1: Parameters of cooling tower studies

	Khan, 2003	Naphon, 2005	Qureshi, 2006	Marques, 2009	Hajidavaloo, 2010
air flow	counter-current	counter-current	counter-current	counter-current	cross-flow
packing	void	plastic plates	void	void	wood blocks
volume (m^3)	0.7	0.011	203.2	1785	633.6
air flow (L/s)	989	60	94 000	285 000	536 000
water (kg/s)	1.78	0.04	94	722	953
water in (°C)	28.7	39	40	40	58
water out (°C)	25.2	23	18	31.3	30
cooling (kW)	26.2	2.7	8685.6	26381	112073
cooling (kW/m^3)	37.4	245.5	42.7	14.8	176.8

Cooling towers are effective at removing heat from water until it approaches the wet bulb temperature of the incoming air (Qureshi and Zubair, 2006). Applications where this is practical include cooling excess hot process water that must be rejected to the environment or cooling a stream of water that must be recycled back to a system. Some advantages of cooling towers over other cooling systems – such as a liquid-liquid heat exchanger – are the extensive contact area of all the droplets for good heat transfer and the fact that evaporation at high temperatures removes much more heat per unit mass than conducting sensible heat away from a liquid.

There are drawbacks to cooling towers associated with their size and operating temperatures. Cooling towers in industry have a large footprint. The commercial tower's volume exceeds typically 1000 m^3, and there is also additional equipment necessary to blow in the large quantity of air needed for evaporation. Larger towers have more cooling capacity and longer residence time for cooling droplets, so by necessity they are used instead of several small bench-scale towers. Since the interior volume contains both water droplets and air, more space is needed than a system that just has a continuous stream of water, such as a heat exchanger. The large volume of air requires blowers, which contribute to noise pollution in the surrounding area. With regard to temperature, the water outlet temperature is limited by the conditions of the entering air. If equilibrium between the exiting water and entering air is achieved, the water would approach the wet bulb temperature of the air. Since air is often drawn from the environment, seasonal fluctuations in temperature and humidity would result in inconsistent cooling temperatures throughout the year; the warm and humid air of summer would have a higher wet bulb temperature, diminishing the cooling quality in the season when cooling is most desirable. Cooling towers are limited by its need for large volume and the inability to drop below the wet bulb temperature of the incoming air.

Refrigeration

Refrigeration is a method of cooling that can allow temperatures to be driven below that of the surrounding environment. Refrigeration is based on a heat pump cycle to take heat from a cool source and reject it to a warmer heat sink. It is the principle behind most air conditioning, food and chemical storage, and industrial cooling.

There are several ways to achieve a refrigeration cycle. Early refrigeration systems were based on ammonia or sulfuric acid in an absorption refrigeration cycle, which makes use of an evaporated working fluid absorbed into a solution as an intermediate step (Afonso, 2006). In an absorption refrigeration cycle, a low-pressure working fluid – such as ammonia – is evaporated at low temperature using heat from a low-temperature source at the evaporator. The resulting vapour is then absorbed into a secondary fluid, such as water, and the solution is later heated at higher temperature to separate the refrigerant vapour from the solution. The hot refrigerant vapour is then condensed, rejecting heat to a heat sink, and an expansion valve reduces the pressure of the fluid before it returns to the evaporator to complete the cycle. An advantage to the absorption refrigeration cycle is that it is possible to run it without any electricity. The primary energy input occurs as the solution is heated to separate the refrigerant from the solution; this heating can come from non-electrical heat sources such as solar energy (Kim and Infante Ferriera, 2008). The combination of ammonia absorbed into water is common for systems where the temperature must be driven below the freezing point of water (Ziegler, 1999), while the combination of water absorbed into a concentrated desiccant solution – such as LiBr – makes use of the higher heat of evaporation of water but is only used down to the freezing point of water (Aphornratana and Sriveerakul, 2007). The ammonia-water and water-LiBr systems are the most prevalent combinations used in absorption refrigeration systems today, and are considered environmentally friendly when minimal electricity and non-toxic working fluids are used.

The more widespread method of refrigeration is the vapour compression cycle. In the vapour compression cycle, the intermediate step of absorbing and separating the working fluid is removed and replaced with a compressor (Afonso, 2006). The compressor raises the pressure and temperature of the evaporated refrigerant, but requires electricity for the work input. A diagram of the vapour compression cycle is shown in Figure 2.2. Starting at the evaporator, where the refrigerant evaporates via heat taken in from a cool heat source, the resulting vapour is compressed to raise its temperature and pressure. The hot vapour is sent to the condenser, which is a heat exchanger between the refrigerant and a hot heat sink that allows the refrigerant to reject heat through condensation. The condensate is then expanded, which cools and lowers the pressure of the liquid refrigerant before it completes the cycle at the evaporator.

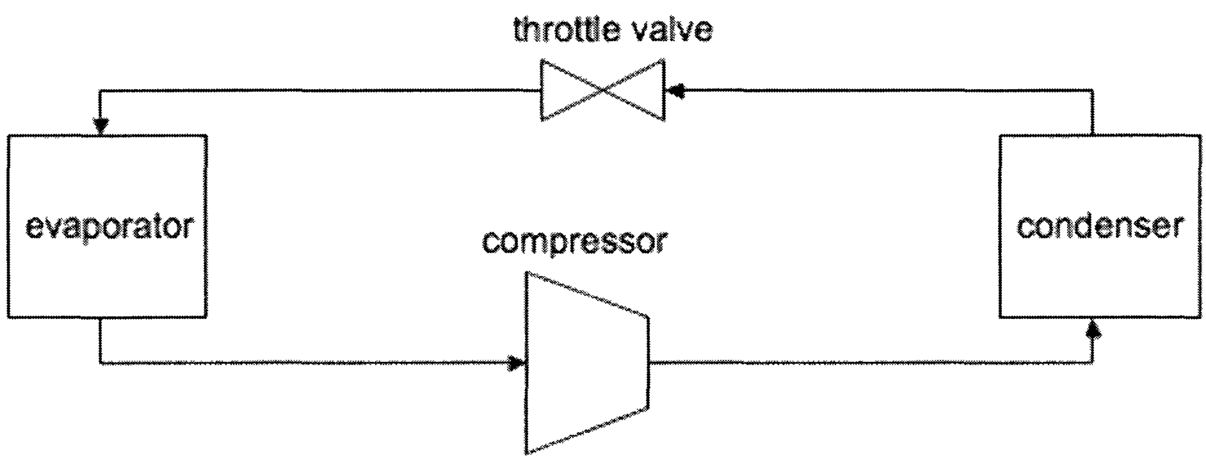

Figure 2.2: The vapor-compression refrigeration cycle

Refrigeration cycles, along with other heat engines, can be measured by its coefficient of performance (COP) (Smith and Van Ness, 2005). COP is a ratio of the heat output of the system against work required by the system. For refrigeration cycles described in this work, COP is defined as:

$$\mathrm{COP}_{\mathrm{cooling}} = \frac{\text{heater load}}{\text{electrical demand of compressor}} \tag{2}$$

The Carnot cycle is a theoretical heat engine that provides the highest COP possible. When the absolute temperatures of the hot and cold sinks of a cycle are known, the maximum COP for those conditions is defined as:

$$\mathrm{COP}_{\mathrm{cooling,\ max}} = \frac{T_C}{T_H - T_C} \tag{3}$$

The vapor-compression cycle is widespread in the refrigeration market because of its many advantages. Even though the cycle requires electrical input rather than the heat input of the absorption refrigeration cycle, usable heat is not always reliable and is a less efficient method of increasing the pressure and temperature of the evaporated refrigerant. When considering the coefficient of performance, the ratio of cooling performance to energy input at the compression site, absorption systems can achieve COPs of 0.41 for ammonia (Wang and Li, 2007) and 0.52 for water-LiBr (Aphornratana and Sriveerakul, 2007). This implies that for every watt of power added for heating the solution, the cooling effect at the evaporator is only about half of that. In contrast, vapor-compression cycles can achieve a COP in excess of 4.0 (Pearson, 2008). The vastly higher COP is the primary reason for choosing vapour compression over absorption, and the transition from ozone-depleting CFC refrigerants to safer refrigerants in the late 20th century helped reduce the environmental impact of refrigeration. As such, vapor-compression cycles are preferred for most household and commercial cooling, leaving absorption systems to be used in places where excess heat is available, such as in some industrial applications.

Within the vapor-compression cycle, it is desirable to increase the COP. For a given cooling requirement, an increase in COP results in less work at the compressor, which reduces the electrical demand of the system. One way to reduce the compressor work is to change the refrigerant, since some vapors will compress more easily than others. As an example, for identical systems, ammonia produces a COP of 4.84, compared to tetrafluoroethane at 4.60 or carbon dioxide at 2.96 (Pearson, 2008). Another way to reduce the compressor work is to reduce the condenser temperature. Since pressure increases with temperature, a reduction in outlet temperature for the compressor reduces the outlet pressure and thus the amount of work that must be done. While a similar reduction in compressor work could be achieved by raising the evaporator temperature, the COP will approach infinity (no work) as the evaporator temperature reaches that of the fluid providing the cooling to the refrigerant in the condenser.

Lowering the condenser temperature will raise the COP in vapor-compression systems. When condensers are designed as heat exchangers, the condensation temperature can be lowered either by lowering the temperature of the cooling fluid absorbing heat from the refrigerant or by improving the heat transfer. Lowering the temperature of the final cooling fluid can be difficult, since for many systems, it is air drawn from the environment, which is often warm in situations where cooling is required. As such, it is necessary to improve the heat transfer at the condenser. Air and water are common fluids for cooling the condenser, but making use of the evaporation of water at the condenser provides the most heat removal and improves the COP (Hosoz and Kilicarslan, 2004). For similar conditions, switching from air-cooled to evaporative-cooled condensers improves the COP from 2.7 to 3.3, and the evaporative system can reach a COP as high as 4.0. Thus, an improvement in heat transfer available from the high heat of evaporation of water provides an increase in COP to desirably reduce the electrical demand of a refrigerator.

Membranes in cooling applications

Membranes are semipermeable barriers used for separation applications. Based on pore size and hydrophobicity of the surface, certain molecules will be able to cross the membrane favourably instead of others, providing the foundation of technologies like reverse osmosis. The separation properties of membranes have also been integrated into experimental cooling systems.

The absorption refrigeration cycle is a promising application for membranes in cooling. In the absorption cycle, a solution is heated to separate the hot refrigerant vapour. This process requires a lot of heat input, some of which ends up heating the residual solution rather than just the vapour. It has been proposed that the energy demand can be reduced by using membrane separation in this step (Kim *et al.*, 2008). This is particularly important for the water-LiBr absorption system, where water has a chemical affinity to remain as hydrates in the solution and has a high heat of evaporation that is more effective at high temperatures (Sudoh *et al.*, 1997). By reducing the pressure on the permeate side of the membrane, water selectively crosses the membrane without needing high temperatures for evaporation (Riffat *et al.*, 2004). While there is more work necessary to reduce the pressure on the permeate side of the membrane, the trade-off is justified by the reduced heating energy demand for water evaporation. The separation can be further enhanced by using a hydrophobic membrane.

Since liquid water will not wet the membrane surface, only water vapour can cross the membrane. This forces the water in the solution to evaporate before crossing, and this evaporation can occur at lower temperatures if the partial vapour pressure on the permeate side is kept low as well, such as through a cool stream of water (Sudoh *et al.*, 1997). In a paper that measured COP for their membrane pervaporation integration, a COP of only 0.06 was achieved, so they suggest that there was not enough residence time for the fast-circulating solution to complete its separation (Riffat *et al.*, 2004).

A separate patent discloses the use of membranes in a process similar to the vapor-compression cycle (Yoshimi *et al.*, 2004). In their system, the evaporator and condenser are replaced with two membrane modules that allow the passage of water vapour, but not liquid water. At the evaporator module, some water is evaporated from a passing stream. This evaporation removes heat from the stream, which then continues for cooling applications. The evaporated water crosses the first membrane, and is then compressed to a higher pressure before entering the condenser. At the condenser, the high-pressure vapour crosses a second membrane to enter a low-pressure stream of liquid water, which becomes heated after the vapour condenses inside the stream. The heated stream is then sent to a cooling tower. This patent uses membranes not for the separation of water from a solution, but for the separation of two phases of water. While this patent makes use of the large heat of evaporation of water, it is not a true cycle in that feed water must be added at the evaporator to balance what is lost at the cooling tower, and there appears to be no mechanism to prevent condensation from occurring inside the vapour side of the condenser module. The COP of this system is not provided, but its use of the evaporation of water for cooling should be beneficial.

Membrane evaporation is the use of a membrane to assist in the separation of a vapour phase from its liquid phase. In this process, sweeping gas is often used as the permeate to limit heat transfer across the membrane (Johnson *et al.*, 2003), but vacuum can also be used (Lomax and Moskito, 1999). Air humidification is a method of air conditioning in hot dry climates (El-Dessouky *et al.*, 2000), but humidifying air from standing water increases the risk of introducing bacterial diseases such as Legionnaire's Disease (Charles and Johnson, 2008). Using a membrane to separate liquid water from its vapour allows air humidification for cooling while still preventing liquid water and any contaminants from entering the air. The air is cooled less than 2°C in this humidification process (Johnson *et al.*, 2003), but a patent claims to provide enough cooling to approach freezing by using the low temperatures associated with vacuum vapour pressure (Lomax and Moskito, 1999).

Membrane evaporation and distillation

Membrane distillation is an alternative to conventional evaporative concentration and filtration techniques for the separation of water from a solution (Curcio and Drioli, 2005). Unlike evaporative concentration, membrane distillation is driven by an induced vapour pressure difference that maintains a flux of water vapour without the need for as much applied heat. Unlike filtration, vapour passes across the membrane instead of liquid, which limits the flux when there is not much difference in concentration, but becomes very effective when extracting water from a solution of high concentration.

In literature, membrane distillation has been divided into four distinct mechanisms based on the method of producing the vapour pressure difference across the membrane (Lawson and Lloyd, 1997). Vacuum membrane distillation (VMD), sweeping gas membrane distillation (SGMD), air gap membrane distillation (AGMD), and direct contact membrane distillation (DCMD) are all used to extract water from solutions of high concentration. The reverse step of extracting water from a dilute solution to a concentrated solution is referred to as osmotic membrane distillation (OMD), but it has been classified separately from other forms of membrane distillation. Schematics of membrane distillation methods and OMD are shown in Figure 2.3. All mechanisms make use of a hydrophobic membrane – such as PTFE – that allows water vapour, but not liquid, to cross, forcing the separation to come from evaporation driven by a vapour pressure difference.

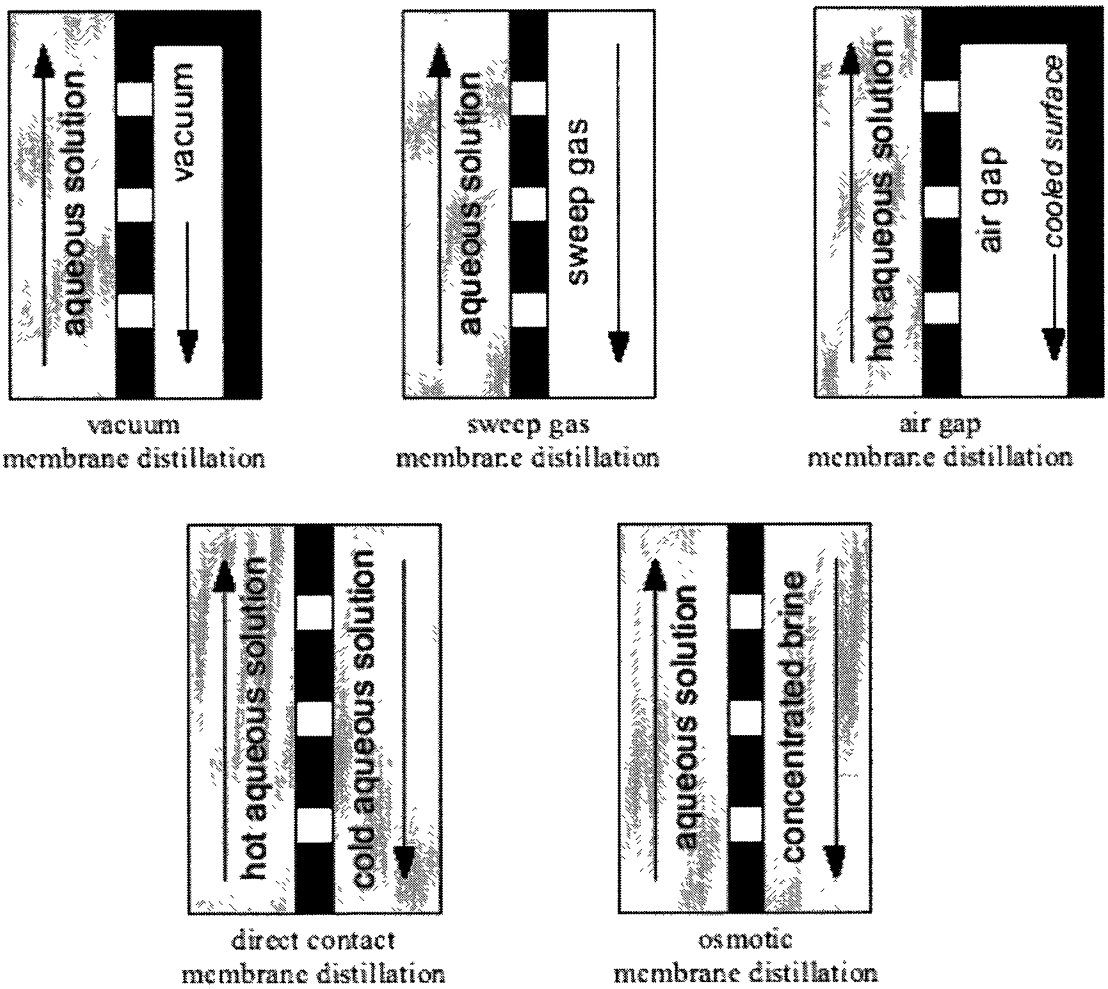

Figure 2.3: Schematic of membrane distillation methods (adapted from Lawson *et al.*, 1997)

There are five ways to increase the vapour pressure difference in membrane distillation. In vacuum membrane distillation, a vacuum reduces the overall pressure on the permeate side (Cabassud and Wirth, 2003). If the membrane support is strong enough to withstand the negative pressure difference, the large vapour pressure difference of VMD produces a high flux of evaporation. Drawbacks to this method include the need to supply electricity to produce the vacuum and that the overall pressure decrease allows any volatile component to separate as well. In addition, most vacuum pumps do not tolerate condensates very well, forcing a condensation system to be introduced to separate the water vapour from the vacuum stream before the pump.

Sweeping gas membrane distillation uses a stream of dry air on the permeate side to draw vapour across the membrane based on a humidity difference (Khayet *et al.*, 2004). Warm, dry air can take in water vapour before it becomes saturated, and a larger stream of air allows more water vapour to be carried away. Unlike VMD, SGMD only needs a blower to supply air and does not depend on a total pressure difference. However, if the pressure of air exceeds that of the liquid water, air can cross the membrane into the water stream. Another disadvantage is the need to either dry or heat the incoming air to increase the capacity of water vapour it can take in before saturating.

Air gap membrane distillation makes use of a cooled surface to condense vapour and keep the vapour pressure on the permeate side low (Alklaibi and Dior, 2005). The stagnant air in the permeate side limits any heat transfer due to conduction or convection, but it also limits the flux to the rate of diffusion of water vapour through air. As such, AGMD has the lowest evaporative flux of the different membrane distillation mechanisms (El-Bourawi *et al.*, 2006).

Direct contact membrane distillation has a hot feed contrasted with a cool stream of water along the permeate side to produce a vapour pressure difference based on a difference of temperature (Kurokawa and Sawa, 1996). By models such as Antoine's equation, vapour pressure is lower at lower temperatures. The heated feed may have a reduced vapour pressure due to the presence of other components in solution, but it is higher than that of the pure, but cold, water stream on the permeate side. DCMD is popular because it requires no special apparatus on the permeate side and only needs a heat source for the feed, which is readily available in most processes. DCMD has been proposed as the mechanism for water-LiBr separation in an absorption refrigeration cycle because of this inherent heating requirement (Sudoh *et al.*, 1997). However, since liquids have higher thermal conductivity than air, it is more difficult to maintain the separate temperatures across the membrane. The hot feed ends up transferring heat to the cold permeate not only through the evaporation process, but also through conduction across the membrane. This detriment is compounded by temperature polarization, where the feed cools slightly at the membrane surface because of the loss of heat from evaporating water, which reduces the vapour pressure in the feed and thus the driving force for separation. Because there is a liquid stream on the permeate side as well, temperature polarization can occur there as well, where the condensing vapour heats the water near the surface of the membrane, increasing the vapour pressure on the permeate side. Nevertheless, temperature polarization is often counteracted by adding more heat to the feed or increasing the turbulence in the streams (Curcio and Drioli, 2005).

A fifth method of separation is osmotic membrane distillation, where the permeate side consisted of a concentrated brine with a very low vapour pressure (Celere and Gostoli, 2002). This process is sometimes ignored compared to the previous four methods because of the impression that the driving force is a concentration difference rather than a vapour pressure difference. However, despite the "osmotic" name attached to it, liquid water does not cross the membrane, and water vapour is driven across based on the inherently low vapour pressure of concentrated solutions. OMD does not require the feed and permeate to be at different temperatures and there is no total pressure difference, making it a useful separation technique where high temperatures in the feed would be detrimental, such as in juice concentration. However, the hygroscopic brines used in OMD can be corrosive to piping and damaging to some membranes, so regular maintenance is needed in this operation.

Membrane evaporation is a similar method that sends water vapour across a membrane, but the feed is pure water. This technique has been used for humidification in air conditioning (Johnson *et al.*, 2003), and is disclosed in a patent that describes approaching the freezing point of water by making use of vacuum pressure (Lomax and Moskito, 1999). Table 2.2 explores the differences between the separation applications of membrane distillation and the cooling applications of membrane evaporation.

Table 2.2: Comparison of membrane evaporation and distillation

	Membrane evaporation	Membrane distillation
purpose	cooling; humidification	separation
liquid composition	pure water	solution
operational T	relatively low (20-60°C)	hot (50-80°C)
T drop along feed or permeate	maximize	minimize
stage of development	underexplored	well established
operational modes	VME, SGME	VMD, SGMD, AGMD, OMD, DCMD

Even though membrane distillation and membrane evaporation serve different purposes, there are similarities between the two techniques. Both systems make use of a vapour pressure difference across the membrane to drive the flux of vapour. By using hydrophobic membranes, no liquid water can cross the membrane, forcing all of the heat and mass transfer to come from evaporation. Interestingly, the use of a hydrophobic membrane to restrict the transfer of liquid (and its sensible heat) serves well for both techniques: membrane distillation does not want any heat from liquid water to enter the permeate and disrupt the temperature difference, while membrane evaporation depends on water transferring heat via evaporation rather than sensible heat because evaporation transfers more heat for less mass.

Heat and mass transfer in membrane evaporation

Membrane evaporation, and the related technique of membrane distillation, depends on the transfer of water vapour across a membrane driven by a vapour pressure difference. While the flux is proportional to the vapour pressure difference (Khayet *et al.*, 2004), transfer is limited by diffusion across the membrane, making mass and heat transfer across the membrane a significant aspect of membrane distillation study.

Diffusion of water vapour through small pores is usually restricted to Knudsen diffusion. Knudsen diffusion is a regime where a molecule is more likely to collide first with a pore wall than another molecule of its phase, such that the mean free path is greater than the pore size of the membrane (Curcio and Drioli, 2005). For gas through a membrane, the effective Knudsen diffusion coefficient, from the Dusty Gas Model is (Curcio and Drioli, 2005):

$$D^{K}_{\iota,eff} = \frac{2\varepsilon r}{3\tau}\sqrt{\frac{8RT}{\pi M_{\iota}}} \tag{4}$$

where M_ι is the molecular weight of the component, and ε, r, and τ are the membrane's porosity, pore radius, and tortuosity, respectively. Based on Knudsen and molecular diffusion, an equation to model the evaporative flux N for SGMD is (Khayet *et al.*, 2004):

$$N = \frac{M}{RT}\left[\frac{3\tau}{4\varepsilon r}\left(\frac{\pi M}{2RT}\right)^{1/2} + \frac{\tau P_{air}}{\varepsilon PD}\right]^{-1}\frac{\Delta P_v}{\delta} \tag{5}$$

where δ is the membrane thickness. It should be noted that temperature, overall pressure, and membrane properties could be assumed constant in this approximation, leading to a simplification that evaporative flux is proportional to vapour pressure difference. While the assumption of constant temperature is valid in many membrane distillation applications, where temperature drops are intentionally minimized and evaporation from solutions is low, it does not account for the temperature changes experienced in studies investigating membrane evaporation. Unfortunately, even studies that investigated membrane evaporation for air conditioning (Johnson *et al.*, 2003; Charles and Johnson, 2008) assumed a constant feed water temperature and observed only air temperature differences. However, within those air humidification studies, the change in air temperature was slight, and with water having a higher heat capacity than air, it is likely that any change in water temperature in those studies would also be minimal. Nevertheless, if water evaporates from a stream that is insulated from the environment, it should cool the remaining liquid based on a heat balance:

$$m_{water} C_P \left(T_{in} - T_{out}\right) = m_{evap} \Delta H_{vap} \tag{6}$$

or rearranged as:

$$T_{out} = T_{in} - \frac{m_{evap} \Delta H_{vap}}{m_{water} C_P} \tag{7}$$

Since the amount of water lost to evaporation is small compared to the total flow of water – even open-air cooling towers experience as little as 0.5% loss (Pearlmutter *et al.*, 2008) – the drop in temperature of the liquid stream would also be expected to be small in most membrane applications. With small temperature drops and models depending on a constant feed temperature, it is often a fair and frequent assumption to expect no temperature change in the feed.

There are still some temperature changes that are expected. Temperature polarization is the effect of temperatures on either surface of the membrane approaching a point of similar vapour pressure. When the surface of either side of the membrane is at the same vapour pressure, regardless of the bulk properties, there is no flux across the membrane. Even though the bulk of the feed and permeate are arranged to have a very high vapour pressure difference, the evaporation and condensation occurring at the surfaces provides detrimental temperature changes. For the case of SGMD, as feed water evaporates at the membrane surface, the evaporation removes some heat from the surrounding water, which cools the surface temperature slightly. On the permeate side, some of the incoming vapour condenses, raising the temperature of the permeate surface slightly. Based on the surfaces, a decrease in feed temperature and increase in permeate temperature reduce the vapour pressure difference between the sides, which in turn reduces the evaporative flux. A temperature polarization coefficient to quantify this effect was proposed (Curcio and Drioli, 2005):

$$TPC = \frac{T_{feed}^{mem} - T_{permeate}^{mem}}{T_{feed}^{bulk} - T_{permeate}^{bulk}} \tag{8}$$

where a TPC of 1 would imply no change between the bulk and the surface, and 0 would suggest that there is no transfer because of identical vapour pressure. Real systems are designed to account for a TPC between 0.4 and 0.7. To counter the effects of temperature polarization, it was found that increasing the feed velocity – to disrupt the lamina near the membrane surface – and increasing feed temperature – to exponentially increase the feed vapour pressure – improved the flux of vapour (El-Bourawi *et al.*, 2006). Finally, conduction across the membrane is still accounted for in models, especially in DCMD where two streams of liquid are involved, but also in SGMD where a fast-moving stream of air removes heat by convection. The conduction transfer coefficient across the membrane is (Curcio and Drioli, 2005):

$$k_m = (1-\varepsilon)k_{polymer} + \varepsilon k_{vapour} \tag{9}$$

Since the conductivity of vapour is usually less than that of polymers, it is encouraged to have as porous a membrane as possible to reduce conduction. A porous membrane also improves the mass transfer, as suggested by Equation (5), but it cannot be so porous that the liquid entry pressure of the membrane reaches the pressure difference across the membrane; this would allow liquid water to cross, diminishing the evaporative benefits of membrane distillation. As such, hydrophobic PTFE membranes are often used in membrane evaporation and distillation, which can accommodate pore sizes of more than 2 microns while still acting as a barrier to liquid.

While the heat and mass transfer effects of membrane evaporation and distillation have been investigated, they are limited by one assumption. Models assume that the feed temperature remains constant during its residence time in the membrane module, which is fair for systems with a short residence time or limited evaporation, but limits the investigation of cooling effects in the processes. In the future, more studies should look into the effect of evaporation on feed temperature as suggested by Equation (7). Nevertheless, the models of mass and heat transfer should still remain valid for points along a membrane surface where temperatures are known.

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Chapter 3: Experimental apparatus and procedures

Introduction

A sweeping gas membrane evaporation cooling device was coupled with a heat pump to increase the range of temperatures for cooling. Evaporation of water into air can only drive the temperature of the water down to the wet bulb temperature of the incoming air; for most conditions, this restriction makes it difficult to cool water below 20°C in evaporative systems such as cooling towers (Heidarinejad *et al.*, 2009). To cool to lower temperatures, a refrigeration cycle can be added, where the SGME provides cooling at the condenser, the hottest point in the cycle. The evaporator of the refrigerator is then used to provide cooling at a temperature below the wet bulb temperature of the environment. It has been shown that making use of the evaporation of water at a refrigerator condenser is more energy-efficient than the conventional technology of cooling the condenser with air or water (Hosoz and Kilicarslan, 2004).

Apparatus

A chiller was coupled with a sweeping gas membrane evaporation system to provide cooling for a heat source. A schematic is shown in Figure 3.1.

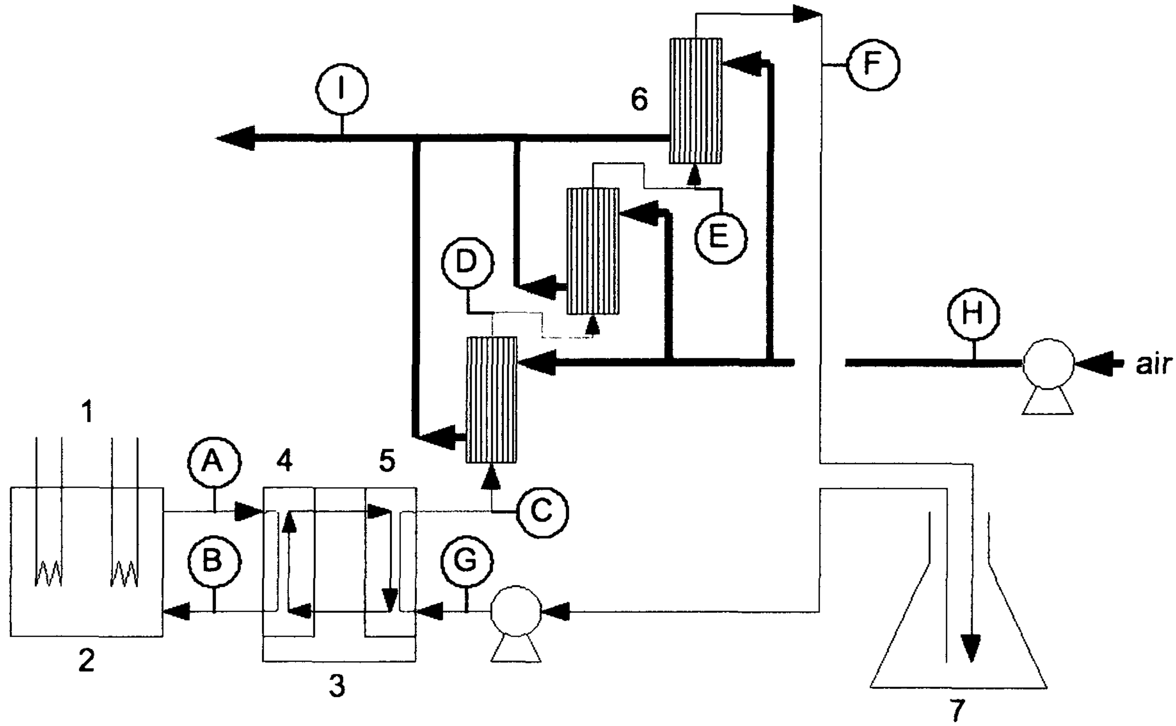


Figure 3.1: Schematic of a SGME/chiller system. 1: immersion heaters; 2: water bath; 3: chiller; 4: evaporator; 5: condenser; 6: hollow fiber membrane modules; 7: water reservoir; A to I: thermocouples

The heat source that was cooled by this system was a set of immersion heaters in a water bath. The immersion heaters (1; Elite Submersible by Hagen, Montreal, QC) provided 200-600 W of heat, adjusted by the number of heaters added. The heaters did have some fluctuation in power output depending on the temperature, so an energy meter (EM100 at Canadian Tire, Ottawa, ON) was used to measure the average heat output. The heaters were immersed in a circulating water bath (2; Cole Parmer, Montreal, QC), which circulated water between the water bath with heaters and the evaporator (4) of the chiller (3).

The chiller was provided through a corporate research collaboration. It was designed to cool water to 15-20°C in hot environments by making use of fan-blown air to cool the condenser. The condenser (5) was modified to be a stainless steel plated heat exchanger (GBM by FlatPlate, York, PA) with a volume of 0.5 L to allow heat to be removed with liquid distilled water. The chiller was charged with R134a refrigerant (1,1,1,2-tetrafluoroethane), which allowed the evaporator temperature to drop as low as 5°C and the condensation temperature could reach as high as 75°C before the chiller shut down due to overheating for protection. During operation, the flow rate through the evaporator was approximately 25 g/s. Also during operation, the electrical demand of the chiller was determined by measuring the current drawn by the compressor from a 28 V power source (Cole Parmer, Montreal, QC).

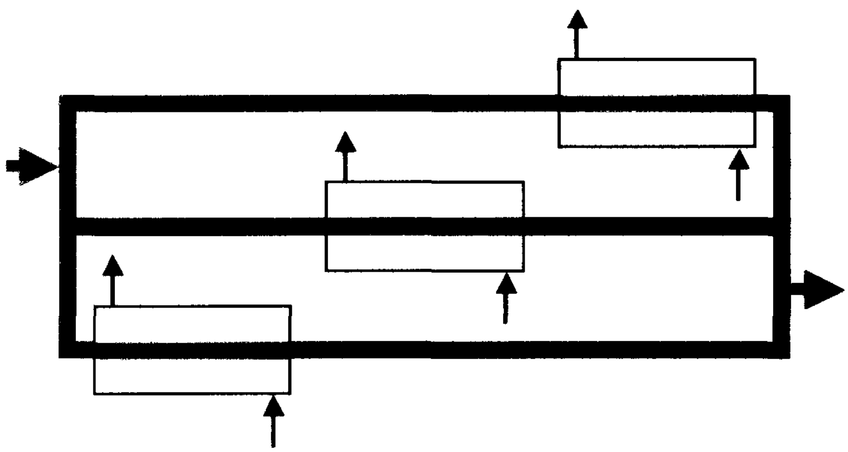
Evaporation took place within three hollow fiber membrane modules (6), as shown in Figure 3.1. The modules were 2.5" x 8" X50 modules produced by Liqui-Cel of Charlotte, NC. The fibers were polypropylene membrane within polypropylene housing with ¼" pipe-threaded connections for both the shell and lumen sides. When operating at 70°C, the manufacturer reports a liquid entry pressure of 2.0 bar. Other properties of the membrane modules are listed in Table 3.1. Because it was expected that higher air flow would be beneficial, the modules were arranged to have air flowing through the shell side and water on the tube side, allowing air flow rates of up to 45 L/min per module.

Table 3.1: Membrane properties

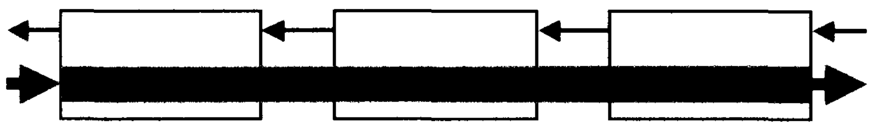
Module volume	0.64 L
Membrane area (per module)	1.4 m^2
Fiber length	202 mm
Fiber ID	0.2 mm
Fiber material	polypropylene
Pore size	0.05 μm
Maximum gas flow (tube side)	25 L/min
Maximum gas flow (shell side)	45 L/min

Cooling water was in a circuit between the chiller condenser and membrane modules, connected with ¼" Tygon tubing throughout. Beginning at the flask (7), which acted as a reservoir of water, the residing mass of water could be measured with a balance (Scout Pro from Cole Parmer, Montreal, QC). Water is sent out with a peristaltic pump (Cole Parmer, Montreal, QC) to the chiller condenser, where it is heated. The hot water proceeds to the three membrane modules, which had the possibilities of being arranged in a parallel cross-current, counter-current, or cross-current configuration with respect to air, as shown in Figure 3.2. While the counter-current arrangement would have been the most compact arrangement and allowed water from one module to be cooled further in later modules, the cross-current arrangement was chosen because the modules were restricted to 45 L/min of air each. The cross-current arrangement allows three times as much air as the series arrangement while only needing to split the air stream.

a.



b.



c.

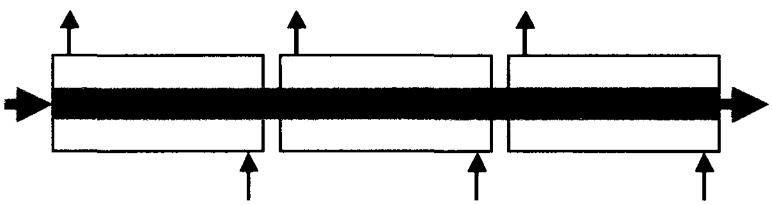


Figure 3.2: Possible arrangements of air and water within three modules. a) parallel cross-current; b) counter-current; c) cross-current

Water cooled within the modules by a combination of evaporation and conduction across the membrane. The cooled water then returned to the flask. Since some mass is lost during each circuit through evaporation, the balance at the flask was used to measure changes from evaporative losses rather than the absolute mass of water. Thermocouples (Omega, Laval, QC) were placed at the entrances and exits of modules and around the chiller to record temperatures. The air supplied by the campus power plant entered the modules at 26°C and 15% humidity, leaving the modules saturated above 95% humidity at varying temperatures depending on the conditions of a particular run.

Experimental procedures

Experiments consisted of over 50 steady-state cooling runs. There were several prior attempts that did not reach steady state as part of the development of the apparatus, but steady-state operation of the chiller could be consistently expected once the condenser, heaters, and modules were arranged in the previously-described apparatus.

A typical run consists of running the chiller for approximately two hours, usually until the amount of water remaining in the reservoir becomes low. To begin, the flask is filled with distilled water collected at room temperature, and water is pumped through the condenser and module circuit to fill any void spaces in the module that may have developed between runs. Similarly, water is circulated between the bath with heaters and the evaporator of the chiller. As simultaneously as possible, air is introduced to the modules, the heaters are turned on, and the chiller is started up, and the run has started. Entry and exit temperatures, the mass of the water in the reservoir, the current drawn from the compressor, and the power of the heaters are recorded every 5 minutes. Initially, the entire system is near room temperature and the chiller does not need to run non-stop to provide cooling. However, as heat from the heaters and compressor raise the overall temperature, the chiller runs constantly to provide cooling down to 16°C. Steady state is usually reached within 30 minutes, with the remaining time used to verify the steady state temperature and develop a profile for evaporative losses. To shut down a run, the heaters, chiller, air, and water pumps are turned off in the reverse order of the start-up procedure.

Several parameters could be adjusted to develop data over many runs. The air supplied to each module was set to 25, 30, 35, and 45 L/min to investigate the effect of air flow on different properties. Air flow rates were chosen that fit on the scale of a flow meter that had a maximum flow rate near that of the module itself. The water circulating between the condenser and modules was also set at 7, 9, and 12 g/s (corresponding to 420, 540, and 720 mL/min) for investigating water circulation effects. 12 g/s represented the maximum flow rate from the peristaltic pump; attempts at using a larger pump added extra heat to the system. It should be noted that, at the highest circulation rate, the Reynolds number was 2400 in the condenser and 18 in a fiber in the module, so the flow was in the laminar region. The heat load supplied by the immersion heaters could also be adjusted by changing the number of heaters. In practice, since the power output fluctuated somewhat, the heat load could be different by as much as 50 W between runs using the same number of heaters. The inconsistency in heat load led to the large number of runs, though the variation in heat load gave greater flexibility for plotting the effects of heat load on different properties. The cooling properties that were determined for each run were the coefficient of performance of the chiller, the ratio of cooling performance to electrical work demand; the quantity of cooling, measured by the amount of evaporation; and the quality of cooling, measured by the outlet temperature of the last cooling membrane module. These three properties indicated the performance, quantity, and quality of the SGME cooling apparatus.

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Chapter 4: Enhanced performance of the vapour compression cycle using sweeping gas membrane evaporative cooling

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Abstract

The vapour compression cycle is a widespread method of refrigeration. In the cycle, liquid refrigerant evaporates in an evaporator, absorbing heat from a low-temperature heat source. The resulting refrigerant vapour is compressed before condensing at a relatively high temperature. The performance of a chiller can be improved by lowering the condensation temperature, which reduces the work of the compressor. It is proposed that a sweeping gas membrane evaporation system can replace conventional air-cooled condensers to provide steady-state chiller operation. The SGME system allows the chiller to cool at steady state, and reduces the condensation temperature to improve the COP to as high as 1.83. It was determined that increasing the air flow and the condenser's water circulation rate increased COP, while there was an optimal heat load for the chiller of 435 W. The SGME-assisted chiller performance was superior to a chiller using an air-cooled condenser.

Introduction

Cooling is needed in a wide variety of applications. Air conditioning, food and chemical storage, and personal cooling equipment all require temperatures to drop below room temperature. Water is a common fluid medium in cooling applications because of its abundance and small impact on the environment, compared to other mediums like methanol, lithium salt solutions, or CFCs.

As a liquid at atmospheric pressure, water can theoretically provide cooling in the range of 0 – 100°C through melting, sensible heat transfer, or evaporation. Melting, which takes advantage of 330 kJ/kg of heat of fusion, provides high energy transfer at a temperature of 0°C, but the production of ice to supply the cooling process also requires a large amount of heat removal at a low temperature heat sink. Liquid water has one of the highest specific heat capacities, but transferring heat to a water reservoir or stream is not as effective per unit mass as a phase change – one kilogram of water would need to increase in temperature by nearly 80°C to achieve the same heat removal as melting one kilogram of ice. In addition, the increase in temperature of the water would reduce the temperature-based driving force for heat transfer. Evaporation, like melting, exploits a high heat of phase change – over 2000 kJ/kg – and is usually the basis for processes that cool to temperatures slightly below room temperature, such as refrigeration or cooling towers.

There are advantages and drawbacks to using evaporation, particularly that of water, in cooling applications. Evaporation takes in a much larger amount of heat per unit mass than other physical processes, and vapour forms from the equilibrium with liquid water at all temperatures between the triple point (0.01°C) and boiling point (100°C at atmospheric pressure). This wide range of temperatures satisfies the requirements for several cooling processes. However, the vapour produced from cooling has a high specific volume compared to the original liquid, so the vapour must be processed – such as by compression in the refrigeration cycle, or it must be released to the environment – as done in cooling towers. In addition, as shown by the Clausius-Clapeyron equation in Equation (1), at low temperatures, vapour pressure is very low, and pressure increases exponentially with temperature (Smith and Van Ness, 2005). As such, evaporation is only effective at relatively high temperatures.

$$\ln P = -\frac{\Delta H}{RT} + C \qquad (1)$$

Heat can be transferred from a low-temperature source to a high-temperature sink using a heat pump. An example of a heat pump is a vapour compression cycle refrigerator, which uses a refrigerant and electrical power to pump heat from a cold source to a relatively hot heat sink. The vapour compression cycle is shown in Figure 4.1. In the cycle, hot liquid refrigerant passes through the throttle valve (1), decreasing its pressure and temperature before entering the evaporator (2). In the evaporator, the refrigerant absorbs heat from an outside source that is at a relatively higher temperature. The heat source can be cooled to below room temperature when a refrigerator is in place. In the evaporator, the refrigerant absorbs the heat and evaporates, before being compressed to a high-pressure high-temperature vapour in the compressor (3). The vapour then enters the condenser (4) at high temperature, where it rejects heat to a relatively cooler sink by condensing. The condensed refrigerant then returns to the throttling valve to complete the cycle. It has been found that using liquid water or the evaporation of water to remove heat from the condenser significantly improves the performance of a refrigeration cycle compared to using air (Hosoz and Kilicarslan, 2004).

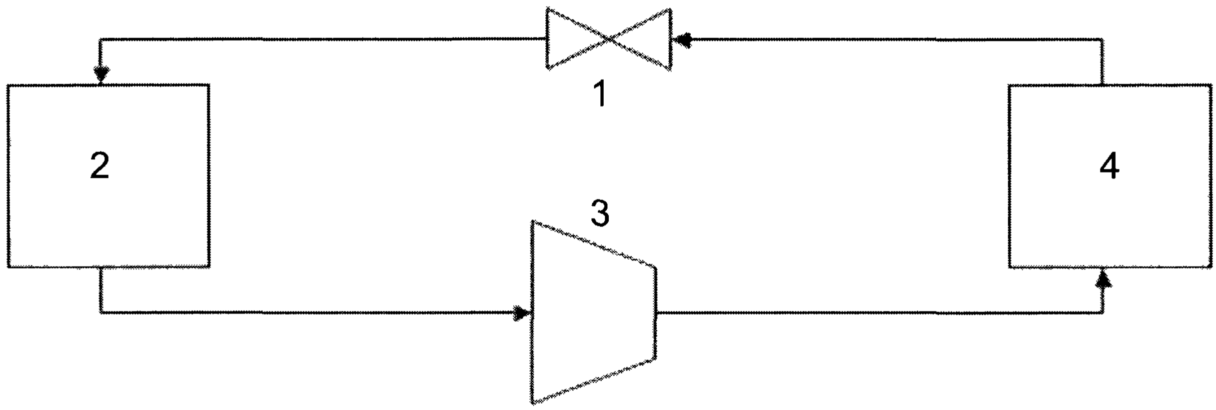

Figure 4.1: Vapour compression refrigeration cycle. 1: throttle valve; 2: evaporator; 3: compressor; 4: condenser

One method for evaporating water is based on the technology of membrane distillation. Membrane distillation is a separation technique that uses vapour pressure difference as a driving force to separate a volatile component out of a liquid stream. What makes membrane distillation different from other membrane separations is the small pore size and hydrophobic nature of the membrane; liquid water cannot pass through the membrane, but water vapour can. This forces liquids to evaporate before separating, vapour pressure difference is the only significant driving force for mass transfer across the membrane. The relationship between evaporative flux N and vapour pressure difference is shown as:

$$N = B\Delta P_{vap} \tag{2}$$

where B is a constant based on membrane properties (Khayet et al, 2004). The vapour pressure difference arises by raising the temperature of the feed and using various methods for keeping the vapour pressure on the permeate side low: vacuum (Cabassud and Wirth, 2003), cold streams (Sudoh et al, 1997), concentrated solutions (Celere and Gostoli, 2002), or dry sweeping gas (Khayet et al, 2004), for example.

If a feed of hot, pure water is used, the system would be membrane evaporation rather than distillation – or it becomes a method of separating liquid and vapour. Hot, pure water should evaporate easily, as long as a sufficient driving force is maintained, such as by using cool, unsaturated sweep gas to take in the humidity in the permeate. As the hot water evaporates, the vapour is separated from the liquid without needing to form a vapor-liquid equilibrium within the feed side. Thus, the latent heat removed by the evaporating water cools down the remaining water in the stream. For a heat of evaporation of 2400 kJ/kg and a specific heat capacity of 4.2 kJ/kg-K, a stream of 1 kg/s of water should cool by 5.7 K for every 1% of mass removed by evaporation, which is a considerable temperature drop for a small amount of lost water.

It is proposed that membrane evaporation is a viable method of removing heat from a refrigerator condenser. With the refrigerator acting as a heat pump, it is possible to achieve cooling below room temperature by harnessing the large heat of evaporation of water.

Materials and methods

In this experiment, a chiller was coupled with a membrane evaporation system to provide steady-state cooling. The full circuit of the streams involved in the apparatus are shown in Figure 4.2.

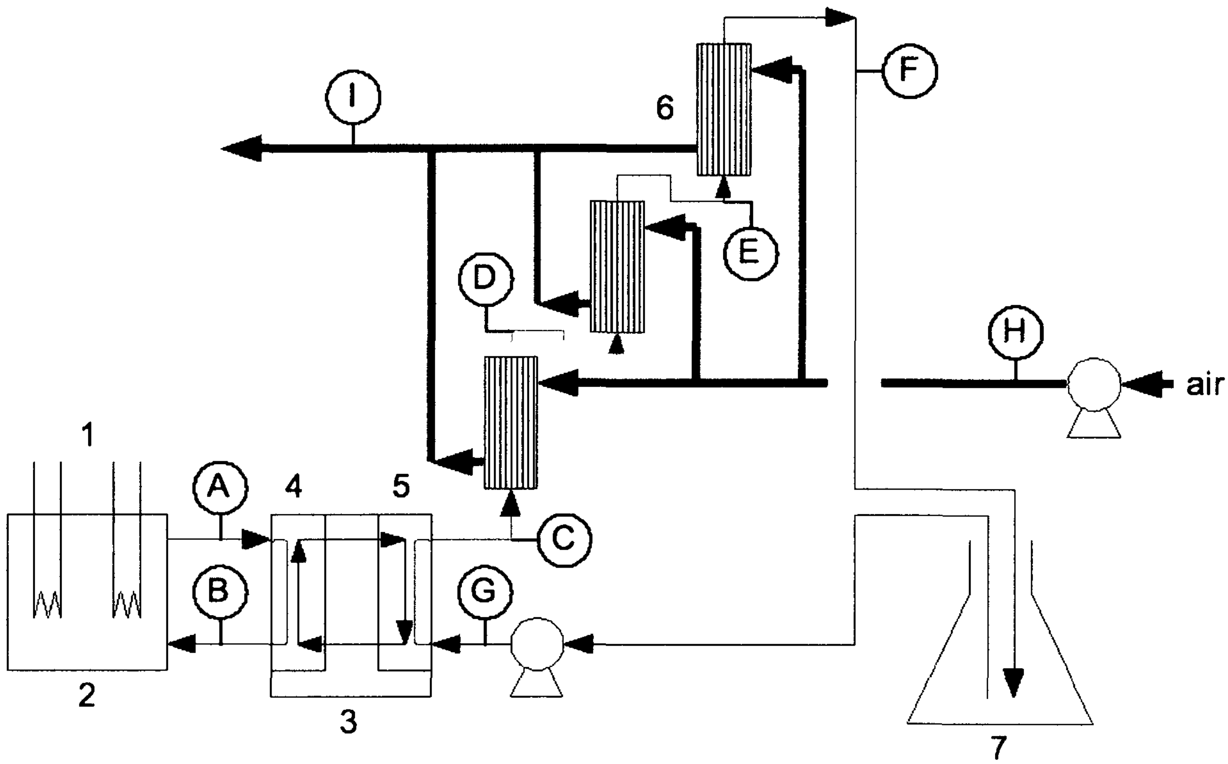

Figure 4.2: Diagram of chiller coupled with membrane evaporation system. 1: immersion heaters; 2: water bath; 3: chiller; 4: evaporator; 5: condenser; 6: hollow fiber membrane modules; 7: water reservoir; A to I: thermocouples

On the cold side of the system, immersion heaters (1; Elite Submersible by Hagen, Montreal, QC) provide the heat load to the system. The heaters were placed in a circulating water bath (2; Cole Parmer, Montreal, QC), which circulated water between the bath and the evaporator side (4) of the chiller (3). Since the immersion heaters had some fluctuation in power output depending on bath temperature, an energy meter (EM100 at Canadian Tire, Ottawa, ON) was used to record the total power consumption of the heaters. The measured power was then taken as the heat load entering the system, for the purposes of determining the coefficient of performance. Temperatures of the circulating water were also recorded.

The chiller was designed to chill a stream of water and was modified to reject heat at the condenser (5) to another stream of water. Its original purpose was to provide personal cooling in hot environments, and its original design used fan-blown air for cooling. Trials suggested that the chiller would have a coefficient of performance of up to 1.0 in hot environments with an air-cooled arrangement. The condenser was modified to a water-cooled version for our experiments.

A second stream of distilled water circulates between the condenser side of the chiller and the membrane evaporation system. At the condenser, water is heated, and the temperature is measured. Hot water proceeds to a set of three modules (6), which were 2.5" X 8" polypropylene hollow fiber modules (Liqui-cel, Charlotte, NC). The membranes had a total surface area of 1.4 m^2 per module, with a pore size of 0.05 microns. Further membrane properties are listed in Table 4.1. Water could pass through either the shell or tube side of the modules, with dry air (measured at 26°C, 15% humidity) passing on the opposite side counter-currently. Between the three modules, there were three plausible arrangements for the flow of air and water: in parallel cross-current, where the streams of air and water are split between the modules; in counter-current, where a single stream passes sequentially through each module; and in a cross-current arrangement, where a single stream of water passes through all three modules, but the air is divided equally so that a smaller flow enters each module in parallel. The different arrangements were tested to determine if there were any discernable differences, and to judge the overall performance. After the hot water cooled in the membrane, the water proceeded to a flask (7) which acted as a reservoir for measuring the mass of remaining water before the water recirculated via a peristaltic pump (Cole Parmer, Montreal, QC) back to the chiller for reheating. The saturated air coming out of the modules was bubbled out through a second flask to remove some of the moisture before being released to the environment. 1/4" flexible Tygon tubing was used throughout the membrane evaporation system for connections, with several thermocouples (Omega, Laval, QC) placed along the circuit for measuring temperatures.

Table 4.1: Membrane properties

Module volume (l)	0.64
Membrane area (m^2)	1.4
Fiber length (mm)	202
Fiber ID (mm)	0.2
Fiber material	polypropylene
Pore size (μm)	0.05
Maximum gas flow (tube side) (L/min)	25
Maximum gas flow (shell side) (L/min)	45

Initially, runs were conducted to determine optimal membrane arrangement and to achieve steady-state cooling. As described earlier, the streams could pass on the shell or tube side within a module, and the modules themselves could be positioned in different configurations. For all membrane arrangements, the modules were oriented vertically such that the water stream enters at the bottom and exits at the top. To test if steady state cooling was possible, the system ran for 90 to 150 minutes, usually until measured temperatures in the circuit became stable.

A typical run starts with ambient conditions and proceeds to steady state. With the desired module arrangement in place, water is circulated through the modules and chiller condenser at flows of 7-12 g/s (420-720 mL/min). The water circulates for a few minutes to fill up empty space in the system. Shortly thereafter, air is added at rates of 25-45 L/min per module. While there will be some evaporative cooling between the water and air at ambient temperatures, it does not appreciably decrease the total amount of water in the system. On the other side of the chiller, heaters are added to the bath pumping the chilled water. Without the chiller running, the heaters will raise the temperature of the chilling water, so the chiller is started up, marking the beginning of the run. At nearly the same time, the energy meter starts recording the power consumption of the heaters. On five minute intervals, the power drawn from the heaters and the chiller is recorded, as is the mass of water in the reservoir and several temperatures: that of the chilled water, the heated water from the condenser, cooled water leaving the modules, water returning to the condenser, and the saturated air leaving the modules. Since the system is initially at room temperature, it takes approximately 20 minutes before temperatures begin to stabilize (the "cooled" water leaving the modules is usually 40-50°C); the run is completed when the temperatures and power loads have been stable for at least six consecutive intervals (30 minutes). Upon completing a run, the chiller and heaters are shut down first, and the water and air are left to circulate for a few minutes longer to help cool down the system.

As will be seen in the results, placing the modules in a cross-current arrangement with air on the shell side was the best choice of configuration. Using this arrangement, the heat loads, circulation rates, and air flow rates were adjusted to determine optimal conditions for low temperatures, evaporation (rate of mass loss of water), and coefficient of performance. For this system, COP was calculated as:

$$\text{COP} = \frac{\text{heater load}}{\text{electrical demand of compressor}} \tag{3}$$

Results

Steady-state cooling from a chiller coupled with a membrane evaporation system was achieved. Figure 4.3 shows an individual typical run with its temperature profiles in the system. The example run was arranged in cross-current with air on the shell side of the modules, with a heat load of 395 W, water circulation of 12 g/s, and air at 45 L/min per module. As shown in Figure 4.3, the temperatures stabilized after approximately 30 minutes, with membrane evaporation cooling the water from 54 to 44°C.

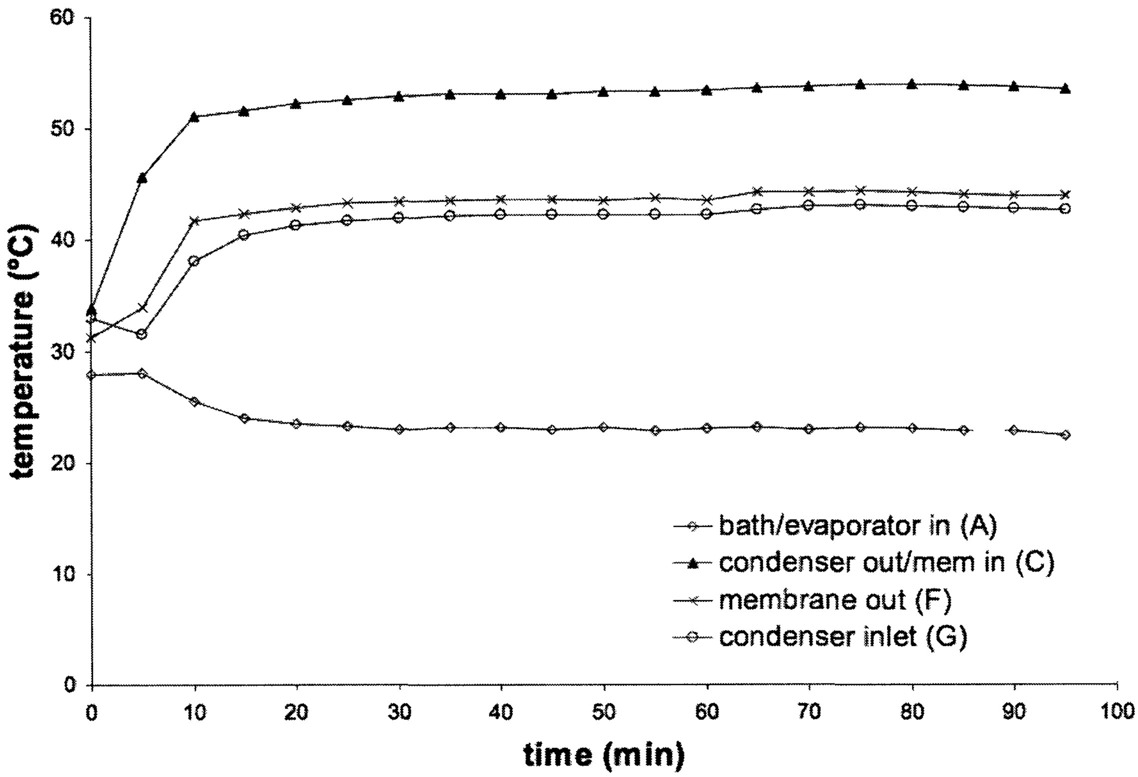

Figure 4.3: Temperature profiles of a steady-state cooling run

Figure 4.4 indicates the evaporative performance of the different module arrangements to determine whether to place three modules with the air and water in parallel cross-current, counter-current, or cross-current. The total amount of air is identical between the tests, which resulted in each of the three air streams in parallel and cross-current having one third of the flow rate as the single stream in the counter-current arrangement. Figure 4.4 shows the amount of evaporation observed at different temperatures over the course of a single run. The counter-current arrangement was best under these conditions, having high evaporation at lower temperatures. Nevertheless, the cross-current arrangement was chosen for future runs because its performance was comparable to counter-current at higher temperatures, and because more air in total can be used in the cross-current arrangement before reaching the module manufacturer's limits on maximum air flow (cross-current makes use of three streams of air, which can all be at the same rate as one stream in counter-current, if necessary).

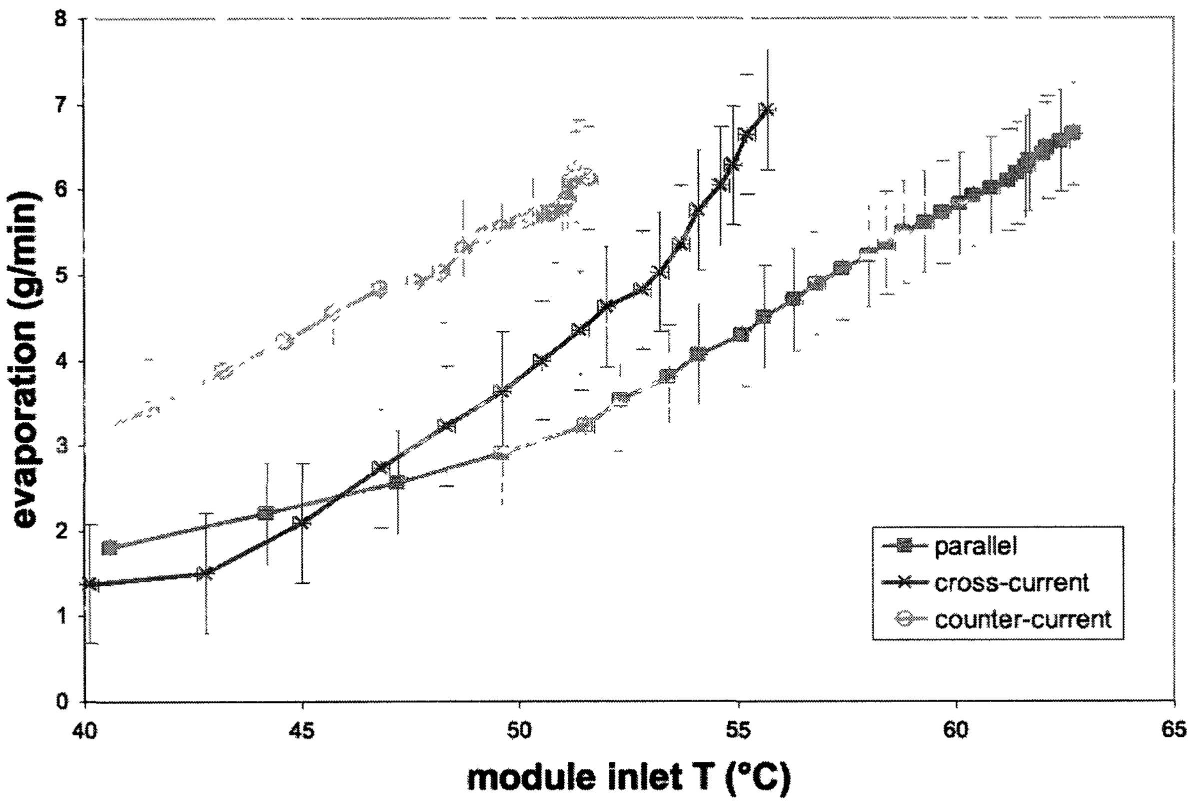

Figure 4.4: Comparison of evaporative performance of different module arrangements.

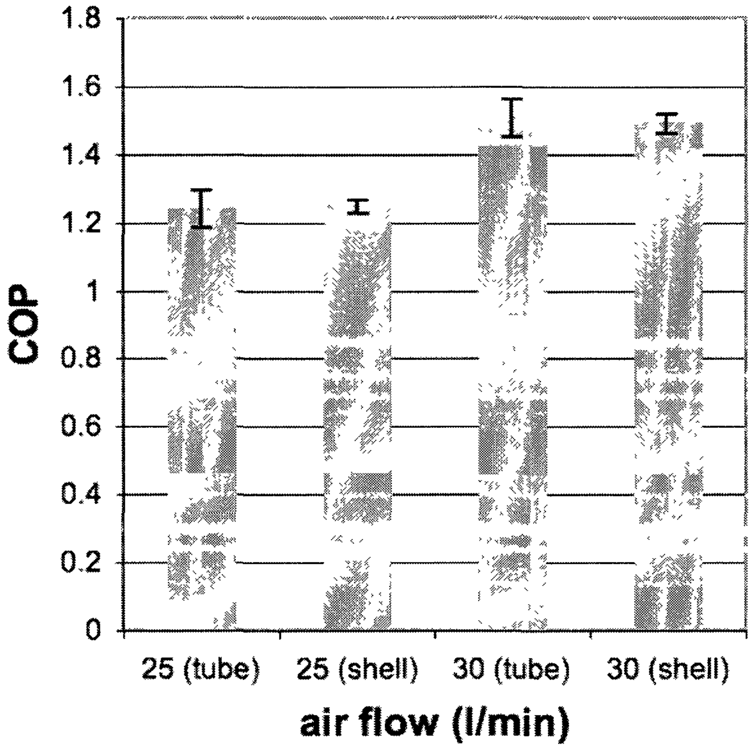

Figure 4.5: Effect of shell-and-tube configuration on coefficient of performance

Figure 4.5 compares the COP of the chiller when air passes through either the tube or shell side of a module. As seen for two different air flow rates – 25 and 30 L/min – there was no significant difference between having air on the tube or shell side. As such, air was passed through the shell side in future runs because having air on the shell side provides more volume in the module for air, allowing higher overall air flow rates – up to 45 L/min rather than the manufacturer's recommendation of 25 L/min for air in the tube side.

Once steady state was confirmed and the arrangement was chosen, runs proceeded with different heat loads, air flow, and water circulation rates. COP was used as an indication of the performance of the chiller.

Figure 4.6 shows the effect of heat load on the chiller COP for runs with water circulating at 12 g/s and air flow of 45 L/min per module. There was an observed maximum

COP of 1.8, which was the highest COP across all runs performed. At low heat loads, the COP increases with respect to heat load as shown in Equation (3). When the heat load is low, the compressor runs efficiently and only draws a low current from its power supply. At high heat loads, the compressor has a higher electrical demand to compress the refrigerant that has risen in temperature. As such, the denominator of Equation (3) dominates, resulting in a decrease of COP at higher heat loads.

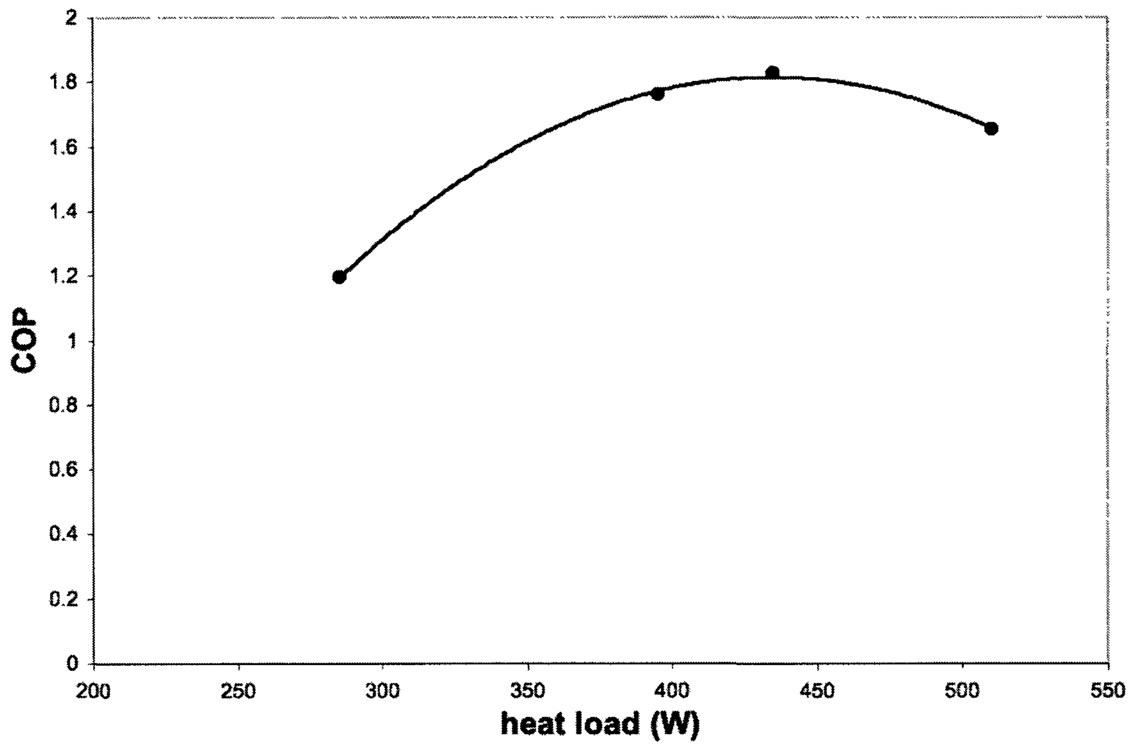

Figure 4.6: Effect of heat load on the coefficient of performance

Figure 4.7 shows the effect of air flow on the performance of the chiller. As the flow rate of unsaturated air increases, more water vapour can be carried away before the air becomes saturated. The increased capacity for water vapour allows more water to evaporate within the membrane modules, which decreases the temperature of the circulating water. When the circulating water is cooler as it passes through the condenser, it increases the temperature difference across the condenser heat exchanger, allowing greater heat transfer. The higher heat transfer experienced because of an increased air flow rate decreases the work of the compressor, which improves the COP of the chiller.

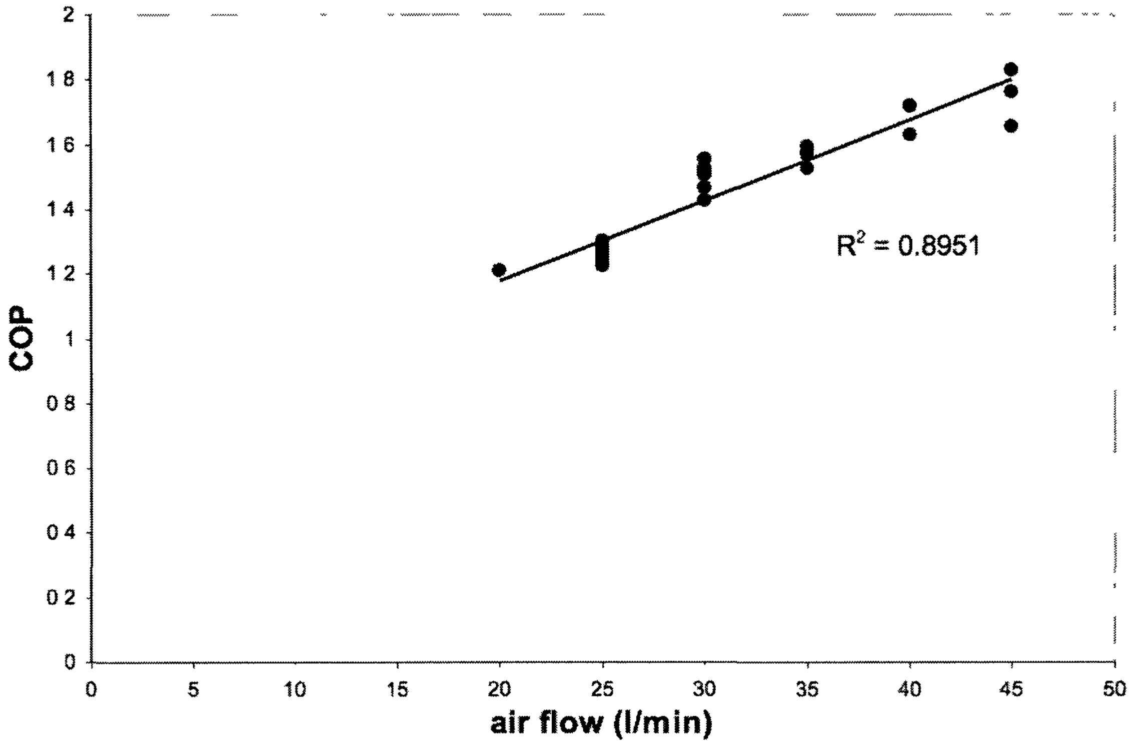

Figure 4.7: Effect of air flow rate on coefficient of performance

Figure 4.8 shows the effect of the water circulation rate on the coefficient of performance. As mentioned earlier, improving the heat transfer within the condenser increases the COP of the chiller. While heat transfer can be increased by lowering the temperature of the circulating water, increasing the circulation rate will improve heat transfer by increasing the heat transfer coefficient within the stream of water. Nevertheless, this improvement is not as statistically significant as that observed by increasing air flow because a higher circulation rate also reduces the residence time for cooling within the membrane modules, resulting in conflicting effects that produce an ambiguous correlation.

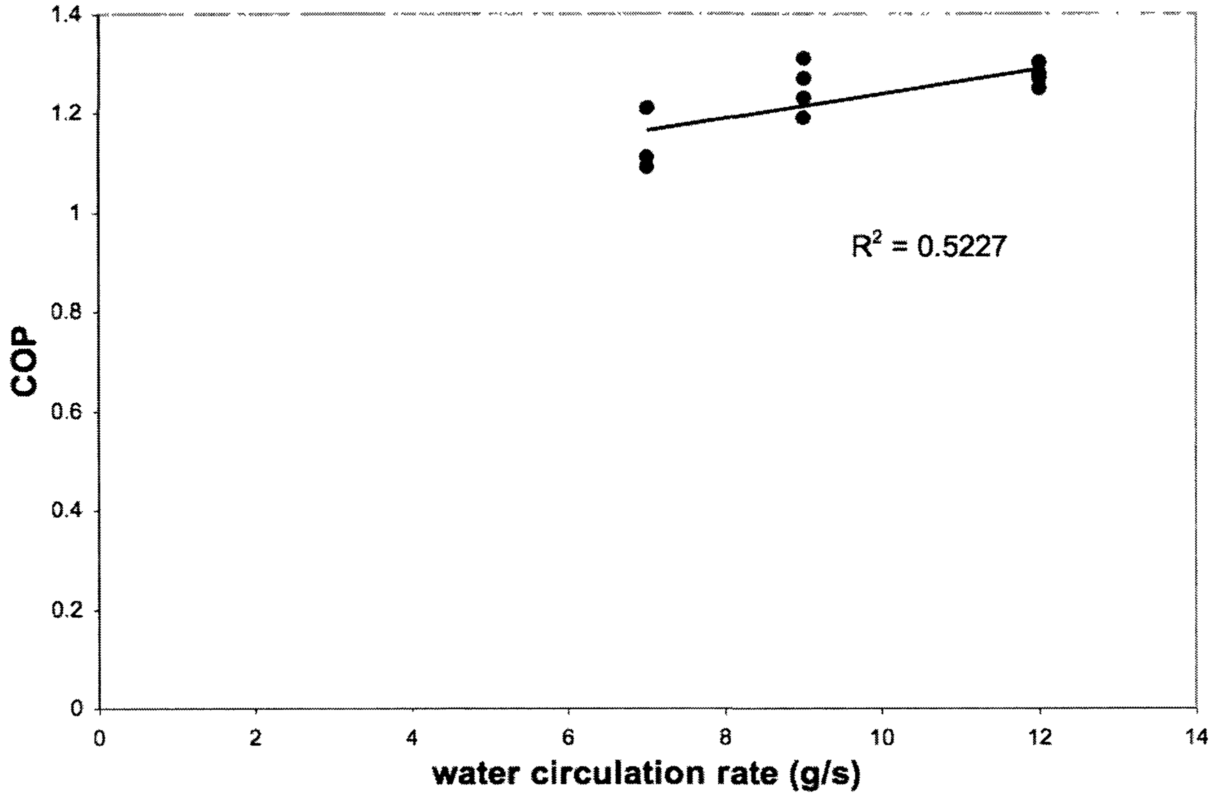

Figure 4.8: Effect of water circulation rate on coefficient of performance

Figures 4.9 and 4.10 indicate properties of the chiller itself. Figure 4.9 shows the relationship between the condensation temperature of the refrigerant in the chiller and the COP for similar heat loads (350-400 W). As the condensation temperature increases, the COP decreases, largely because the compressor needs to put in more work to achieve the pressures experienced at the higher temperatures. This is confirmed with Figure 4.10, plotting condensation temperature against the actual electrical demand. Since COP is inversely proportional to electrical demand, the observed increase in electrical demand is expected. As such, it appears that keeping the condensation temperature low (by improving the membrane cooling) is the major factor for improving the COP of the chiller. This agrees with the observations made with increasing air flow and water circulation rate.

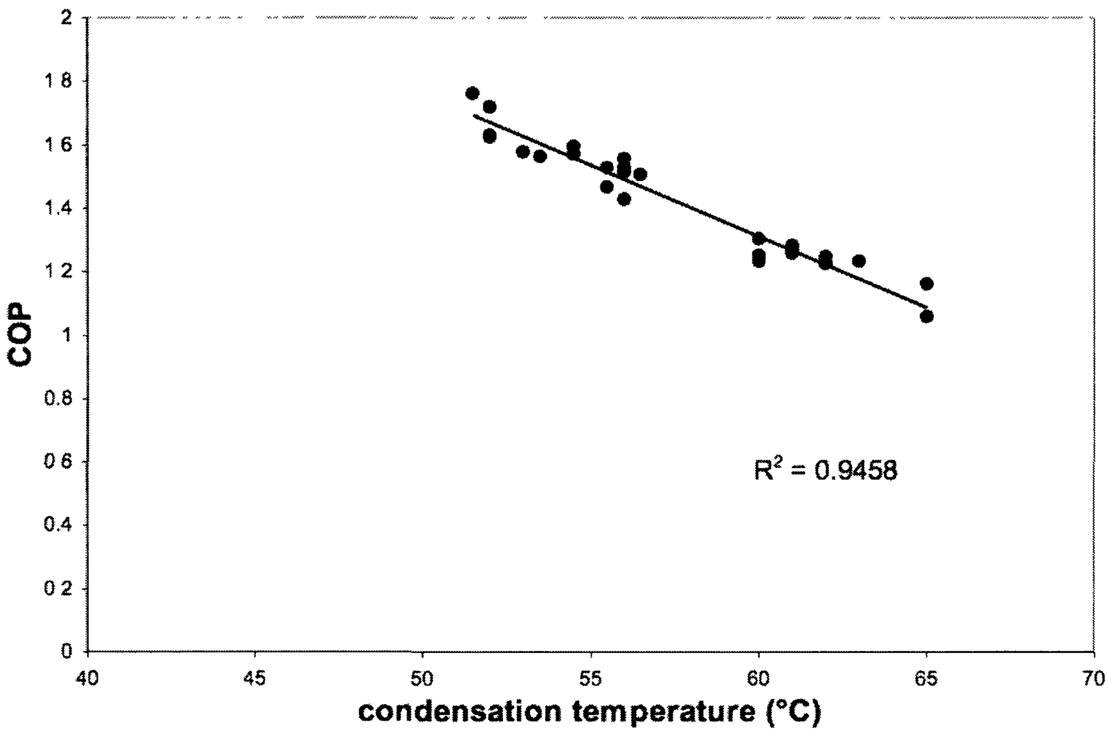

Figure 4.9: Effect of condensation temperature on chiller COP

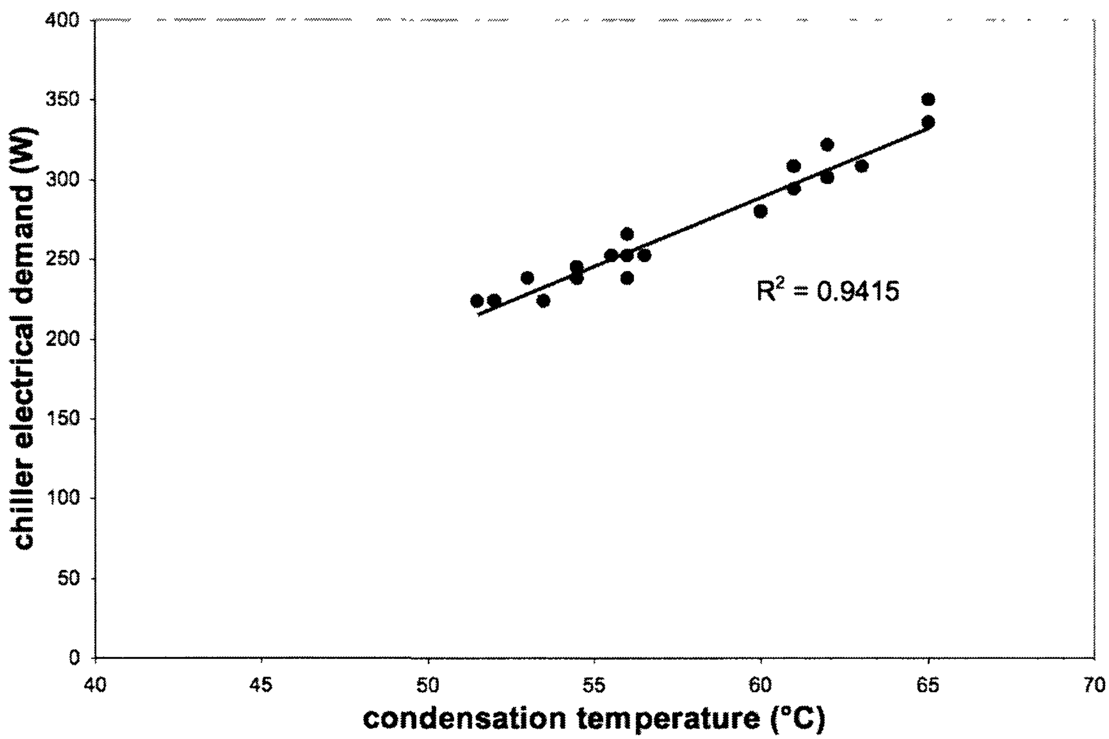

Figure 4.10: Effect of condensation temperature on chiller electrical demand.

To summarize the results, a heat balance was also performed, as shown in Figure 4.11. The sum of measured energy inputs (heat load and electrical demand) were plotted with the measured heat outputs (evaporative cooling and the heating of the air). A unity line, representing an ideal situation of measured heat inputs balancing the measured heat outputs, is also shown for comparison. The runs fit on a line that falls slightly below unity, with an intercept of -146.5. Having a slope of one indicates that heat input does not affect other properties (circulation, COP, and others) in a way that would affect the heat outputs, and the intercept suggests that 146 W is lost, regardless of other conditions, to the environment either at the chiller or in the piping and membrane housing.

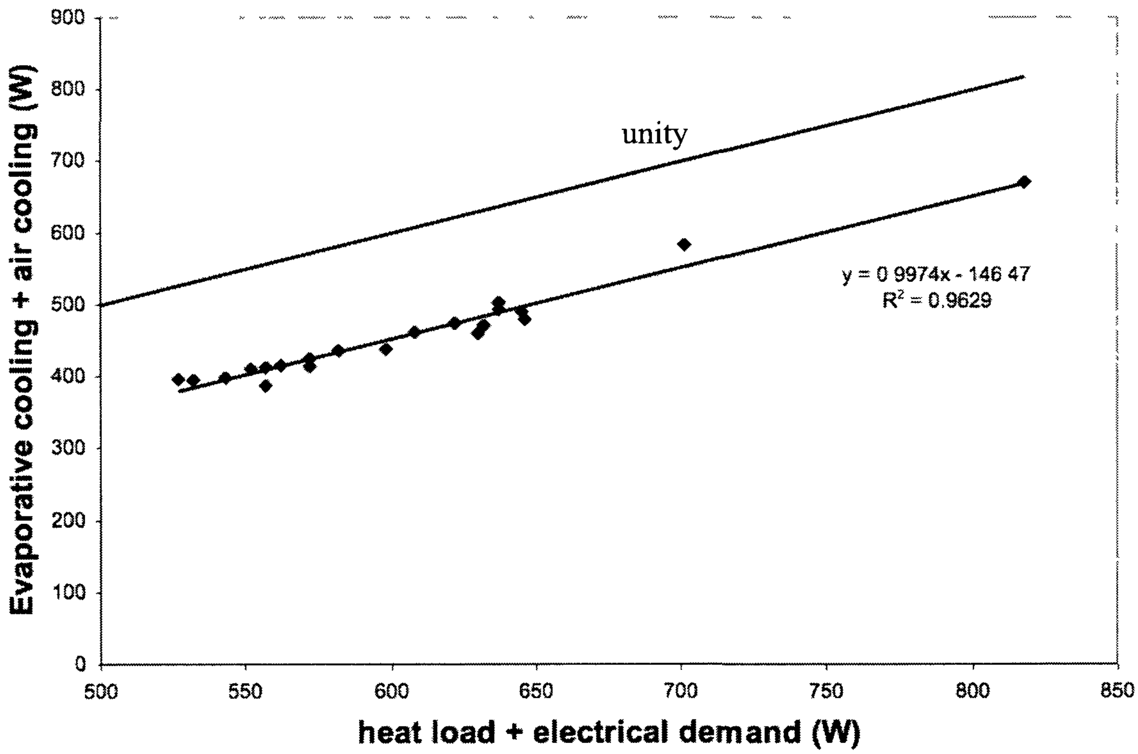

Figure 4.11: Heat balance of the chiller and membrane evaporation system

Finally, a summary of the conditions of the runs that provided the highest COP is listed in Table 4.2. As suggested by Figure 4.6, there is an optimal heat load to produce the highest COP. For the studied chiller, that maximum occurred with a heat load of 435 W. The air flow rate and water circulation rate were high in this run, which agrees with the observations in Figures 4.7 and 4.8, respectively. The relatively low membrane outlet temperature also allowed the refrigerant to condense at a lower temperature, which improved COP according to Figure 4.9.

Table 4.2: Conditions of highest COP run

	highest COP run
Heat load (W)	435
Air flow (L/min)	135
Circulation rate (g/s)	12
Membrane inlet/condenser out (°C)	55.0
Evaporative cooling (W)	455
Membrane outlet (°C)	43.7
COP	**1.83**

Conclusions

A membrane evaporation system was successfully coupled with a chiller to achieve steady-state cooling. Having three modules placed in a cross-current arrangement with air on the shell side was the most effective configuration. As listed in Table 4.2, the highest COP achieved was 1.83, which is much higher than the COP of the chiller when it used fan-blown air for cooling. The steady state, high COP, and compact nature of using water rather than air concludes that an evaporation-based system is better for removing heat from the condenser of a chiller.

While steady state was achieved in runs of varying air flow rates, it appears that having more air improves the performance of the system. A larger volume of air has a larger capacity for evaporated water before it becomes saturated, and the presented results were limited only by the manufacturer's specifications of the modules.

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Chapter 5: Sweeping gas membrane evaporative cooling: an effective replacement of cooling towers

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Abstract

Cooling towers are the conventional method of evaporative cooling. These towers require a large footprint and high air flow to be effective, making them practical for only large-scale cooling. A sweeping gas membrane evaporative system is proposed to produce the cooling effects of a tower at a small scale while using air more efficiently. A SGME system was developed that could cool water from 60°C to less than 40°C while removing more than 500 W of heat through evaporation. The quantity of evaporative cooling increased with increasing air flow, while the quality of cooling increased with decreasing water circulation rate. Normalized for volume, the SGME provided more heat removal than cooling towers, and required less air per unit of heat removal.

Introduction

There are numerous applications for cooling in industrial and personal use. Air conditioning, refrigeration, cooling garments, food storage, and chemical processing all make use of cooling technologies. Water is a common fluid used in cooling because it is ubiquitous, non-toxic, and has excellent physical properties for heat transfer.

The evaporation of water is an effective use of water for cooling when operating in conditions above the ambient temperature, such as with air conditioning or dissipating waste heat. Water has a large heat of evaporation of approximately 2400 kJ/kg, which allows a large amount of energy to be transferred to a small mass of cooling fluid. This is more effective than making use of the heat capacity of water for sensible heat transfer (Hosoz and Kilicarslan, 2004). However, from the Clausius-Clapeyron equation (Smith and Van Ness, 2005):

$$\ln P = -\frac{\Delta H}{RT} + C \tag{1}$$

it can be seen from the exponential relationship between vapour pressure and temperature that, at low temperatures, very little evaporation can occur before vapor-liquid equilibrium is reached. As such, evaporation becomes a more effective cooling method when operating at higher temperatures, such as above room temperature.

In industry, a widespread technology for cooling hot water to near room temperature is the cooling tower. In a cooling tower, hot water is sprayed downward to a collection pool where the water exits at a lower temperature. As the water descends, it is in contact with drier and cooler air, which causes the water to lose some heat through conduction and most of its heat through evaporation (Khan *et al.*, 2003). Air flows through the tower either by natural draft (Pearlmutter *et al.*, 2008) or blown air (Naphon, 2005), and the air flows either upwardly counter-current to the falling water droplets (Heidarinejad *et al.*, 2009) or perpendicular to the droplets (Hajidavaloo *et al.*, 2010), which is called "cross-flow" in this paper to distinguish from the SGME "cross-current" arrangement.

Various studies have determined the performance of cooling towers of different sizes, ranging from a bench-scale column to large towers used in industry. A sampling of parameters from recent studies into cooling towers is shown in Table 5.1.

Table 5.1: Parameters of cooling tower studies

	Khan, 2003	Naphon, 2005	Qureshi, 2006	Marques, 2009	Hajidavaloo, 2010
air flow	counter-current	counter-current	counter-current	counter-current	cross-flow
packing	void	plastic plates	void	void	wood blocks
volume (m^3)	0.7	0.011	203.2	1785	633.6
air flow (L/s)	989	60	94 000	285 000	536 000
water (kg/s)	1.78	0.04	94	722	953
water in (°C)	28.7	39	40	40	58
water out (°C)	25.2	23	18	31.3	30
cooling (kW)	26.2	2.7	8685.6	26381	112073
cooling (kW/m^3)	37.4	245.5	42.7	14.8	176.8

As seen in Table 5.1, cooling towers are effective at cooling large quantities of water to near room temperature. These temperatures, which are close to the environmental temperatures in warm climates where cooling is needed, are low enough for either rejecting waste water to the environment or returning process water to a system, two common uses of cooling towers. Cooling towers are advantageous in these conditions over other heat removal systems – such as heat exchangers – because temperatures are suitable for evaporating water and the droplets provide extensive contact surface area for heat transfer between the water and air phases.

There are drawbacks to cooling towers. In systems where large quantities of water must be cooled, large towers must be constructed. Since most of the interior volume of a tower is taken up by air, cooling towers require a large footprint compared to a system that handles water in a more compact space, such as a shell-and-tube heat exchanger. The large volume of air need for a tower requires powerful blowers, which consume energy and give off noise to the surrounding area. Finally, as will be shown later, scaling down towers to smaller sizes requires an excessive amount of air for the same performance, possibly because smaller columns do not have enough contact residence time between the water and air. Achieving the performance of a cooling tower with a small footprint would greatly extend the range of applications for cooling tower. These small-scale applications include personal and equipment microclimate cooling, microreactors, and the heat sinks for small-scale air conditioning and refrigeration

A technique for evaporating and separating water in a compact space is membrane evaporation. In membrane evaporation, a hydrophobic membrane separates water from a permeate phase with low vapour pressure, and water vapour passes through the membrane from a driving force of vapour pressure difference. This is similar to membrane distillation, a separation technique used in desalination and solution concentration. Membrane evaporation and distillation differs from osmosis-based techniques in that liquid water never passes through the hydrophobic membrane; the hydrophobic material and small pore size force only vapour to pass, which in turn forces water to evaporate before diffusing. The vapour pressure difference can be achieved in several ways, all by adjusting what occurs on the permeate side. Low permeate vapour pressure can come from: vacuum (Cabassud and Wirth, 2003), cold streams (Sudoh et al, 1997), concentrated solutions (Celere and Gostoli, 2002), or dry sweeping gas (Khayet et al, 2004), for example.

Sweeping gas membrane evaporation (SGME) has similarities to a small-scale cooling tower. Both techniques force hot water to evaporate into a dry air phase, and both make use of a high surface area – cooling towers through fine droplets and SGME through hollow fiber membrane modules. The major difference is that in SGME, the large surface of water evaporation is provided by the membrane rather than water droplet. As a result, the need of large void spaces between water droplets is eliminated, making a compact cooling device with a small footprint possible. Furthermore, water and air phases do not come in contact with each other, but are instead separated by a hydrophobic membrane that only allows vapour to pass. Isolating the phases from each other is advantageous for cooling purposes, however – with a barrier in place, negligible heat will transfer between the air and water by means of conduction, forcing all of the cooling to come from evaporation. The isolation can also allow the water to be cooled below the entrance temperature of the air, rather than forming a thermal equilibrium. As such, the membrane barrier allows a larger quantity of cooling for a smaller amount of evaporation than a cooling tower. Cooling towers can experience evaporative losses of 0.5% of the inlet flow (Pearlmutter *et al.*, 2008) to 5% (Hajidavaloo *et al.*, 2010), which decreases the amount of remaining cooled water available for recirculation over time.

This study makes use of a sweeping gas membrane evaporative system for cooling, which is unique from most membrane distillation studies that investigate separation. The SGME system can then be compared to cooling towers on the basis of quality (cooled outlet temperature), quantity (amount of heat removal), and efficiency (the most heat removal with the least amount of blown air).

Materials and Methods

In this experiment, a sweeping gas membrane evaporation system was coupled with the condenser of a chiller to provide steady state cooling. The chiller acted as a heat pump to take in a room temperature heat source and reject high-temperature heat to the membrane arrangement, which acted as a heat sink for the chiller. A diagram of the apparatus is shown in Figure 5.1.

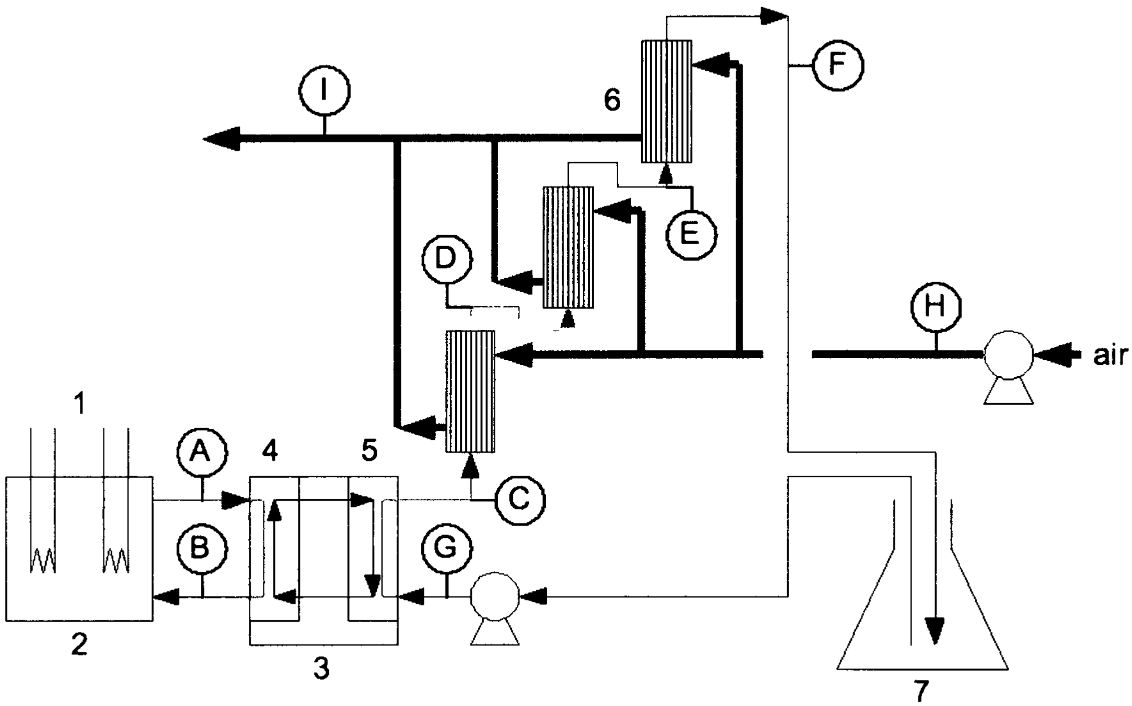

Figure 5.1: Sweeping gas membrane evaporation apparatus. 1: immersion heaters; 2: water bath; 3: chiller; 4: evaporator; 5: condenser; 6: hollow fiber membrane modules; 7: water reservoir; A to I: thermocouples.

To act as a heat source for the membrane cooling system, immersion heaters (1) warmed a circulating water bath (2; Cole Parmer, Montreal, QC). The water bath circulated water between the heaters and the evaporator side (4) of the chiller (3). Since the immersion heaters had some fluctuation in power depending on the temperature, an energy meter recorded the total power consumption of the heaters. The evaporator of the chiller returns cold water to the bath, where it is reheated by the heaters.

A second stream of distilled water circulates between the condenser side of the chiller (5) and the membrane evaporation system. At the condenser, water is heated, and the temperature is measured. Hot water proceeds to a set of three modules (6), which were 2.5" X 8" polypropylene hollow fiber modules (Liqui-cel, Charlotte, NC). The membranes had a total surface area of 1.4 m^2 per module, with a pore size of 0.05 microns. Further membrane properties are listed in Table 5.2. Water passed through the tube side of the modules, with dry air (measured at 26°C, 15% humidity) passing on the shell side countercurrently. Air passed through the shell side because the modules could handle a larger volume of air on that side. The three modules were oriented vertically and placed in a cross-flow arrangement, where water passed upward through the modules in series, while a fresh stream of air entered each module. Because of the limitations of the module capacity, this cross-current arrangement could handle three times as much air (up to 135 L/min total) as a counter-current arrangement where a single stream of air passed sequentially through the three modules (45 l/min total). After the hot water cooled in the membrane, the water proceeded to a flask (7) which acted as a reservoir for measuring the mass of remaining water before the water recirculated via a peristaltic pump (Cole Parmer, Montreal, QC) back to the chiller for reheating. The saturated air coming out of the modules was bubbled out through a second flask to remove some of the moisture before being released to the environment. 1/4" flexible vinyl tubing was used throughout the membrane evaporation system for connections, with several thermocouples placed along the circuit for measuring temperatures.

Table 5.2: Membrane properties

Module volume (L)	0.64
Membrane area (m^2)	1.4
Fiber length (mm)	202
Fiber ID (mm)	0.2
Fiber material	polypropylene
Pore size (μm)	0.05
Maximum gas flow (tube side) (L/min)	25

Maximum gas flow (shell side) (L/min)	45
Maximum operating temperature (°C)	70
Liquid entry pressure at 70°C (bar)	2.0
Liquid entry pressure at 40°C (bar)	4.8

Several runs were conducted to determine optimal conditions for steady-state cooling. A typical run starts with ambient conditions and proceeds to a warm steady state. Water is circulated through the modules and chiller condenser at flows of 7-12 g/s (420-720 mL/min), and proceeds for a few minutes to fill up empty space in the system. Shortly thereafter, air is added at rates of 25-45 L/min per module. While there will be some evaporative cooling between the water and air at ambient temperatures, it does not appreciably decrease the total amount of water in the system. On the other side of the chiller, heaters are added to the bath pumping the chilled water. Without the chiller running, the heaters will raise the temperature of the chilling water, so the chiller is started up, marking the beginning of the run. At nearly the same time, the energy meter starts recording the power consumption of the heaters. On five minute intervals, the power drawn from the heaters and the chiller is recorded, as is the mass of water in the reservoir and several temperatures: that of the chilled water, the heated water from the condenser, cooled water leaving the modules, water returning to the condenser, and the saturated air leaving the modules. Since the system is initially at room temperature, it takes approximately 20 minutes before temperatures begin to stabilize (the "cooled" water leaving the modules is usually 40-50°C); the run is completed when the temperatures and power loads have been stable for at least six consecutive intervals (30 minutes). Upon completing a run, the chiller and heaters are shut down first, and the water and air are left to circulated for a few minutes longer to help cool down the system. Runs continued with varying heat loads, circulation rates, and air flow rates to determine optimal conditions for cooling quality (membrane outlet temperature) and quantity (amount of evaporation). The data was then used to compare performance with cooling towers.

Results and Discussion

Steady state cooling was achieved with the membrane evaporation system. Figure 5.2 shows the power profiles of a typical run: a heat load of 395 W, water circulation of 12 g/s, and air at 45 L/min per module. After approximately 40 minutes of start-up from ambient conditions to steady-state temperatures, Figure 5.2 shows that the amount of evaporation stabilized at 410-440 W, removing 75-80% of the heat added at the condenser – the remaining heat was lost to the environment. The electrical demand of the chiller was constant during the steady-state period, implying that the chiller was also operating at steady state. Steady state was also achieved in several other runs under different conditions, allowing the demonstration of the effect of other parameters on the performance of the membrane evaporation system.

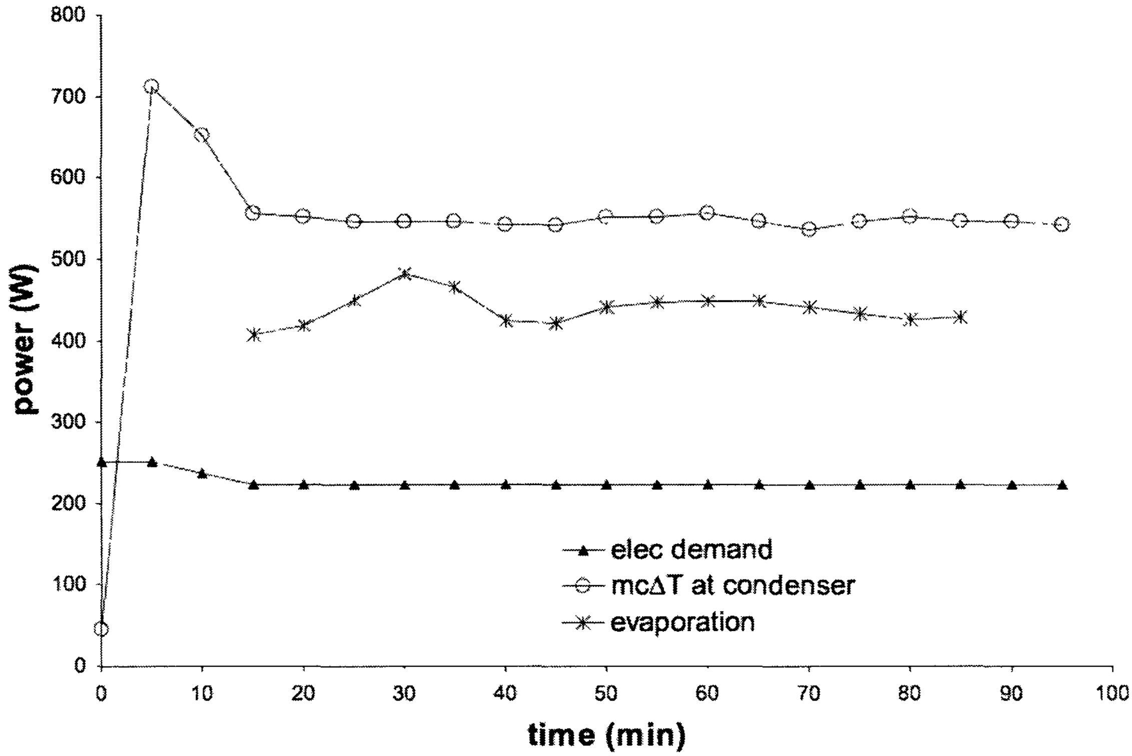

Figure 5.2: Power profiles of a steady-state cooling run

One measure of the performance of the membrane cooling system is the cooling quality, or the membrane water outlet temperature. Figure 5.3 shows the effect of heat load on the outlet temperature with constant circulation and air flow rates. An increase in heat load corresponds to an increase in the cooled temperature, which can be attributed to a higher condenser temperature that is dissipating more heat.

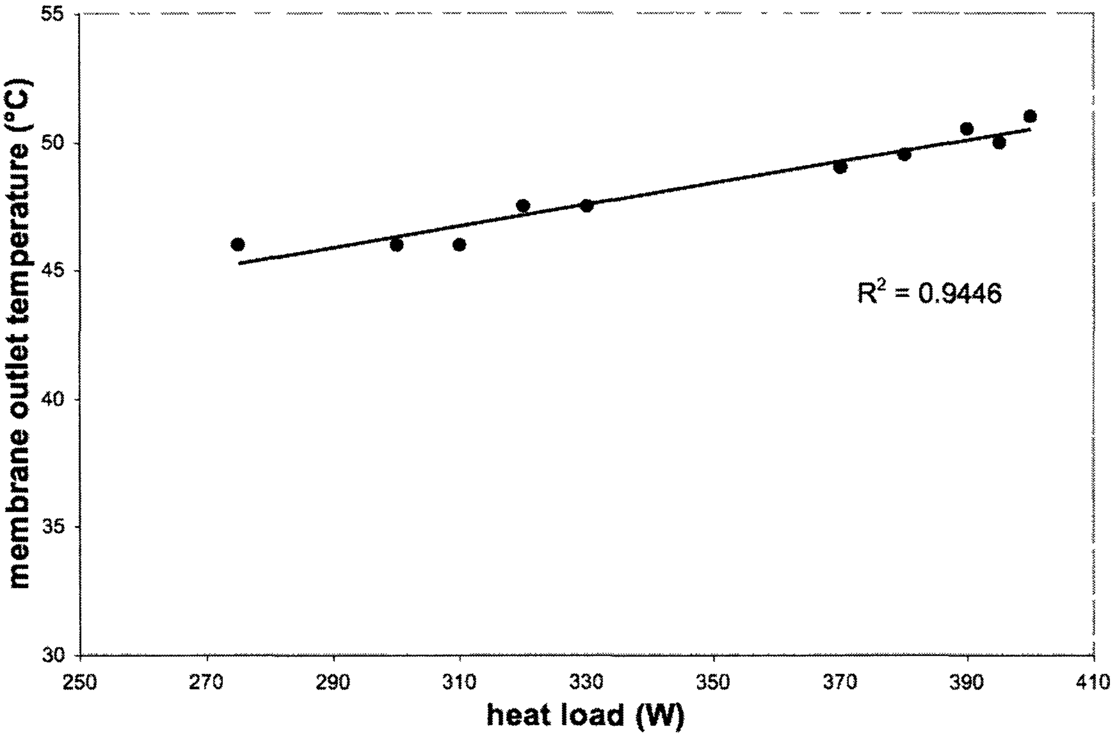

Figure 5.3: Effect of heat load on membrane water outlet temperature

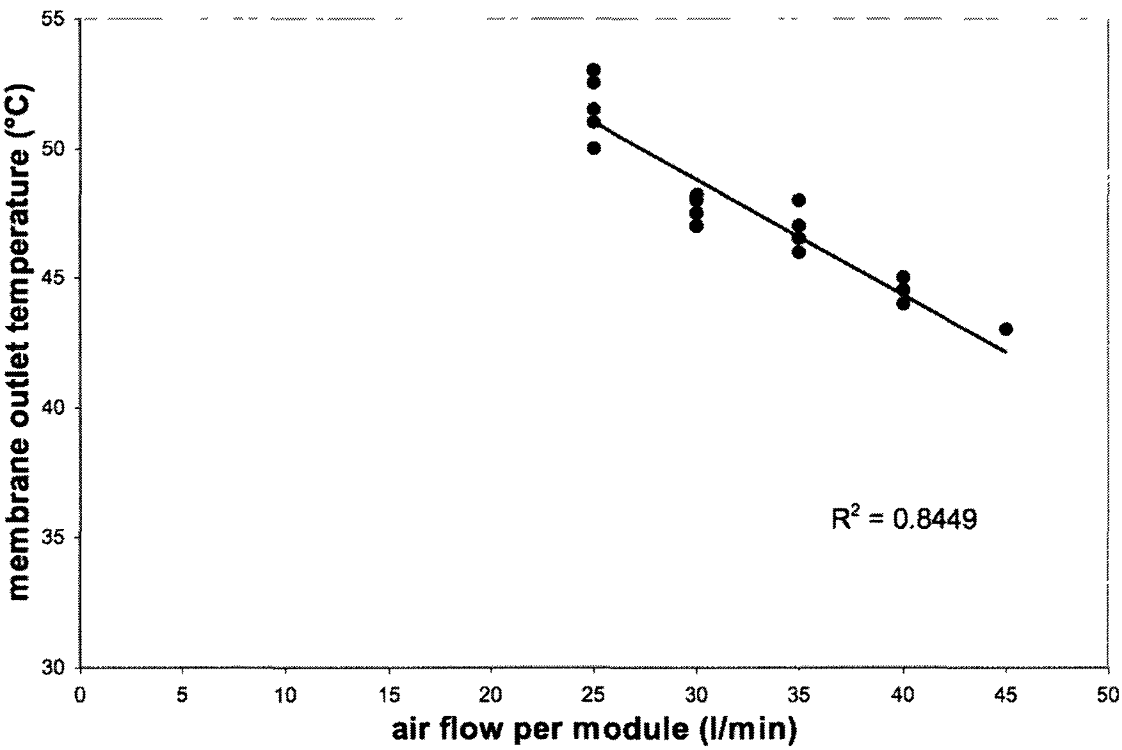

Figure 5.4: Effect of air flow on membrane water outlet temperature

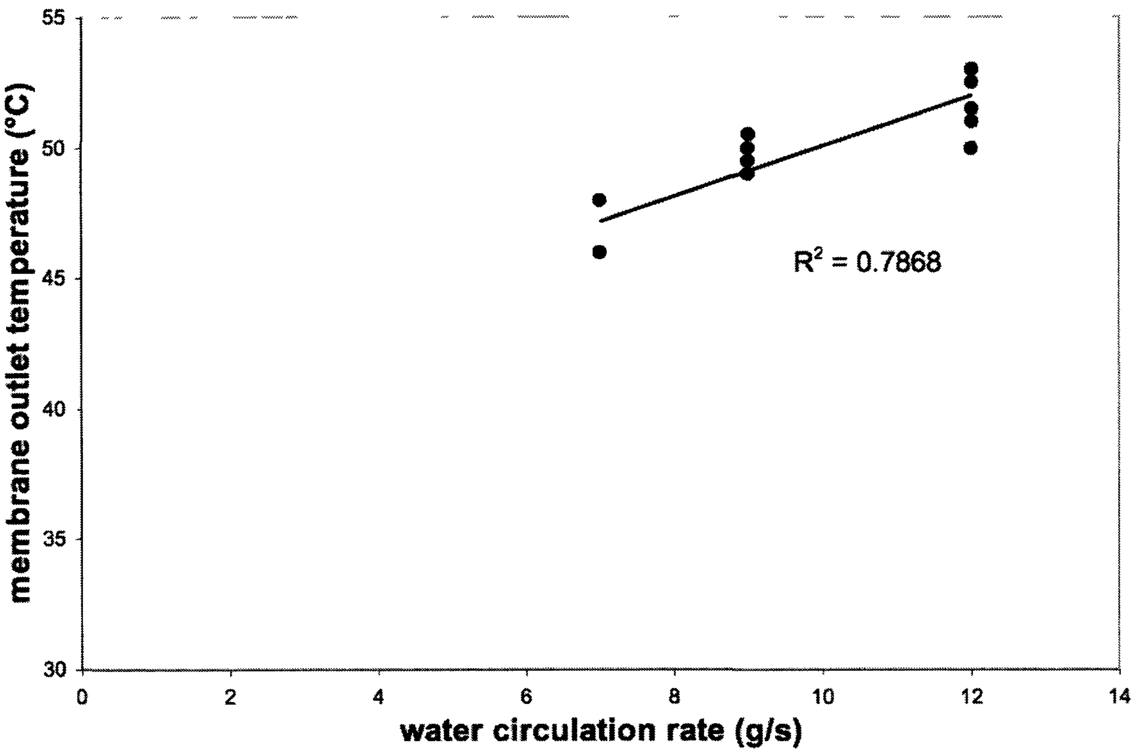

Figure 5.5: Effect of water circulation rate on membrane water outlet temperature

Figure 5.4 shows that an increase in air flow drives down the membrane water outlet temperature. With a larger volume of air, more water vapour can pass through the membrane before the air becomes saturated. This forces more evaporation to occur, which cools the circulating water. Also, since the air enters at a lower temperature than the cooled water, the increased air flow rate will remove more heat from the membrane surface by convection, although the insulative properties of the polypropylene membrane minimizes the heat loss by conduction across the membrane. Figure 5.5 shows that a decrease in water circulation rate will reduce the outlet temperature. This is because, as will be shown later in Figure 5.8, the evaporative heat removal is a weak function of water circulation rate in the tested range when air flow and heat load were constant. As such, for a constant heat load and air flow, the heat removal Q from the water stream is essentially constant. Based on heat capacity (Q =

$mC_P\Delta T$), a decrease in mass flow rate of water should increase the temperature drop within the module. Though not shown in earlier figures, the lowest outlet temperature was 39.5°C, achieved with low circulation rate (7 g/s) and high air flow (45 L/min per module).

Another indicator of the performance of the membrane evaporation system is cooling quantity. This was measured by the evaporative heat removal, which was calculated based on the steady-state slope of the decrease in mass in the reservoir. Evaporative heat removal does not include any other heat losses – such as conduction across the membrane or walls of the module – that may contribute to the cooling effect, and with no detectable leaks in the system, it was assumed that all lost mass was part of the evaporative cooling process.

Figure 5.6 shows the effect of membrane inlet temperature on the quantity of cooling. Membrane inlet temperature was used instead of heat load because a graph based on temperature should reveal an exponential relationship, based on equation (1). The exponential increase in the water saturation vapour pressure leads to a greater vapour pressure difference across the membrane. This increased vapour pressure difference increases the driving force for mass transfer by evaporation across the membrane. The data did produce an exponential fit, with greater evaporation at higher temperatures.

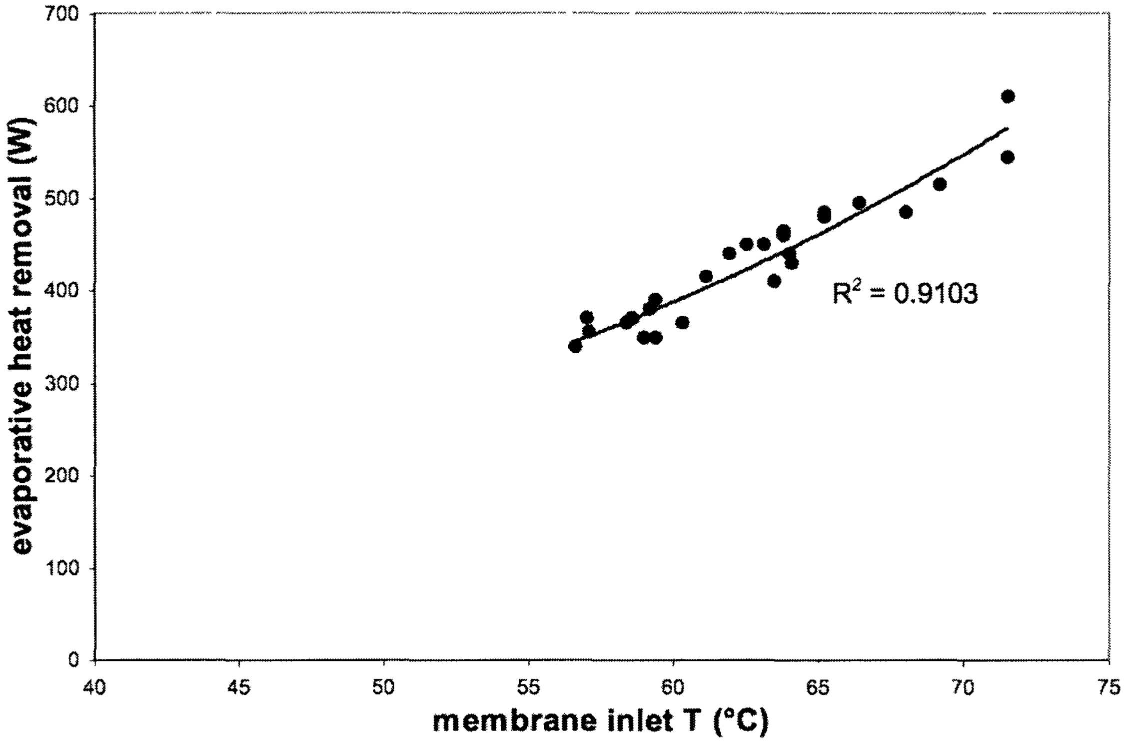

Figure 5.6: Effect of membrane inlet temperature on evaporative heat removal

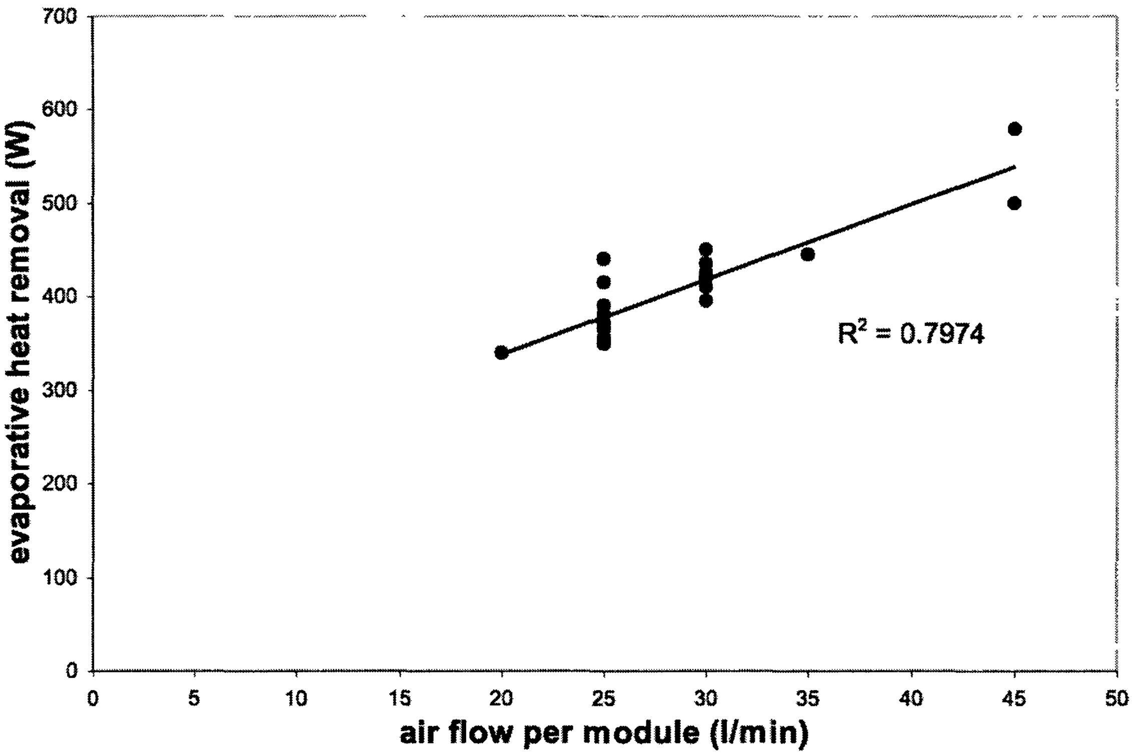

Figure 5.7: Effect of air flow on evaporative heat removal

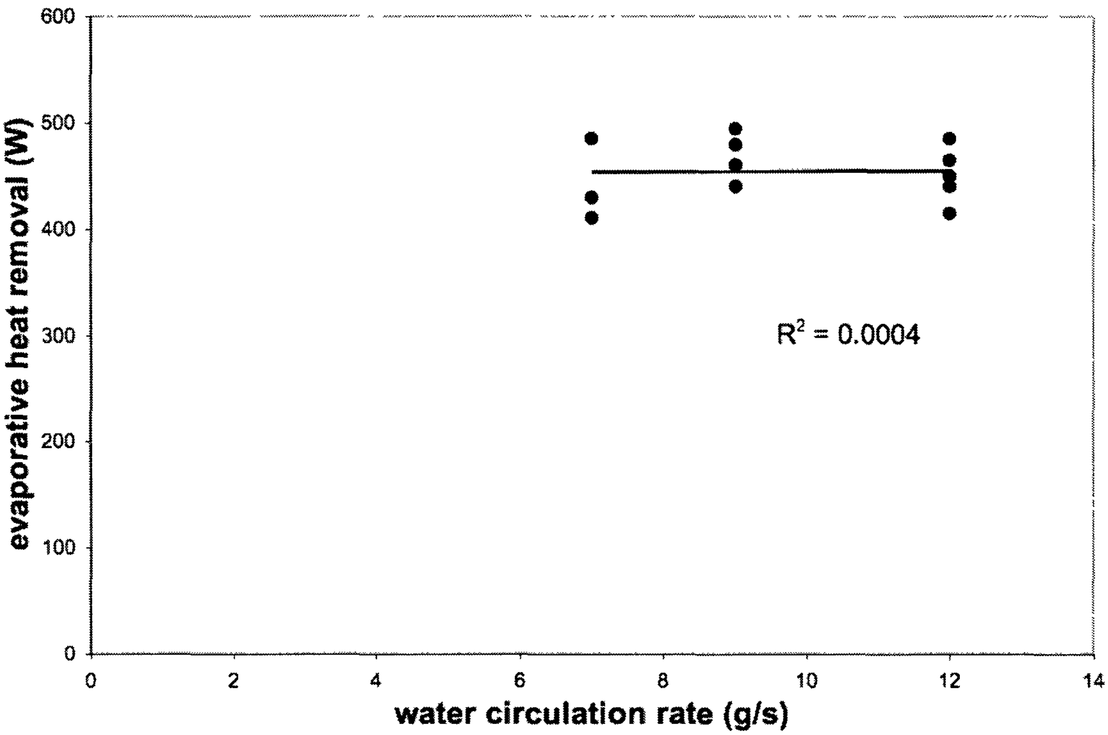

Figure 5.8: Effect of water circulation rate on evaporative heat removal

Figure 5.7 shows the effect of the air flow rate on the quantity of cooling. Increasing the volume of air increases the amount of evaporation, which is expected because of the greater quantity of water vapour that can enter the air phase before saturation. The higher level of evaporation for large volumes of air agrees with the larger drop in outlet temperature observed earlier in Figure 5.4. For completeness, Figure 5.8 shows no relationship between circulation rate and the amount of evaporation. A higher water velocity would be expected to reduce the temperature polarization boundary layer near the surface of the membrane, which would keep the surface temperature of the water warmer and closer to the bulk water temperature and thus improve evaporation at the higher temperatures. However, there was an opposing factor occurring that caused the increased water velocity to inhibit evaporation as well. It is likely that the increased circulation rate forced the water to exit the condenser at a lower temperature than if the circulation rate was lower, which occurs when the heat output from the chiller is constant. A lower condenser exit temperature leads to a lower temperature at the entrance of the modules, which would reduce the vapour pressure and thus the driving force for membrane evaporation. Lower module entrance temperatures were observed at high circulation rates, so this is the likely explanation for the negation of the original benefit of reducing temperature polarization.

A summary of the conditions for the runs with the best cooling quantity and best cooling quality is shown in Table 5.3. The highest evaporative heat removal was 580 W, which was achieved with a high air flow of 135 L/min and also one of the highest heat loads tested over the course of the runs. The high heat load raised the module entrance temperature and the air flow was set to the maximum for the modules; both factors positively contributed to more evaporation. The higher circulation rate likely did not affect the evaporation directly, as suggested by Figure 5.8, but it did allow more heat to be removed from the condenser so that a heat load could be used. In contrast, the lowest membrane outlet temperature of 39.5°C occurred with a lower heat load and circulation rate than the run with the most evaporation, but it also benefited from a high air flow rate. These three effects agree with Figures 5.3, 5.4, and 5.5, respectively. The nearly identical coefficients of performance between the two superlative runs indicated that the chiller was not overworked or underperforming because of the conditions of those specific runs.

Table 5.3: Conditions of superlative runs

	most evaporation	lowest temperature
Heat load (W)	510	435
Air flow (L/min)	135	135
Circulation rate (g/s)	12	7
Membrane inlet/condenser out (°C)	60.3	59.9
Evaporative cooling (W)	580	500
Membrane outlet (°C)	45.2	39.5
Chiller coefficient of performance	1.66	1.64

The results of the membrane evaporation system were then compared to cooling tower data in the literature, as presented in Table 5.1. However, since the data covers a wide range of volumes, as well as a range of entrance temperatures, it was necessary to normalize the data to the same scale. The heat removal power (based on $mC_P\Delta T$) was divided by the volume of the tower for each tower, and also divided by the vapour pressure of water at the entrance of each tower. This normalization provides a clearer visual picture of which towers provide the greatest quantity of heat removal, with giving an absolute advantage to large towers or towers with high entrance temperatures. Figure 5.9 compares the normalized heat removal in the cooling tower studies with that of a sample run using membrane evaporation. One study (Naphon, 2005) was a bench-scale column with a high quantity of cooling. However, it was suspected to use more air than other studies, so Figure 5.10 introduces a further normalization that divides the heat removal by the air flow, providing a measure of air effectiveness.

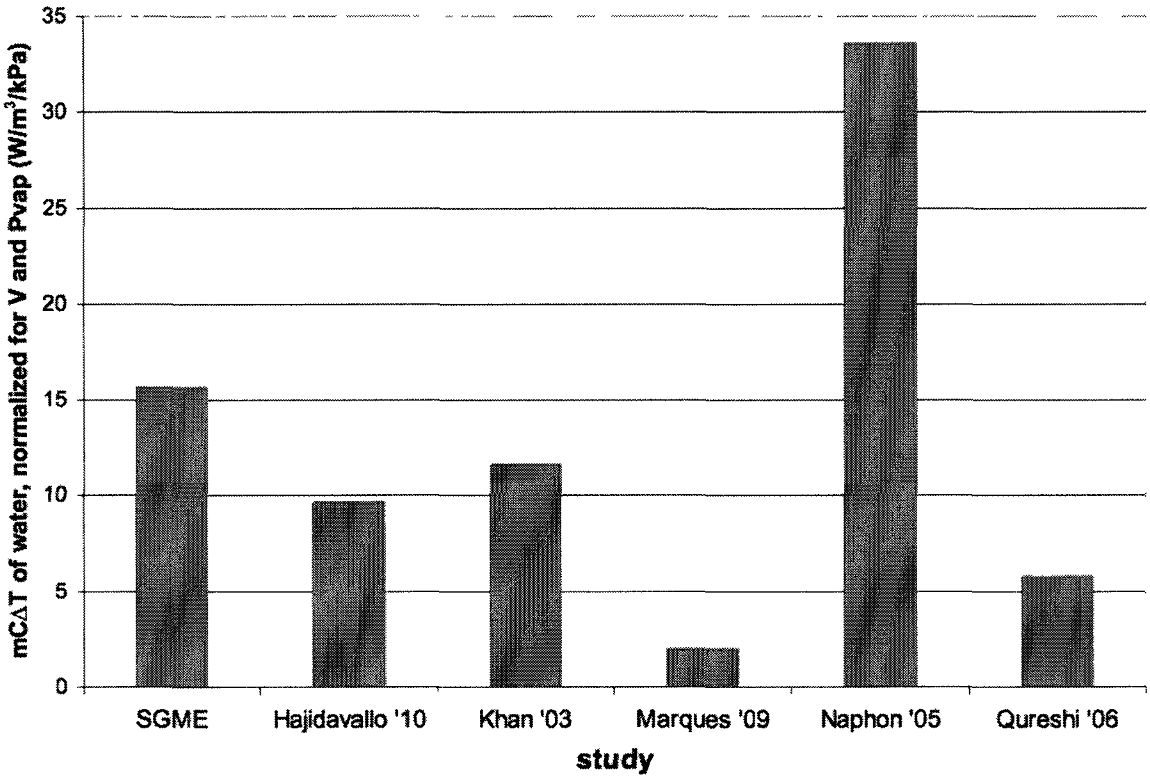

Figure 5.9: Normalized heat removal of membrane evaporation and cooling towers

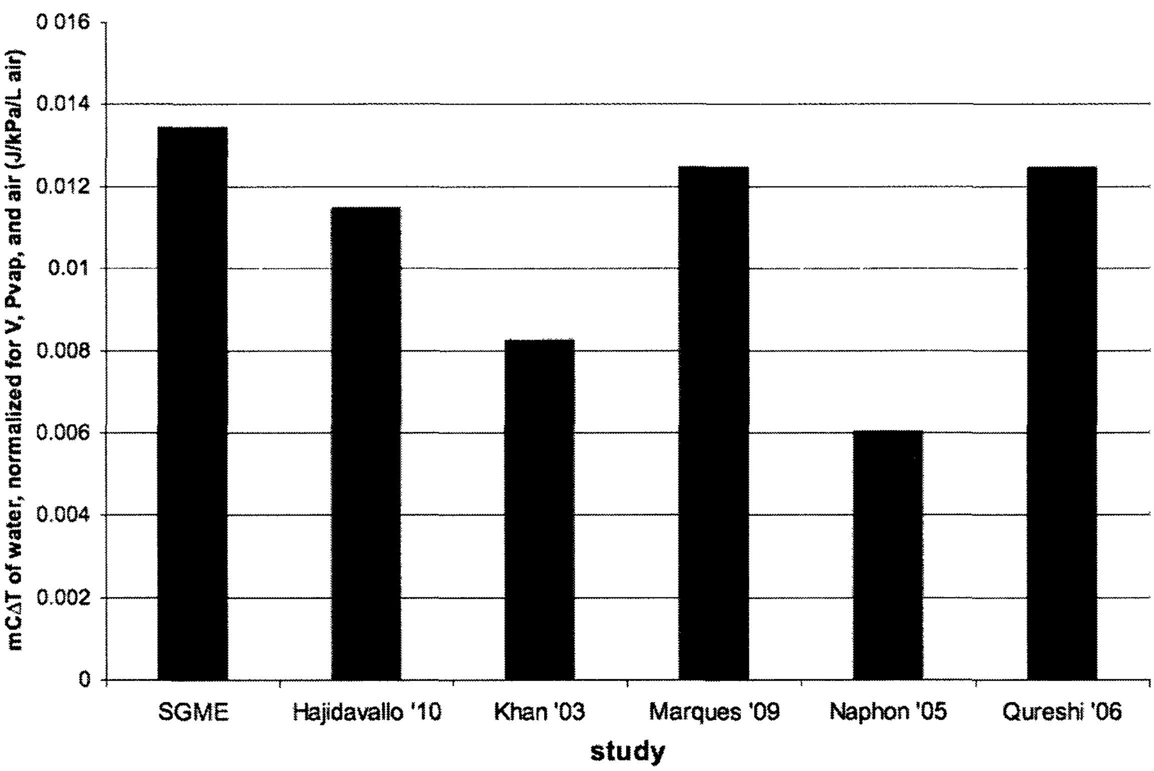

Figure 5.10: Heat removal of towers, normalized for air flow rate

The benefits of membrane evaporation compared to cooling towers become apparent in Figures 5.9 and 5.10. Figure 5.9 is an indicator of the quantity of cooling, when adjusted for scale and temperature. The SGME run had higher evaporation than all the larger towers except one, which was a bench-scale column using excess air. Since it had excessive air, it would be expected that more heat removal occurred, as suggested by Figure 5.7. However, a normalization for air flow rate, as shown in Figure 5.10, gives an indication of the efficiency of the cooling system – it is desirable to have more heat removal per unit of air. In terms of efficiency, the SGME system was still higher than the other studies with large towers. Interestingly, the two studies that could be considered bench scale (Khan *et al.*, 2003 and Naphon, 2005) both decreased upon normalization for air flow. This would suggest that small scale cooling towers are less efficient for air use than larger towers. Since droplet size can be consistent regardless of tower size, larger towers can contain more droplets and thus more surface area for evaporative cooling, and larger towers also provide a longer residence time as droplets fall. These detriments of small scale make the efficiency of the SGME system, smaller than the bench scale columns, even more impressive. Despite the inherent drop in air efficiency of small towers, the membrane system was still the most efficient cooling method in the comparison. Figure 5.11 is a plot of normalized heat removal (not adjusted for air flow) against air flow per unit volume. Only the three large towers are shown so that a clear line can be fitted while excluding bench scale data with excessive air. When several data points from the membrane cooling runs are added, all points lie above the efficiency line produced from the tower data. Regardless of operating conditions, sweeping gas membrane evaporation is a more efficient use of air for cooling than towers.

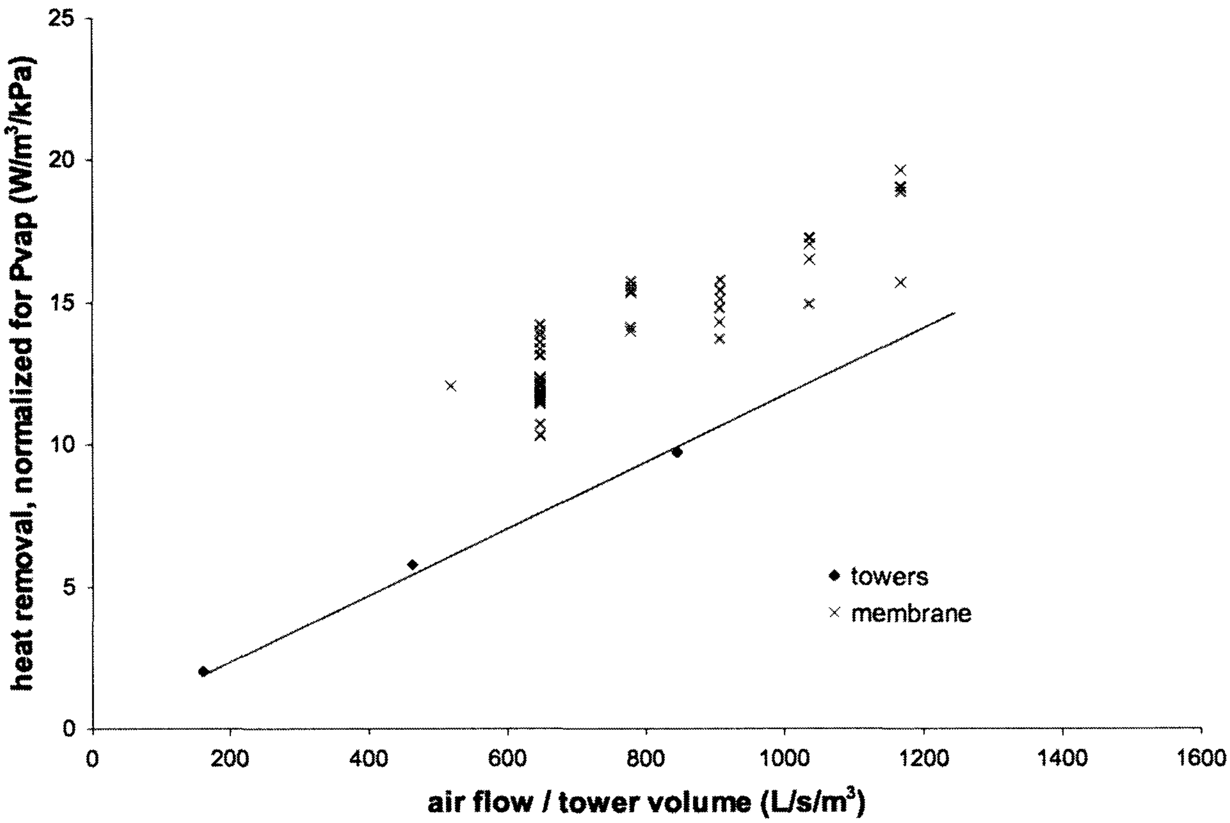

Figure 5.11: Cooling efficiency of membrane runs compared to large cooling towers

A membrane module is more efficient than a cooling tower in two ways. First, there is no need to have droplets – and all the air gaps between droplets – in a hollow fiber membrane module. The fibers carry a continuous stream of water, which takes up much less volume than the same mass sprayed into fine droplets. This reduces the size footprint of the modules, providing a volume efficiency. Secondly, the membrane acts as barrier to improve the effectiveness of evaporation compared to other methods of heat transfer. While droplets give a high surface area for evaporation, the high surface area also leads to sensible heat transfer between the liquid and air to form a thermal equilibrium. If evaporation would lead to a water temperature lower than the entering air, the air warms up the water. Similarly, hot water can also remove heat by the less effective sensible heat transfer rather than by evaporation. In a membrane module, there is only a small amount heat transfer across the insulative polymer membrane. It is possible for water to be cooled below the temperature of nearby air because air cannot transfer heat across the membrane and back into the water. In addition, the hydrophobic nature of the membrane means that water does not conduct heat across the membrane; instead, the only heat removal comes from evaporation, which has a very high latent heat. Forcing all cooling to come from evaporation also reduces the water losses in a membrane module compared to a cooling tower. For all membrane runs, 1.5% (standard deviation of 0.3%) of the water entering the module is lost to evaporation. In comparison, the lowest water loss for a tower was found to be 0.5% (Pearlmutter *et al.*, 2008), with other studies as high as 5% (Hajidavaloo *et al.*, 2010). Thus, sweeping gas membrane evaporation also has an advantage over cooling towers in the form of evaporation efficiency.

Conclusions

It has been shown that a sweeping gas membrane evaporation system is a more effective cooling method than cooling towers. The SGME system operates at a smaller scale than towers, uses a smaller volume footprint, and makes more efficient use of the air for evaporation. As such, this technology is recommended for miniature cooling applications, such as personal or electronics cooling.

There were limitations with the amount of air that could be used. All results suggest that more air improves the performance of the modules, but the modules were limited to 45 L/min of air. Larger hollow fiber membrane modules are commercially available, which can accommodate a larger air flow rate, but come at the expense of a larger size footprint.

Adding more small modules would be another way to increase the total amount of air, but there will likely be diminishing cooling effectiveness with each additional module in series.

Membrane runs could also be investigated at fixed membrane inlet temperatures rather than fixed heat loads. Having a fixed inlet temperature can allow direct comparisons between other data in the literature without normalizing for vapour pressure. A fixed inlet temperature can be achieved with a hot water bath, but a hot water bath does not reflect real-world cooling applications like a chiller would.

Overall, steady-state cooling can be achieved with a sweeping gas membrane evaporation system. Results showed that decreasing the heat load and circulation rate improved the quality of cooling, while the quantity of cooling increased with increased heat input. Both quality and quantity improved with a higher air flow rate. When normalized for volume, inlet temperature, and air flow, the membrane modules are more effective at cooling than both bench-scale and industrial size cooling towers.

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Chapter 6: Discussion of results

Discussion of vapour compression refrigeration results

Chapter 4 outlines the results of producing a cooling system that improves the coefficient of performance of a chiller based on a vapour compression refrigeration cycle. The chiller was designed with an air-cooled condenser, and can cool water exiting the evaporator to 15-20°C. Because the chiller operated in hot environments, air temperature was high and the COP was less than 1.0. The condenser was modified to allow a stream of water to absorb heat from the condensing refrigerant, with the stream of water cooled by a membrane evaporative system before being recycled back to the condenser.

The membrane evaporative system provided enough cooling at the condenser to allow the chiller to operate at steady state. The chiller could remove heat from a heat source in the range of 250 to 600 W, and cooled water leaving the evaporator to 16°C. The evaporator outlet temperature was held constant because, upon startup, the chiller ran at full capacity to drop the source to its target temperature, then scaled back to balance the supplied heat load after a few minutes. For all the different configurations and parameters tested, each run resulted in steady-state cooling until the water circulating through the membrane modules had evaporated off. All runs had a COP above 1.0.

Changing the parameters of the system produced different effects on the coefficient of performance. Preliminary work determined that the three modules should be arranged such that water flows through the tube side of all of them in series, while the air stream passing through the shell side is split into three parallel streams for each module. This cross-current arrangement allowed the maximum amount of air to be supplied to the system. Indeed, Figure 4.7 showed that an increase in air flow will increase the COP, largely because the increased volume of unsaturated air allows more cooling water to evaporate before the air becomes saturated, and the increased amount of evaporation results in more cooling at the condenser. While the effect was not as statistically significant as air flow, increasing the water circulation rate also somewhat improved the COP (Figure 4.8) because of the increased turbulence inside the heat exchanger at the condenser compared to a low circulation rate. Figure 4.6 demonstrated that there is an optimal heat load to be supplied to the chiller to maximize COP. When all other parameters are constant, a low heat load corresponds to a low COP because the chiller is not providing a large amount of cooling. When the heat load is high, the system temperature – especially the condensation temperature – increases because the refrigerant is absorbing more heat at the evaporator. As the condensation temperature increases, so too does the condensation pressure, forcing the compressor to do more work to bring the refrigerant vapour to the condensation pressure. This increased amount of work increases the electrical demand of the system, which reduces the COP. For our set of parameters, an optimal heat load was found at 435 W, with other conditions outlined in Table 4.2. These conditions resulted in the highest observed COP of 1.83. Based on the temperatures from the run from the highest actual COP, the theoretical maximum COP was 5.79, resulting in a cycle that performed at 32% of the maximum.

The condenser modification significantly improved the performance of the chiller. It has been previously established that using an evaporative condenser instead of an air-cooled condenser improves the COP and the range of operable evaporation temperatures (Hosoz and Kilicarslan, 2004). By converting from an air-cooled condenser to the membrane evaporative system, the COP of the chiller was nearly doubled and the cooling process used a smaller volume of air. Circulating water returned to the condenser at 40-50°C, which is similar to the air temperatures experienced in the fan-based system, but the improved heat transfer that came from using water instead of air greatly reduced the refrigerant condensation temperature and improved the COP. The observed relationship between condensation temperature and COP was shown in Figure 4.9. A separate study (Pearson, 2008) suggests that for R-134a refrigerant, the COP should theoretically reach 4.60, but it was shown mathematically rather than experimentally, and was based on a low condensation temperature of 30°C. While a lower condensation temperature will improve COP, a chiller actually becomes unnecessary if the condensation temperature is below that of the heat source at the evaporator. If the heat source was above 30°C, such as in a hot environment, it is more effective to use whatever is cooling the condenser for cooling the environment, rather than introducing a heat pump cycle between a hot source and a relatively cool sink. In addition, if the temperature of the water exiting the condenser drops too low, evaporation becomes less favourable because of the exponential decrease in vapour pressure with respect to temperature (Smith and Van Ness, 2005). As such, while it still remains desirable to reduce the condensation temperature and improve the electrical efficiency of the chiller, the environmental conditions must also be considered to determine if such operation is practical. The results of Chapter 4 demonstrated that a membrane evaporative system improved the performance of a chiller designed to operate in hot environments.

Discussion of sweeping gas membrane evaporation results

Chapter 5 describes the cooling performance of a sweeping gas membrane evaporative (SGME) system and compares it to the conventional evaporative cooling technology of cooling towers. Using the condenser of a chiller as a heat source, a heated stream of water was cooled by a set of three hydrophobic hollow fiber membrane modules (Liqui-Cel by Membrana, Charlotte, NC). The hot water had a high water vapour pressure compared to the water vapour pressure in the air stream passing on the shell side of the modules. This vapour pressure difference forced feed water to evaporate and cross the membrane as vapour, cooling the circulating water in the process.

The membrane evaporative system provided steady cooling at constant temperature. Though water is constantly being lost to evaporation, the rate of evaporation was constant, as suggested by Figure 5.2, which resulted in a constant rate of heat removal. For the various runs, there was evaporative heat removal of 300-650 W, largely depending on the heat load. The evaporative heat removal is larger than the heat load because there is also a contribution of heat from the work of the compressor.

Parameters were adjusted to determine their effects on the amount of evaporation and the system temperatures. It was determined in Chapter 4 that arranging three modules in cross-flow with air on the shell side was the optimal arrangement, and the increased volume of air was also favourable for increasing the amount of evaporation. Figure 5.7 showed that evaporative heat removal increased with greater air flow, and the results were only limited by the maximum air flow recommended by the module manufacturer. That increase in evaporation corresponded to a decrease in membrane outlet temperature, as observed in Figure 5.4. Figure 5.5 showed that increasing the circulation rate increased the cooled outlet temperature; this arises from a heat balance where, for a constant heat removal suggested by Figure 5.8, an increase in mass decreases the temperature drop along the cooling modules. Thus, to cool the water before entering the condenser, the circulation rate should decrease. However, the change in circulation rate does not correspond to a significant change in the amount of evaporative heat removal, as shown in Figure 5.8. It is probable that, regardless of circulation rate, only a fixed amount of vapour can cross the membrane for a given air flow rate and temperature, suggesting that diffusion across the membrane into saturated air is the limiting step of this cooling process. This circulation rate effect is noticeable when comparing the run with the largest heat removal against the run with the coolest outlet temperature (Table 5.3): despite the other conditions being similar, the run with the largest heat removal had a high circulation rate while the run with the lowest outlet temperature had a low circulation rate. As such, there will be different operating conditions depending on whether heat removal – quantity of cooling – or outlet temperature – quality of cooling – is most important.

The novelty of the membrane evaporation system is the ability to cool water. Though the similar technology of membrane distillation is quite developed (El-Bourawi *et al.*, 2006), most studies assume the liquid feed temperature to be constant, either because the feed is pre-heated to a desired temperature or because the feed tends to be a concentrated solution with a low vapour pressure to begin with (Sudoh *et al.*, 1997). A membrane evaporation study that used a similar technique to humidify air for air conditioning purposes (Johnson *et al.*, 2003) experienced temperature drops of less than 2°C – compared to a drop along the modules of up to 20°C in this experiment – although their tests were performed near room temperature, which is less favourable for evaporation. A patent describes using vacuum membrane evaporation to cool water to near the freezing point (Lomax and Moskito, 1999), but it does not give specific temperatures, and the low temperatures become possible only with the introduction of the electrical demand of a vacuum pump to drive the water vapour pressure low enough to form an equilibrium at those temperatures. The sweeping gas membrane evaporation system presented in this work has the ability to cool water by a significant temperature drop without resorting to the high-energy demands of a vacuum system.

The sweeping gas membrane evaporation system was compared to cooling towers, since both techniques are based on cooling water by evaporation into air. Several studies on cooling towers were listed in Table 5.1, ranging from a 0.011 m^3 bench scale column (Naphon, 2005) to a commercial tower with a volume of 1785 m^3 (Marques *et al.*, 2009). When normalized for volume and water inlet temperature, Figure 5.9 showed that the SGME system had larger heat removal than all but one tower. Closer investigation of that tower revealed that it was the bench-scale column and it used far more air, per unit volume, than this system. Since more air will increase the amount of evaporative heat removal, a further normalization based on air flow was introduced in Figure 4.10. For a constant volume, air flow rate, and inlet temperature, the SGME system produced more cooling than any tower. Thus, the membrane-based system is a more compact method of heat removal than a conventional cooling tower, and there is the benefit of providing more cooling simply by adding on more modules. Figure 5.11 showed that for a given air flow rate, the SGME membrane system produced more cooling than towers. The membrane evaporation system is compact both in terms of total volume and air flow because it uses a membrane to separate liquid water and air. The membrane barrier forces nearly all heat transfer to occur through the evaporation process, rather can conduction across the membrane or between liquid water and air. A study that developed a model of heat transfer within a cooling tower (Khan *et al.*, 2003) found that evaporation contributed to as little as 63% of the total heat transfer. In comparison, the SGME runs averaged 89% of the total heat removal through evaporation. Since evaporation requires much more energy per unit mass than transferring sensible heat between fluids, the system that makes more effective use of evaporation produces more cooling for a smaller mass of water. Additionally, having a membrane barrier prevents liquid water droplets from being entrained in the passing air. Cooling towers can have as much as 5.0% of the circulating water lost during the residence time in a tower (Hajidavaloo *et al.*, 2010) due to entrainment and evaporation, but the SGME system loses an average of 1.5% to only evaporation. Reducing the amount of water lost in the system decreases the demand for fresh water, improving the efficiency of the cooling system.

The sweeping gas membrane evaporation system is a more compact and effective means of heat removal than conventional cooling towers. The system removes more heat per unit volume than cooling towers, and the modular design allows the system to be scaled to different cooling capacities simply by adding or removing membrane modules. It is more effective than cooling towers both in terms of the increased heat removal per unit of air and in the reduction of water losses from evaporation. Reducing air requirements reduces energy demand for blowing, allowing a SGME system to be more energy efficient than conventional evaporative cooling. Chapter 5 demonstrated that a SGME system is a superior substitute to cooling towers.

Refrigeration and membrane evaporation hybrid system

There were two components to the hypothesis of this thesis. Chapter 4 demonstrated that an evaporation-based condenser improves the coefficient of performance of a chiller compared to one with an air-cooled condenser. Chapter 5 compared a sweeping gas membrane evaporation system with cooling towers to show the superior cooling capacity of the SGME system. Though refrigeration and membrane evaporation are different concepts, the results of both sections reveal that it is possible, and preferable, to use a chiller together with a sweeping gas membrane evaporation system for enhanced cooling performance.

The SGME system works with the chiller to provide steady-state cooling. As was shown in Figures 4.3 and 5.2, the temperatures and energy transfers within the coupled system reach a steady state after approximately 20 minutes of start up, continuing until the reservoir of circulating water is exhausted by evaporation across the membranes. The chiller was originally designed to cool water to less than 20°C when a supply of air at 40-50°C in a hot environment is available for cooling the condenser. The performance of the chiller improved upon introducing evaporative cooling at the condenser, fed by circulating water entering the heat exchanger at 40-50°C, depending on specific conditions of the run. While a conventional method of evaporative cooling above the wet bulb temperature is a cooling tower, the large volume and air requirements of a tower were replaced with a small sweeping gas membrane evaporation system. The compact nature and modular arrangement of the system gave flexibility in the design of a system that could provide sufficient heat removal from the condenser with a lower air requirement.

The results of the two chapters are linked by the quality and quantity of cooling in the SGME system. Quality is measured as the membrane water outlet temperature, which corresponds to the inlet temperature at the condenser, while the quantity of cooling is the evaporative heat removal. As was shown in Figure 4.9, reducing the condensation temperature increases the COP of a vapour compression cycle. The reduced condensation temperature decreases the work done by the compressor to pressurize the refrigerant vapour, which reduces the electrical demand of the chiller. One way to reduce the condensation temperature is to reduce the condenser water inlet temperature. This is achieved by reducing the water temperature at the outlet of the membrane modules, accomplished by either increasing the air flow (Figure 5.4) or decreasing the circulation rate (Figure 5.5). Indeed, an increase in air flow did correspond to an improvement in chiller performance (Figure 4.7), but the same result was not observed for circulation rate (Figure 4.8), which suggests that increasing the circulation rate increases turbulence and improves heat transfer at the condenser. This leads to another method of potentially reducing the condensation temperature: improving heat transfer. Increasing the circulation rate at the condenser led to an increase in chiller performance, but it had no significant effect on the quantity of evaporative cooling (Figure 5.8). While a reduced circulation rate did produce a lower condenser inlet temperature, the condenser outlet temperature was higher. This is seen by comparing the run with the highest COP (Table 4.2) against the run with the lowest membrane outlet temperature (Table 5.3). The high COP run had a circulation rate of 12 g/s and a condenser outlet temperature of 55.0°C. The low temperature run had a circulation of 7 g/s and a condenser outlet temperature of 59.9°C. The higher condenser outlet temperature led to a slightly higher average temperature within the condenser, which decreased the temperature difference with the refrigerant and reduced the heat transfer across the condenser. This circulation rate decrease led to a 10% decrease in COP compared to the high

COP run. It is clear that the quantity and quality of cooling in the membrane modules affects the performance of the vapour compression cycle via heat transfer in the condenser.

Improving the coefficient of performance is important. When the SGME system stands alone, it can be debated whether a larger quantity of evaporative cooling or a low outlet temperature is desirable, depending on the needs of the situation. However, when it is coupled with a chiller, it is important to have efficient heat transfer in the condenser to reduce the condensation temperature as much as possible and reduce the electrical demand. This saves energy and the improved heat transfer allows larger heat loads at the evaporator to be used. As such, for a SGME system coupled with the vapour compression cycle, increasing the air flow and increasing the water circulation rate, while maintaining an optimal heat load, provides the most enhanced performance for the chiller.

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Chapter 7: Conclusions and future recommendations

Conclusions

A sweeping gas membrane evaporation (SGME) system was developed to supply cooling to the condenser of a vapour compression refrigerator. A chiller designed for hot environments with an air-cooled condenser was provided in an effort to improve its performance. As mentioned in Chapter 1, two hypotheses were made regarding the SGME-chiller arrangement:

- A small membrane cooling device should provide the cooling capacity required for the steady-state operation of a commercial chiller, and do so with higher performance than a conventional air-cooled condenser.
- Sweeping gas membrane evaporation removes more heat per unit volume and is more efficient in the use of air than conventional evaporative cooling technology, such as cooling towers.

The SGME system did supply sufficient cooling capacity for the steady-state operation of the chiller. Figures 4.3 and 5.2 show that after an initial start-up phase of approximately 20 minutes, temperature and power profiles throughout the system remained constant for the duration of the runs. Given a sufficient supply of water and electricity, the SGME-chiller system is expected to run indefinitely.

The coefficient of performance of the chiller increased when converting from an air-chilled condenser to the SGME system. The chiller was designed for operation in hot environments and could achieve a COP of 1.0 using air for cooling. In contrast, every run performed with the SGME apparatus had a COP above 1.0, with a maximum of 1.83. The circulating water entering the condenser was at approximately the same temperature as air in the environments the chiller was designed for (40-50°C), and the SGME required only 135 L/min of air, compared to over 1000 L/min supplied by cooling fans. Additionally, it was found that the COP could be increased by increasing the air flow rate and the water circulation rate, while the chiller had an optimum heat load of 435 W. The SGME improves the performance of the chiller and reduces the demand for air compared to the air-cooled arrangement.

The SGME system removes more heat on a unit volume basis and is more efficient than cooling towers. As shown in Figure 5.10, when normalized for volume and air flow, the SGME system provides more heat removal than the investigated cooling towers. While the membrane-based system was much smaller than the towers, the flexibility of attaching additional modules allows the SGME to be scaled up in situations where more cooling is demanded. Figure 5.11 demonstrates that the membrane system removes more heat per unit volume of air than cooling towers, reducing the need for energy-intensive and noise-producing fans. By reducing the air requirements, energy requirements are also reduced, allowing SGME to be more energy efficient than cooling towers. The SGME system is more compact because the water does not need to be sprayed; the hollow fiber membranes act as the surface area for mass transfer rather than the surface of droplets. The system is also more compact because there are no losses of liquid water due to entrainment in air; the membrane forces water to evaporate before it is transferred to the air stream. The design of the sweeping gas membrane evaporation system allows it to be more compact and efficient than conventional evaporative cooling systems.

In summary, the SGME system was a superior method of cooling the condenser of a vapour compression refrigerator. The membrane-based system was also superior to cooling towers in terms of size and air efficiency.

This work described a novel method of incorporating established membrane separation technology with cooling. While earlier studies have shown that evaporative cooling is a very effective method of heat removal, there has been no prior investigation into using the evaporation inherent in membrane distillation for cooling. This work demonstrates that the sweeping gas membrane evaporation technique can successfully provide cooling, and does so with higher efficiency than conventional evaporative technology.

Recommendations

Though the sweeping gas membrane evaporation system was superior to both air-cooled condensers and cooling towers, there is room for improvement. As shown in Figures 4.7 and 5.7, increasing the air flow increases both the COP of the connected refrigerator and the quantity of evaporation. Thus, future designs based on this work should consider increasing the air flow further. However, increasing air flow may require larger membrane modules or other equipment that would increase the system's footprint. Another consideration about increased air flow is the resulting increased air pressure. As the air pressure increases, so too does the driving force to send air through the membrane into the circulating water. To counter this, the water pressure must be increased, which requires larger pumps and could be impacted by the liquid entry pressure of 2.0 bar of the modules. Changing the membrane to a more hydrophobic material, such as PTFE, would increase the liquid entry pressure to compensate. Increasing air flow appears to be the most effective method of improving performance and efficiency, but it comes with potential drawbacks.

A second concern is that the SGME has one drawback compared to the air-cooled condenser. The SGME system requires a supply of water to replenish evaporative losses. This is difficult to accommodate in the environments the chiller was originally designed for, where only ambient air is available. As such, a water recycling system could be investigated to further reduce the need to replace water: the vapour lost in the exiting air stream would be condensed and returned to the water reservoir. However, the recycling system would need some form of cooling itself, increasing the footprint of the original design. Instead, the SGME system could find use in situations where water is available. Applications where cooling is needed and water is available include air conditioning, microreactor and electronics cooling, and any industrial processes where a conventional cooling tower is too large. The membrane barrier in SGME limits the release of water-borne contaminants to the air stream, so the system can be used in situations where air quality is important, such as air conditioning. While the SGME system does have some drawbacks in its design compared to other refrigerator condensers, it can still be useful for replacing small-scale evaporative cooling systems.

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Appendix A: Run conditions

The experimental conditions of the runs conducted with the SGME system and chiller are shown in Table A.1. The runs are listed in chronological order. There were preliminary tests with different apparatus arrangements and run conditions, but those did not achieve steady-state cooling and are not included.

Table A.1: Experimental conditions of SGME-chiller runs

Date	Heat load (W)	Water circulation (g/s)	Air flow rate (L/min)
3-Feb-10	275	7	25
4-Feb-10	345	7	25
4-Feb-10	350	7	25
5-Feb-10	415	7	25
5-Feb-10	430	7	25
8-Feb-10	380	9	25
8-Feb-10	400	9	25
9-Feb-10	370	7	25
9-Feb-10	390	9	25
10-Feb-10	370	9	25
11-Feb-10	280	7	25
12-Feb-10	305	7	25
16-Feb-10	300	9	25
16-Feb-10	310	9	25
17-Feb-10	305	12	25
17-Feb-10	395	12	25
18-Feb-10	350	12	25
19-Feb-10	320	12	25
19-Feb-10	365	12	25
22-Feb-10	275	9	25
22-Feb-10	320	9	25
22-Feb-10	330	9	25
23-Feb-10	370	12	30
23-Feb-10	360	12	30
24-Feb-10	395	9	25
24-Feb-10	385	12	30
24-Feb-10	380	12	30
25-Feb-10	305	12	20
5-Mar-10	300	12	25
5-Mar-10	390	12	25

8-Mar-10	375	12	25
8-Mar-10	370	12	25
9-Mar-10	380	12	30
9-Mar-10	370	12	30
10-Mar-10	385	12	35
10-Mar-10	385	12	35
11-Mar-10	365	12	40
12-Mar-10	375	12	35
12-Mar-10	385	12	40
15-Mar-10	380	12	35
15-Mar-10	385	12	40
16-Mar-10	325	7	35
16-Mar-10	315	9	35
17-Mar-10	350	7	40
17-Mar-10	364	9	40
24-Mar-10	510	12	45
24-Mar-10	395	12	45
25-Mar-10	285	12	45
26-Mar-10	435	7	45
26-Mar-10	435	12	45

Appendix B: Sample calculations

Unless otherwise specified, sample calculations were based on the run that produced the highest COP, completed March 26, 2010. Physical and thermodynamic properties of water were collected from reference tables (Smith and Van Ness, 2005).

Experimental measurements

Data collected directly from the experiment is presented in Table B.1. Unless otherwise indicated by time, readings are based on the steady-state phase that was fully established after the first hour. Measurements were recorded every 5 minutes.

Table B.1: Experimental data from March 26, 2010

Property (units)	Measurement
Heat load from heaters (W)	435 ±2
Voltage supplied to compressor (V)	28 ±0.5
Current drawn from compressor (A)	8.5 ±0.3
Air flow per modules (L/min)	45 ±0.5
Water circulation at evaporator (g/s)	25 ±0.5
Water circulation at condenser (g/s)	12 ±0.5
Evaporator water inlet temperature (°C)	24.5 ±0.3
Evaporator water outlet temperature (°C)	16.8 ±0.3
Condensation temperature (°C)	53.0 ±0.5
Condenser water inlet temperature (°C)	43.7 ±0.3
Condenser water outlet temperature (°C)	55.0 ±0.3
First module water outlet temperature (°C)	50.6 ±0.3
Second module water outlet temperature (°C)	47.1 ±0.3
Third module water outlet temperature (°C)	44.8 ±0.3
Air inlet temperature (°C)	27.1 ±0.3
Air outlet temperature (°C)	53.6 ±0.3
Air inlet relative humidity (%)	18 ±5
Air outlet relative humidity (%)	96 ±5
Reservoir mass at 55 min (g)	2290.0 ±0.5
Reservoir mass at 75 min (g)	2068.2 ±0.5
Reservoir mass at 95 min (g)	1833.2 ±0.5
Reservoir mass at 115 min (g)	1608.3 ±0.5

Chapter 4 calculations

The electrical demand of the compressor is calculated with Watt's Law:

$$P = IV$$
$$P = 8.5 \text{ A} \times 28 \text{ V}$$
$$P = 238 \text{ W}$$

Together with the heat load, the coefficient of performance is calculated:

$$\text{COP} = \frac{\text{heater load}}{\text{electrical demand of compressor}}$$
$$\text{COP} = \frac{435 \text{ W}}{238 \text{ W}}$$
$$\text{COP} = 1.83$$

Chapter 5 calculations

To determine the quantity of cooling by evaporation, the mass of the circulating water reservoir was recorded every 5 minutes. With no leaks in the system, and the membranes impermeable to liquid water, it was assumed that all water mass loss could be attributed to evaporation. Using a linear fit in Microsoft Excel, the rate of mass loss was found to be 11.39 g/min, or 0.1898 g/s. At the membrane outlet temperature of 44.8°C, the heat of evaporation of water is 2397.1 J/g. The evaporative heat removal is then calculated:

$$Q_{evap} = \dot{m}_{evap} \Delta H_{vap}$$
$$Q_{evap} = 0.1898 \tfrac{\text{g}}{\text{s}} \times 2397.1 \tfrac{\text{J}}{\text{g}}$$
$$Q_{evap} = 455 \text{ W}$$

With the rate of evaporation known, the fraction of circulating water lost to evaporation can also be determined:

$$\% \text{ loss} = \frac{\text{evaporation rate}}{\text{circulation rate}} \times 100\%$$

$$\% \text{ loss} = \frac{0.1898 \text{ g/s}}{12 \text{ g/s}} \times 100\%$$

$$\% \text{ loss} = 1.6\%$$

The total heat removal from the water circulating through the membrane modules can be attributed to evaporation, conduction across the membrane, and losses to the environment. The total heat removal, based on the temperature drop of the water, is:

$$Q_{mem\ cooling,total} = \dot{m}_{water} C_{P,water} \left(T_{w,in} - T_{w,out}\right)$$

$$Q_{mem\ cooling,total} = 12 \tfrac{\text{g}}{\text{s}} \times 4.18 \tfrac{\text{J}}{\text{g}\bullet\text{K}} \times (55.0 - 44.8)\ \text{K}$$

$$Q_{mem\ cooling,total} = 511.6 \text{ W}$$

Knowing the total heat removal, the fraction of cooling associated with evaporation is:

$$\% \text{ evaporative cooling} = \frac{Q_{evap}}{Q_{mem\ cooling,total}} \times 100\%$$

$$\% \text{ evaporative cooling} = \frac{455 \text{ W}}{512 \text{ W}} \times 100\%$$

$$\% \text{ evaporative cooling} = 89\%$$

For reference, the heat absorbed by the air inside the modules is calculated first by determining the molar flow rate, based on the ideal gas law at standard conditions:

$$\dot{n}_{air} = \frac{P\dot{V}_{air}}{RT}$$

$$\dot{n}_{air} = \frac{101.3 \text{ kPa} \times 2.25 \tfrac{\text{L}}{\text{s}}}{8.314 \tfrac{\text{L kPa}}{\text{mol K}} \times 298 \text{ K}}$$

$$\dot{n}_{air} = 0.0920 \tfrac{\text{mol}}{\text{s}}$$

When using the heat capacity of a gas, several formulae make use of tau, the ratio between the final and initial temperatures:

$$\tau_{air} = \frac{326.8 \text{ K}}{300.3 \text{ K}} = 1.088$$

Using this, the heat absorbed by the air is:

$$Q_{air} = \dot{n}_{air} R\left[A + \frac{B}{2}T_{air,in}(\tau + 1) + \frac{D}{\tau T_{air,in}^2}\right](T_{air,out} - T_{air,in})$$

where the heat capacity constants for air are $A = 3.355$, $B = 0.000575$, and $D = -1600$.

$$Q_{air} = 0.092 \times 8.314\left[3.355 + \frac{0.000575}{2} \times 300.3 \times (1.088 + 1) - \frac{1600}{1.088 \times 300.3^2}\right](326.8 - 300.3)$$

$$Q_{air} = 67.8 \text{ W}$$

The heat absorbed by the air is similar to the difference between the total heat removal and the heat removed by evaporation. There is some variation due to heat loss to the environment and water vapour condensing in the air stream exiting the module.