

**Personal-portable Cooling Garment Based on Adsorption
Vacuum Membrane Evaporative Cooling**

By

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ABSTRACT

A cutting edge man-portable AVMEC cooling garment was demonstrated to be able to provide sufficient cooling for personnel working at mediate activity loads. Studies were first carried out in a well controlled vacuum desiccator at room temperature to elucidate the effects of several key parameters on the performance of an AVEC device, which was similar to AVMEC except that membrane was not involved. Under the best condition, an average cooling capacity of 179 W/M^2 was achieved in a period of four hours and cooling continued at a slowly declining rate for another four hours afterward. The temperature of water was maintained at approximately $12.5 \text{ }^\circ\text{C}$ after the pseudo steady state was established. Then, it was shown that the AVMEC cooling pads were able to provide a cooling capacity of 277.4 W/m^2 in a $37 \text{ }^\circ\text{C}$ ambient environment (incubator). The temperature of the cooling core surface was maintained in a range of $20 - 21.8 \text{ }^\circ\text{C}$ in the one-hour test period. No power supply was required except for the initialization stage, which took 5 minutes. Furthermore, human subject tests with or without wearing NWBC (Nuclear Warfare Biological and Chemical) suit demonstrated that, a AVMEC garment composed of 12 cooling pads were able to maintain the core body temperature of the subjects below $38.5 \text{ }^\circ\text{C}$ for up to 90 minutes while the subject was walking on a treadmill at a speed of 2 miles per hour in an environment of $40 \text{ }^\circ\text{C}$ and 50% RH (relative humidity). These results indicate that the AVMEC garment is a promising man-portable personal cooling technology.

RÉSUMÉ

Un vêtement de refroidissement portable fonctionnant par RAEMV (Refroidissement par Adsorption et Évaporation avec Membrane sous Vide) a démontré sa capacité à fournir suffisamment de refroidissement pour du personnel travaillant à des degrés d'activités moyens. Des études ont d'abord été conduites dans un dessiccateur sous vide bien contrôlé à température ambiante pour élucider les effets de certains paramètres clés sur la performance d'un appareil de RAEV, qui est similaire au RAEMV sauf qu'aucune membrane n'intervient. Sous la meilleure condition, une capacité de refroidissement moyenne de 179 W/M² a été réalisée dans une période de quatre heures et le refroidissement s'est poursuivi par la suite à un taux bas et déclinant pour un autre quatre heures. La température de l'eau a été maintenue à approximativement 12.5°C après que le pseudo régime permanent ait été établi. Ensuite, il a été montré que les sacoches de refroidissement par RAEMV étaient capables de fournir une capacité de refroidissement de 277.4 W/m² dans un environnement de 37°C (incubateur). La température de la surface du centre du refroidissement a été maintenue dans la plage de 20 – 21.8 °C pour une heure de test. Aucune alimentation électrique n'a été requise excepté pour la période d'initialisation qui a pris 5 minutes. De plus, des tests sur des sujets humains portant ou ne portant pas des uniformes NBC (Nucléaire Biologique et Chimique) ont démontré que, des vêtements de RAEMV composés de 12 sacoches de refroidissement étaient capables de maintenir la température interne du corps en bas de 38.5°C pour plus 100 minutes pendant que le sujet marchait sur un tapis roulant à une vitesse de 2 milles par heure (3.2 km/h) dans un environnement de 40°C avec 50% d'HR (humidité relative). Ces résultats indiquent que le vêtement de RAEMV est une technologie de refroidissement personnel portable pleine de promesses.

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CHAPTER 1

Introduction

1.1 Background

Heat stress management is being paid more and more attention for the protection of people who are working in hot, humid environments or wearing special impermeable protecting clothes. Heat stress could lead to heat-related injuries and deaths which are the most severe consequences of thermal imbalance in workplaces among workers. In fact, from a thermal balance standpoint, when workers are faced with hot environmental conditions, their thermoregulation mechanisms automatically set off internal steps to keep the core body temperature at 37°C (Flomenbaum, Goldfrank et al. 2006) and the skin temperature at comfortable ranges (around 33°C) (Nunneley 1970). The skin temperature and the core body temperature are regulated differently. While the former depends on blood flow beneath skin and environmental conditions, the latter is regulated by the brain. Consequently, an efficient auxiliary cooling method should address adequately both, skin temperature and core body temperature to provide effective heat mitigation (Nunneley 1970; Lim et al. 2008)

Another group of people who may benefit from cooling technologies are heat sensitive individuals including multiple sclerosis (MS) patients (60% to 80% of MS patients are heat sensitive) (Flensner and Lindencrona 1999; White et al. 2000; Edlich et al. 2004). For all these heat exposure cases, personal cooling has been reported to often be the best way to

meet the body's heat removal needs (Nunneley 1970; Chauhan 1999; Flensner and Lindencrona 1999)

In some situations such as mines, deserts, and other open environments, experts find it is better to cool people directly rather than cooling their entire surroundings; therefore, they developed the techniques for personal cooling (PC), also referred to as microclimate cooling (MC).

Three conventional techniques of PC have emerged on the basis of downsizing cooling systems that were originally developed for large scale space cooling or industrial process cooling. They are fluid cooled garments, FCGs (including water cooled garments and air cooled garments), phase change material (PCM) cooling garments, and evaporative cooling garments (ECG).

The most common fluid coolants of FCGs are water and air. They circulate or are distributed within the garment and absorb excess heat from the body. These devices are typically composed of efficient miniature chillers, conditioned units with fluid tubing and a circulation pump (for liquid cooled garments, LCG), or an air blower (for air cooled garments, ACG), and power supply. FCGs are currently recognized as the most efficient PC technologies, which can achieve good cooling capacities (Flouris and Cheung 2006). In the case of the water-cooled jacket, the observed temperature gradient between microclimate - jacket - ambient conditions provides 160 to 170 W of cooling per square metre of body surface area (Nag et al. 1998). Unfortunately, they are limited by lack of portability due to the requirement of heavy refrigeration units and continuous power supply. Consequently, users are restricted within a small space, e.g. inside a vehicle (Flouris and Cheung 2006).

PCM are gels or solids which will absorb excess heat from the body as latent heat of fusion when transferring from solid phase to liquid phase. Heat absorption occurring during this phase change process (melting) favours the maintenance of the body temperature at 37°C. Many PCMs have melting temperatures ranging from 4° to 40°C (Zalba et al. 2003), which makes them attractive cooling devices in terms of human comfort as the temperature range overlaps with the ideal environmental conditions for human being which is between 20° to 27°C, it is slightly lower than the skin temperature that has been determined to be 33°C on average for resting subjects (Nunneley 1970). Garments made with PCMs are simple, easy to use, require no energy supply, and are usually portable. The cooling capacities achieved with PCMs are based on the PCMs' latent heats of fusion, which are moderate compared to the heat of water evaporation, with the largest latent heats reported to be around 330 kJ/kg for PCMs having melting temperatures ranging from 4° to 40°C (Zalba et al. 2003). Standard commercial vests, which were reported to contain up to 5 lbs of PCMs, could cool for about two hours (Shapiro and Epstein 1983) at a capacity of up to 95 W, which was much lower than what was achieved with FCGs. They also needed to be recharged after usage (Shapiro and Epstein 1983; Zalba et al. 2003; Bendkowska et al. 2008).

ECGs are based on water evaporation. The water evaporates by absorbing heat dissipated from the body to help mitigate the heat stress of the wearer (Perez et al. 1994; Grzyll and Balderson 1997; Lee and Hwang 2006; Rothmaier et al. 2008). The large latent heat of water evaporation, which is around 2400 kJ/kg (Perry 1997), promises large cooling capacity with a mass of water that is portable to man. ECGs are portable, lightweight, offer comfortable range of operating temperatures, and do not require power supply for operation. The main drawback is that ECGs do not work in highly humid environment. Thus under hot humid

environmental conditions, where personal cooling could offer the largest benefits, ECGs become less reliable.

1.2 Conceptual design of the AVMEC garment

Since evaporative cooling aims to harness the latent heat of water evaporation for cooling, adding a desiccant which has a strong attraction for water vapor is expected to facilitate the water evaporation greatly. Water vapor will keep on being transferred towards the desiccant as long it is unsaturated (or until another limiting equilibrium reaction is reached in the overall operation). The concept of adsorption vacuum evaporation is similar to that used at large scale to produce cooling in absorption refrigeration (Liu et al. ; Dieckmann et al. 2008).

Introducing membrane technologies into the desired cooling technique is another improvement that theoretically can be made to enhance evaporative cooling. Recent studies have pointed out the great potential that membrane technology has for the achievement of effective temperature reduction in evaporative cooling settings (Johnson et al. 2003; Lee and Hwang 2006; Peinemann et al. 2006). Adding membrane to evaporative cooling devices has great advantages. Hydrophobic membrane can provide a barrier for liquid/vapor phase separation because it allows only vapor to permeate to the other side and retains liquid water. In such a way, evaporation and adsorption of vapor, which releases heat, could take place in different compartment within a small enclosure. This is important for microclimate cooling, where weight and size of the garment are important concerns.

Many membrane materials usually employed for membrane distillation can also be used in membrane evaporative cooling. Membrane distillation is a separation process used to separate the more volatile compounds from a liquid mixture (El-Bourawi et al. 2006). As for

membrane evaporative cooling, its purpose is to achieve heat removal using water's large latent heat of vaporization to provide the barrier for vapor/liquid phase separation (Johnson et al. 2003). The two processes are comparable in the fact that they are both thermally driven and use hydrophobic membrane materials (Johnson et al. 2003; Sidney 2003; El-Bourawi et al. 2006; Sanchez et al. 2007). It has been demonstrated by the previous researchers in our lab that for this application PTFE20 (Teflon with a 0.2 μ m pore size) was the best membrane out of many distillation membranes. Distillation membranes (DM) are proven hydrophobic membrane materials able to withstand vacuum, as they are used in vacuum membrane distillation (Curcio and Drioli 2005; El-Bourawi et al. 2006). They are thus good candidates for this study, which involved vacuum. The choice was made by examining two important properties: their pore size and liquid entry pressure (LEP). The LEP is defined as the trans-membrane pressure difference at which the liquid penetrates into the pores of the hydrophobic membrane (García-Payo et al. 2000). When the LEP is high, the membrane can withstand very high trans-membrane pressure differences before a drop of liquid water goes through the pores onto the other side of the membrane. Thus, high LEP values are vital for the optimal performance of the membrane evaporative cooling system.

Another important feature of AVMEC is that a vacuum initialization period at the very beginning of each run was involved to create a high degree vacuum between the cooling core and the adsorption core. Vacuum cooling is based on the rapid evaporation of moisture from the surface and within the objects due to the low surrounding pressure (Ozturk 2009). Vacuum has played an important role as it facilitated the evaporation process. Vacuum pump was applied to provide required vacuum degree, which was lower than the saturation vapor pressure for the liquid in the feed side at corresponding temperature to allow boiling. The

hydrophobic membrane would separate the remaining liquid water from the vapor allowing the vapor permeates to the other side. Dissipation of the latent heat of vaporization takes places at the same time, resulting in a cooler temperature of the remaining water. As water evaporation directly depends on the surrounding vapor pressure, when water is put in a control volume and the pressure inside is decreased, the vapor moves to the surrounding atmosphere. Thus, vacuum evaporation is another effective means to achieve cooling (Decker 1993; McDonald and Sun 2000; Bird et al. 2002; Dostal and Petera 2004; Da-Wen and Liyun 2006). In membrane distillation, for instance, vacuum is used for the very purpose of creating a low pressure on one side of the membrane to promote heat and mass transport towards the vacuumed side (El-Bourawi et al. 2006).

Since the vacuum pump and power supply is required only in the very short initialization stage (typical 5 minutes, see later results), the AVMC cooling garment could be practically considered an unpowered cooling mechanism that would render the much desirable portability to the cooling garment for a given operation period. On the other hand, when compared to conventional unpowered evaporative cooling methods such as the aforementioned three-layer laminate garments and the water-soaked cooling garment, AVMEC garments could provide much larger cooling capacities and are independent of ambient humidity because high degree vacuum is generated and maintained throughout the operation period, which would drastically increase the evaporation of water.

This cooling technique combination should provide very efficient cooling capacity based on the water latent heat of vaporization (2400 kJ/kg). The features of portability, light-weightness, comfortable range of temperatures, and good durability should remain. And now, more added benefits are: cooling independent of ambient humidity and temperature,

enhanced cooling capacity, and an anticipated affordable energy input tradeoff, compared to FCGs.

1.3 Objectives

The ultimate objective of this study is to design a personal-portable adsorption vacuum membrane evaporative cooling (AVMEC) garment that is capable of providing sufficient cooling and operates independent of ambient humidity. To accomplish this goal, the following studies were carried out:

- Concept validation:
 - Proof of the concept of AVEC using desiccator tests
 - Proof of concept of AVMEC using cooling pads
- Studying the effects of key parameters on cooling performance
- Prototyping
- Human subject tests

1.4 Structure of the thesis

The thesis is organized as follows: Chapter 1 is the introduction, which aims to provide the background, the objectives and the structure of this thesis work. Chapter 2 is the literature review, which provides the overview of the microclimate cooling starting with human thermoregulation, then the various existing PC technologies, the heat and mass transfer in vacuum evaporative cooling, and the adsorption vacuum evaporation. In chapter 3, the manuscript “Adsorption vacuum evaporative cooling” was presented. Following that, Chapter 4 is a manuscript entitled “Man-portable cooling garment based on adsorption

vacuum membrane evaporation”. Finally, in Chapter 5, conclusions and recommendations are presented.

CHAPTER 2

Literature review

2.1 Background

Some particular occupations have a special need of heat stress management. For example, firefighters face a trade-off between personal protection and thermal strain when performing firefighter activities. Protective clothing is required to shield the person from the excessive radiation heat. However, it at the same time prevents the dissipation of metabolic heat from the body. Those protective clothing eliminate the evaporative heat loss by decreasing water vapour permeability, thus the thermal balance of body may be upset to create a condition of heat-stress. In this case, trapped metabolic heat produced by working muscles, as well as heat gained from the environment, produce an increased heat stress. The heat stress can cause body core temperature to rise, leading to a series of disorders such as heat rash, heat cramps, heat exhaustion, fainting and heat stroke, some of these conditions are life-threatening. (McLellan 2006)

The operations that have a high potential for invoking heat stress usually involve high ambient temperatures, strong radiant heat sources, high humidity, direct physical contact with hot objects, or strenuous physical activities. Such scenarios most occur in work places such as iron and steel foundries, nonferrous foundries, brick-firing and ceramic plants, glass products facilities, rubber products factories, electrical utilities (particularly boiler rooms), bakeries, confectioneries, commercial kitchens, laundries, food canneries, chemical plants, mining sites, smelters, and steam tunnels. Outdoor operations conducted in hot weather, such

as construction, refining, asbestos removal, and hazardous waste site activities, especially those require workers to wear semi-permeable or impermeable protective clothing, are also likely to cause heat stress among exposed workers (McLellan 2006).

In the majority of cases, heat stress may be prevented, or at least reduced to a less risk level. Research has shown that personal conditioning systems, for example, personal cooling garments, are an economical and effective means of reducing heat stress and extending working times in thermally stressful environments (Allan 1988; Bishop et al. 1991). The earliest efforts in this field were carried out by Royal Aircraft Establishment in late 1950's and early 1960's on garments used for maintaining thermo-neutrality in pilots who were exposed to high temperature environment due to sunlit aircraft cockpit (Chauhan 1999).

2.2 Human thermoregulation

Human thermoregulation serves the dual function of controlling internal temperature and external heat dissipation (Lim et al. 2008). The overall requirement is to balance the heat input and output of the body. Humans maintain their core temperatures within a small range, between 36 and 38°C (Arens and Zhang 2006).

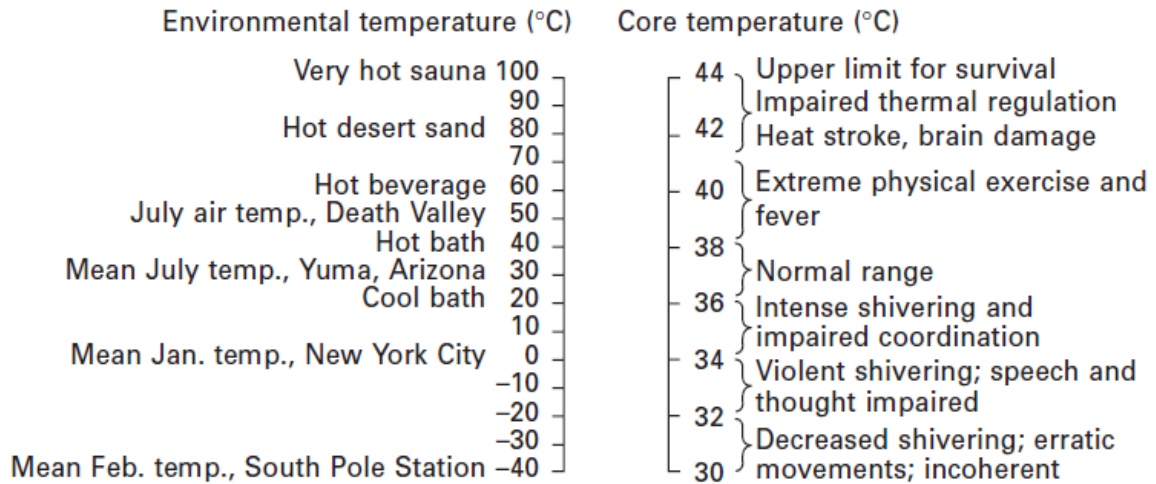


Figure 2.1 Ranges of environmental and human body temperatures (Arens and Zhang 2006)

The heat exchanges between the body and the environment can be carried out by means of radiation, convection, evaporation and conduction. Depending on the environment, radiation, convection or evaporation could be the primary means while conduction is usually negligible. Solar radiation and long-wave radiation from surfaces warmer than the skin temperature can warm the skin as a function of its color and surface emissivity. Although in most conditions convection and evaporation carry metabolic heat away from the body, hot winds may cause the skin to warm, when the body's sweat supply rate is insufficient to keep up with evaporation, and sensible gains exceed evaporative losses.

Heat exchange between human body and environment can be presented as the equation below (Nunneley 1970):

$$M - S = W + R + C_v + C_d + E \quad (2.1)$$

Where,

M - Metabolic heat production;

S - Storage (zero at equilibrium);

W – External work;

R – Radiation in the electromagnetic spectrum;

C_v – Convection by motion of fluid (air and water) surrounding the body;

C_d - Conduction by contact with a solid (usually negligible);

E – Evaporation, including obligatory and thermoregulatory component.

Metabolic heat production is determined by the energy needed for basic body processes plus any external work. Some comprehensive review and research have been conducted by Cheuvront and Gonzalez on human thermoregulation (Crandall et al. 2008). Human thermoregulation generally refers to four mechanisms: sweating, shivering, vasodilatation, and vasoconstriction. Sweating increases body heat loss by increasing sweat evaporation. Shivering produces heat by the involuntary movement of muscles. Vasodilatation and vasoconstriction refers to changes in blood vessel diameter, which affect skin temperature by changing the rate of blood exchange with the interior (Arens and Zhang 2006). The internal conductance governs the flow of heat from body core to periphery. In hot environment, increased conductance below the skin surface (due to increased blood flow) facilitates heat transfer from body interior to the skin. Then convection and evaporation of sweat carries the heat away from the surface of the body to the environment. In cold environments, muscle tensing and shivering increases heat production and body temperature. Decreased blood flow resulting decreased conductance below skin keeps the heat from escaping to the cold environment. These body heat control mechanisms are able to maintain human body core

temperature within a very small range in spite of variation in metabolic output that can exceed an order of magnitude above the base value, and similar variation in the heat loss rate from body to the environment.(Cheuvront, Gonzalez 1997)

2.2.1 Mechanisms of skin self-cooling

Skin is the major organ to control the heat input and output through the surrounding environment. The area of skin on the body can be estimated from the body's height and weight, using a relationship developed by DuBois in 1915 (Underwood and Ward 1966):

$$A_{skin} = 0.202M^{0.425}L^{0.725} \quad (2.2)$$

Where:

A_{skin}	The skin area in m ² ;
M	Subject's mass in kg;
L	Subject's height in m;

A 1.65 m person weighing 73 kg will have a skin surface area of 1.8 m², a commonly used figure for 'standard' men. The range of surface areas from school-age children through large adults is 0.8 through 2.4 m².

Skin functions through three mechanisms, including sensible heat transfer at the skin surface (via conduction, convection, and radiation), latent heat transfer (via moisture evaporating and diffusing through the skin, and through sweat evaporation on the surface), and sensible plus latent exchange via respiration from the lungs (Arens and Zhang 2006). Dripping of liquid sweat from the body or discharge of bodily fluids causes relatively small amounts of heat exchange, but exposure to rain and other liquids in the environment can cause high rates of

heat loss or gain. In case of heat stress, the body's first response is circulating blood to the skin, increasing skin temperature and allowing it to give off some heat. In heavy work, one's muscles require more blood flow thus reducing the amount of blood available for the skin to release heat.

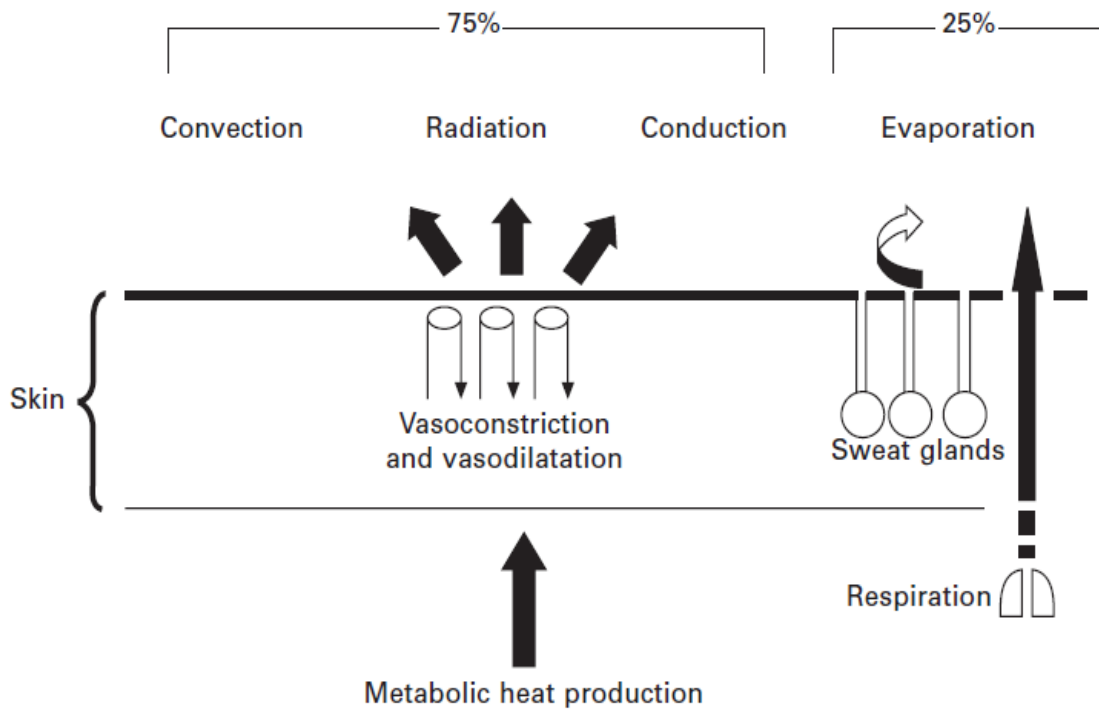


Figure 2.2 Heat transfer through and above the skin (Arens and Zhang 2006)

Figure 2.2 shows heat transfer above and below the skin surface. In a neutral environment, where the body does not need to take thermoregulatory action to preserve heat balance, evaporation provides about 25% of total heat loss, and sensible heat loss provides 75%. During exercise, these percentages could be reversed.

Sweating is an important means that helps one's body cool off, because the skin became warm when doing exercise or heavy load work resulting in the temperature gradient between

the skin surface and the environment to diminish, therefore sensible heat transfer becomes insufficient to remove the body's metabolic heat. Evaporation of body moisture is a highly efficient heat removal process, and complex physiological mechanisms are evolved to encourage evaporation under conditions of heat stress, and to minimize it otherwise, both to avoid overcooling and to minimize the water lost of the body. It should be mentioned, however, that high sweat rate may produce fluid and electrolyte imbalances and eventual collapse (Chauhan 1999). Therefore unless salts and water loss through sweating is replaced, as well as when levels of humidity are low enough to permit sweat to evaporate, evaporation cooling by sweating should not be encouraged. Respiration is the other means for the body to cool down, although it is not as important as sweating.

2.2.2 Environmental factors on human body cooling

The natural mechanisms for body heat management are limited, environmental factors also affect the extent of stress that a worker may face under a hot working area, such as humidity, temperature, wind speed, and radiant heat like that coming from a furnace or the sun. Reflective clothing, which can vary from aprons and jackets to suits that completely enclose the worker from neck to feet, is a technique utilising reflection of radiation heat as a heat-management mechanism to stop the skin from absorbing radiant heat. However, since most protective clothing does not allow air exchange through the garment, the benefits of radiant heat reduction may be offset by the loss in evaporative cooling.

2.3 Conventional personal cooling garments:

Different types of personal cooling garments have been developed and studied to conserve the comfortable microclimate, such as air cooled garments, water cooled garments, ice vest

and evaporative cooling vest with wetted media. Those different cooling garments concept, advantages and disadvantages are presented below:

2.3.1 Air cooled garments

Air cooled garments, by directing compressed air around the body from an air supplying system, allow the removal of heat by both evaporation of sweat and convection. These garments included miniature jets at the end of each of the distribution pipes, the purpose of these being to break up the boundary air layers and thus promote sweat evaporation. They also helped to equalize flow down each of the tubes which were of very different lengths (Allan 1988).

The garments can employ either ambient air or pre-cooled air to better fit their application. The use of air cooled devices under humid environments can keep the suit and body dry, avoiding the wetting caused by condensation when water-cooled devices are used (Epstein et al. 1986). However, air cooled devices have been considered inferior to water or ice-cooled garments because of the low heat capacity of air (Constable et al. 1994). It needs continuous power supply, and the entire garment is relatively heavy and bulky due to the packed tubing built in the garment. As a result, air cooling garments are not portable. Meanwhile, the air cooled garments were found inadequate for many applications, notably for hot industrial trades and for space suits, owing to the large volume of air required for cooling to achieve the ideal condition at high metabolic rates or extreme environment (Nunneley 1970; P.K.Nag 1998). The greatest advantage occurs when circulating air is that it employs the natural heat dissipating mechanism (evaporation of sweat).

2.3.2 Liquid cooled garments (LCG)

The possibility of using water as a cooling medium for personal cooling was first mentioned by Dr John Billingham in 1959 (Allan 1988). However, it was not until several years later that DesBurton who was working at Royal Aircraft Establishment developed a functioning liquid-cooled suit (Nunneley 1970). The suit presented a prototype water cooled garment (WCG) built of 40 polyvinyl chloride tubes threaded in to a suit of cotton underwear. Water was piped to the ankles and wrists while manifolds at mid-thorax. The suit was comfortable even when high heat loads necessitated low water temperatures and despite the existence of wide skin temperature differences when comparing sites directly beneath cooling tubes with sites lying between tubes (Nunneley 1970).

Liquid cooled garments range from a hood, which cools the head only, to vests and "long johns," which offer partial or complete body cooling (Bishop et al. 1991). Use of this equipment requires a battery-driven circulating pump, liquid-ice coolant, and a container (Cadarette et al. 2006). The national aeronautics and space administration (NASA) from United States also developed similar garments. More recently, it has been shown that water-cooled garments are the most effective personal cooling devices in commercial use at present (Allan 1988).

The efficacy of water cooled garments depends on the rate, temperature and heat capacity of the circulating liquid (water with 10% ethylene glycol), the length and insulation of the connecting tubes, and the thermal conductivity of the tube wall. (Nag 1998). The effective use time of the water-cooled garment is limited by the amount of water available in the reservoir and by the build-up of inlet water temperature. The garment with one-half litre internal container was sufficient for this purpose. When the outer layer of the double-layered

water reservoir was packed with ice, the circulating water temperature was maintained in the range of 15 to 18°C for ambient dry-bulb temperature of 40°C over a 2-h period (Nag et al. 1998).

In the case of the water-cooled jacket, the observed temperature gradient between microclimate - jacket - ambient conditions provides 160 to 170 W of cooling per square metre of body surface area. This level of cooling is much higher than required for the individuals under light physical activity and is highly beneficial to reduce the heat load for individuals engaged in moderate physical activity in heat (Nag et al. 1998).

Theoretical comparison of air and water cooled garments showed that water's high heat capacity conferred marked engineering advantages in decreased pumping power, lowered system weight, and less garment bulk (Nunneley 1970). Also, the WCG can easily be combined with other protective clothing.

However, practical problems still remain in the design and operation of WCG. The full suits are expensive and must fit the wearer closely with ideally no material intervening between the tubing and skin. Suit reliability is another problem under hard daily use where the water spill could have serious consequences including wet clothing, electrical short circuits, or steam production causing burns. Practically, methods of cooling control require further development (Allan 1988).

2.3.3 Phase changing material

Phase change materials are solids that melt in a temperature range of 0 to 30°C, making them good candidates for personal cooling. Natural PCMs, such as ice or dry ice (solid CO₂) have been used since the 1960's (Chauhan 1999). These garments remove heat by latent heat of

sublimation or latent heat of melting. The heat exchange between the skin and PCM is thus primarily due to conduction.

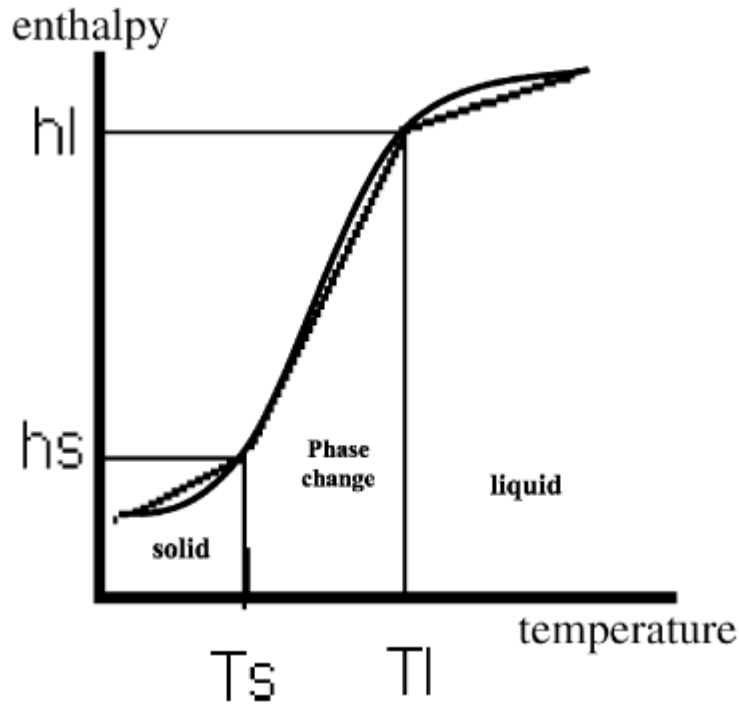


Figure 2.5 Enthalpy variation with temperature (Zalba et al. 2003)

Dry ice removes heat by sublimation. When dry ice sublimates the latent heat of sublimation has a cooling effect which removes 573 kJ/kg of heat energy. However, its disadvantage is that it sublimates at -78°C , and therefore, can cause "cold burns" to the skin. Also need to mention is that it is expensive, not readily available and can be poisonous if CO_2 builds up in an enclosed area (Chauhan 1999).

Ice vests are the most available PCM cooling garment in the market. The cooling offered by ice packets can last 2 to 4 hours at moderate to heavy heat loads, and frequent replacement is necessary. However, ice vests do not hinder the worker and thus permit maximum mobility.

Cooling with ice is also relatively inexpensive, but they are fairly heavy. The latent heat of ice, although the largest among that of PCMs, is only 334 kJ/kg at 0°C (Zalba et al. 2003). Consequently, they can only provide moderate cooling capacity in a limited operation time. Although this system has the advantage of allowing wearer mobility, the weight of the components limits the amount of ice that can be carried and thus reduces the effective use time. The heat transfer rate in liquid cooling systems may limit their use to low-activity jobs. Even in such jobs, their service time is only about 20 minutes per pound of cooling ice. To keep outside heat from melting the ice, an outer insulating jacket should be an integral part of these systems. The garments with ice packets or solid carbon dioxide in plastic packets secured to the vest have some advantages for short duration emergency exposures to high heat.

When the PCM has a fusion temperature that is much lower than the body temperature (e.g. ice), thermal shock may be incurred due to sudden drop in body temperature. Consequently, care should be taken in choosing appropriate PCMs with adequate operating temperatures (Nunneley 1970; Zalba et al. 2003).

2.3.4 Evaporative cooling garment

As it has been realized that sweat evaporation involved in air cooled garments can effectively increase the cooling performance, the water soaked garments has also been investigated by Technique International and other organizations. Rothmaier et al. (2008) proposed the design with concept of three-layer laminates, which employed the membrane technology and was based on body heat driven evaporation of water out of a skin-contacting reservoir.

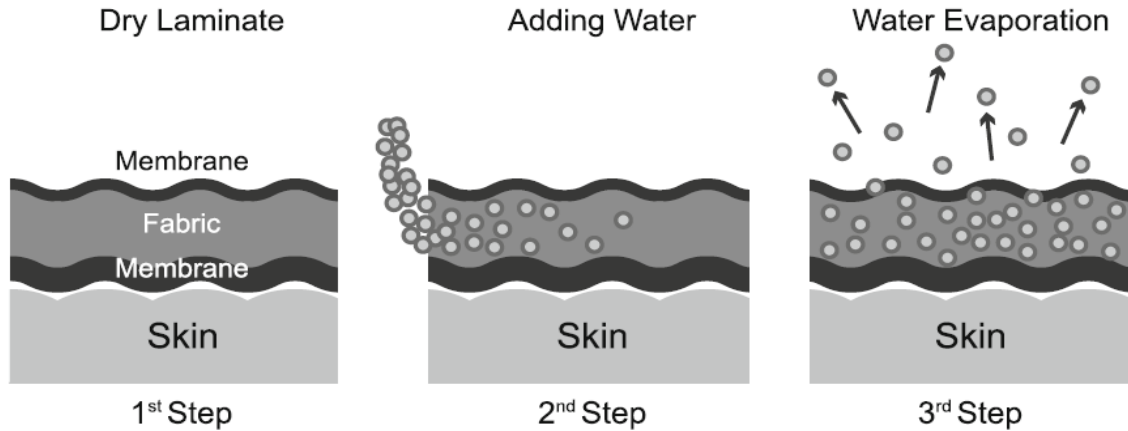


Figure 2.6 Diagram of the three-layer laminate and the principle of water evaporation from reservoir

Figure 2.6 shows the principle of evaporation of this concept. The garment was designed to directly contact with the wearer's skin (gaps of air between skin and membrane would lead to reduced cooling efficiency due to lower thermal conductivity). The water between the layers diffuses as vapour through the outer membrane. It removes latent heat required for evaporation, therefore, providing cooling effect to the underlying skin and tissue.

The great advantage of this type of cooling garment is that it is light and flexible, and therefore can be portable.

2.4 Evaporative cooling systems

Evaporative cooling, which takes advantage of the large latent heat of water evaporation (approximately 2400 kJ/kg at 25 °C), is an energy-saving and environmental friendly way to cool down the surrounding temperature (Wu et al. 2009). It has been investigated for a long time, and been applied to different areas, such as air conditioning (Yasu Tai and Bomalaski 1993; Hajidavalloo 2007; Alizadeh 2008), evaporative condenser for residential refrigerator

(Nasr and Hassan 2009), cooling towers, and agricultural and food products processing and storage (Olosunde et al. 2009; Ozturk 2009).

2.4.1 Evaporative cooler

The typical applications for evaporative cooling systems are cooling towers (Pascal Stabat 2003; Bhattacharya, Mondal et al. 2010), in which water is cooled as a result of sensible heat exchange with the air and latent heat transfer due to evaporation of the water; air conditioning system for buildings (Hajidavalloo 2007; wang 2009; Liu et al. 2010), evaporative cooling which can be driven by low-grade heat source such as solar energy and waste heat with relatively low cost has been studied as the world weather tend to have a long-term climb due to increasing “greenhouse effect” and “ozone-layer” damage (E.bakaya-Kyahurwa). There exist various methods of evaporative cooling, including direct evaporative cooling (DEC) and indirect evaporative cooling (IEC), both of which rely on re-circulating water through a wetting media by spraying or allowing water to trickle through the media. DEC configurations allow water to come into direct contact with the air to be cooled, while IEC configurations utilize a heat exchanger with cooled air from a DEC unit.

Evaporative condensers for vapor compressor refrigeration systems (Hajidavalloo 2007; Nasr and Hassan 2009), in order to improve the performance of window-air-conditioners in hot weather (about 40-50°C), the technique of employing evaporative cooling pads in the condenser of window-air-conditioners to enhance heat transfer rate in the condenser has been developed.

The advantage of evaporative cooling is that it increases efficiency compared to conventional air-conditioning. However, the disadvantages of evaporative cooling include water

consumption and the potential for microbial growth due to a supply of stagnant water that is in contact with air.

2.4.2 Vacuum evaporative cooler

Vacuum cooling is based on the rapid evaporation of moisture from the surface and within the objects due to the low surrounding pressure (Ozturk 2009). In an area of less pressure, evaporation happens faster because there is less exertion on the surface keeping the molecules from launching themselves. As water evaporation directly depends on the surrounding vapor pressure, when water is put in a control volume and the pressure inside is decreased, the vapor moves to the surrounding atmosphere. And vacuum cooling, as any other evaporative cooling technologies depend on latent heat of evaporation to remove the sensible heat of target objects (Ozturk 2009; Ramon et al. 2009). It can make evaporative cooling more efficient.

Traditionally, vacuum cooling has been used in food processing industry for pre-cooling of vegetables. Vacuum cooling of fresh product has been around since the 1950's. It has also been used industrially to make food products such as evaporated milk for milk chocolate, and tomato paste for ketchup (Da-Wen 2004; Olosunde et al. 2009). An increase in vacuum results in lowering the boiling point of the moisture, causing rapid evaporation. As the water evaporates, it absorbs the heat from the product and the temperature of the product is reduced.

2.4.3 Membrane evaporative cooler

Membrane evaporative cooling is similar to other membrane technologies including membrane distillation (Curcio and Drioli 2005) and membrane

humidification/dehumidification (Charles and Johnson 2008). As water evaporates from the membrane surface, energy is extracted from the water as the heat of vaporization which is then replaced by heat from the incoming air. The driving force of membrane evaporation is the difference between the partial vapour pressure at liquid/vapour interface in membrane pores and that at the membrane surface facing the environment. By employing one kind of hydrophobic membrane to let water vapour permeates through the tiny pores and separate from water side. Under controlled conditions, evaporation of water takes place at the liquid-vapor interface in membrane pores with cooled water staying on the liquid side of the membrane while letting vapor escape into the surrounding environment.

Most conventional evaporative cooling systems involve collection and recirculation of water to keep the wetting media or misting region saturated, which includes water directly contact with outside air. To prevent bacterial growth resulting Legionnaire's disease, Pontiac Fever and humidity fever, etc (Johnson et al. 2003), it usually require maintenance at significant costs including frequent inspections, cleaning, and addition of anti-bacterial agents in order to prevent against such diseases. To avoid these inconveniences, the indirect contact evaporative cooler employing hollow fibre membrane has been developed. Instead of conventional wetting media in direct contact evaporative cooler, hollow fiber membranes lets water-air interaction occur through membrane pores of a size ($<0.1\mu\text{m}$) too small to allow the microbes and bacteria to pass, meanwhile allowing water vapour to transfer. Another advantage for membrane evaporative cooling is that there is a large surface area per unit volume that facilitates heat and mass transfer therefore decreasing water consumption. The area packing densities for hollow fivers range from 2000 to 30,000 m^2/m^3 (Johnson et al. 2003; Charles and Johnson 2008). Furthermore, membrane evaporative cooler can

potentially require no recirculation pumps as feed water is replenished at the rate of uptake under required minimal pressure.

2.4.4 Evaporative equilibrium

If evaporation takes place in a closed vessel, the escaped molecules accumulated as a vapor above the liquid. Some of the water molecules return to the liquid phase, it happens more frequently as the density and pressure of the vapor increase. When the overall movement of water molecule escaping and returning to liquid equals each other, it reaches the equilibrium status.

The boiling point corresponds to the temperature at which the liquid's vapor pressure equals the surrounding environmental pressure. If the heat of vaporization and the vapor pressure of a liquid at a certain temperature are known, the vapor pressure at the desired temperature can be calculated by using the Clausius-Clapeyron equation, which is generated for the equilibrium state of a system consisting of vapor and liquid of a pure substance:

$$P_2 = P_1 e^{-\frac{\Delta H_{vap}}{R} \left(\frac{1}{T_2} - \frac{1}{T_1} \right)} \quad (2.3)$$

Where:

T Temperature, K

R The universal gas constant, $8.314 \text{ J}\cdot\text{K}^{-1} \cdot \text{mol}^{-1}$

P Vapour pressure at the corresponding temperature, atm

ΔH_{vap} The heat of vaporization of the liquid, J/mol

So according to the equation, with a reference temperature and boiling point (for example water's normal boiling point is 100 degree Celsius at 1 atm), the boiling point temperature will decrease at lower system pressure.

2.4.5 Mass and Heat transfer in vacuum membrane evaporation

Membrane distillation (MD), which is a separation process but has similar mass and heat transfer behaviors as a membrane evaporative cooling device, has been well studied. It usually refers to a thermally driven transport of water vapor through a porous hydrophobic membrane, while mass and heat transfer in vacuum membrane for cooling purposes has rarely been investigated.

The vacuum membrane distillation (VMD) process is that a feed solution is in contact with one side of a micro porous membrane and vacuum is pulled on the opposite side. The model describing VMD has been widely investigated and its ability to represent the separation process has been already well proven and presented. In VMD processes mass transport and heat transport are always happened at the same time. The driving force for mass transfer in VMD systems is the difference in the partial pressure of water vapor across micro porous hydrophobic membrane. The recognized transport mechanisms for mass transfer across the membrane are usually molecular diffusion and Knudsen diffusion and, sometimes, Laminar flow. Molecular diffusion has a partial pressure difference as driving force and non-identical molecules that are in the way form the resistance to mass transfer. The driving force for Knudsen diffusion is also a partial pressure difference, but in this case molecules bounces into the membrane matrix, which form the resistance to mass transfer. Knudsen diffusion is thus important for small pores and / or low pressure. Finally, viscous flow has a total

pressure difference as driving force, and the membrane matrix forms the resistance against it. In a VMD configuration, the molecular diffusion is not adequate due to the very low value of the partial pressure of the air inside the pores. Consequently, the Knudsen and viscous flow diffusion should be chosen as more appropriated.

In the VMD process Knudsen diffusion is dominant with respect to molecular diffusion. Thus, the flux of a permeating species, N is linearly related to its partial pressure difference across the membrane, can be expressed as below,

$$N = K_m \Delta P_i = K_m (P_v - P_o) \quad (2.4)$$

Where,

K_m Permeability coefficient (Bandini, Saavedra et al. 1997), depending on temperature as well as on some geometric characteristics of the membrane

ΔP_i Partial pressure difference across the membrane

P_v Water vapor pressure at the membrane surface at temperature T_w

P_o The pressure on the vacuum side

As it is well known, the water vapor pressure at liquid – vapor interface (in Pa) may be related with the temperature (in K), by using the Antoine's equation,

$$P_v = \exp\left(23.1964 - \frac{3816.44}{-46.13 + T}\right) \quad (2.5)$$

2.5 Adsorption facilitated evaporation process

As evaporation happened at the surface of water, the vapors need to be removed to keep a continuous vapor flux. While a vacuum pump was used to remove vapor for some cooling chamber (Ozturk 2009), vapor condensation is another alternative for cooling (Hajidavalloo 2007; Olosunde et al. 2009). But for designing a personal cooling garment, lightweight and portability are important considerations. In such a scenario, employing desiccant for vapor removal, which does not require a continuous power supply, could be a good choice.

A desiccant is a material having very strong affinity for moisture (Lee and Hwang 2006; Liu et al. 2010). It can help evaporative cooling being more effective. Desiccants are a type of sorbent having large affinity for water and have been extensively used for dehumidification or drying in air processing applications. Depending upon adsorbent and adsorbate phases, adsorption systems may be classified as solid/gas and solid/liquid. Heat of adsorption are either derived from adsorption isotherms, or determined experimentally using the calorimetric method, referred to as differential heat of adsorption.

2.5.1 Heat of vapour adsorption

Adsorption is a surface phenomenon occurring at the interface of two phases, in which cohesive forces including Van der Waals forces and hydrogen bonding that act between the molecules of all substances irrespective of their state of aggregation. Surface forces or unbalanced forces at the phase boundary cause changes in the concentration of molecules at the solid/liquid interface. All involve evolution of heat of adsorption. The heat of adsorption is usually small in physisorption processes and large in chemisorption. Adsorbents can be restored to original conditions by a desorption process usually involving the application of

heat, except in some cases chemisorption processes may be irreversible. The general term “sorption” is used when both adsorption and absorption occurs simultaneously (Srivastava and Eames 1998).

2.5.2 Desiccant

There are a number of research on thermally driven air conditioner that uses liquid desiccants as working fluid (absorbent). The choice of desiccant will have a profound effect on the design. Halide salts such as lithium chloride and lithium bromide are strong desiccant as they are extremely hygroscopic, Andrew Lowenstein has reported a good review which reported that a saturated solution of lithium bromide can dry air to 6% relative humidity and lithium chloride to 11% (Lowenstein 2008).

Among the possible halide salts that could be used as a desiccant, lithium chloride has by far been the most widely applied. However, lithium bromide is almost exclusively used in absorption chillers that use water as the refrigerant. It is shown that the bromide ion in the solution is more easily ionized than is the chlorine ion. In slightly acidified solutions, the bromide ion can be oxidized to bromine, which even in trace amounts can cause odor problems. Those doing early work with lithium bromide have encountered this problem. However, it may be possible to control the pH of the liquid desiccant to avoid this problem (Lowenstein 2008).

LiCl could be a good candidate material since it has good desiccant characteristics and does not vaporize in air at ambient conditions. However, it has the disadvantage of being corrosive (Fumo and Goswami 2002). Lithium Chloride is another type of absorbent. When water is absorbed on this material it changes to a hydrated state.

Regarding the choice of desiccant used in the systems, another alternative is Calcium Chloride (CaCl_2), which is considerably less expensive than LiCl. However, CaCl_2 has a lower crystallization mass fraction, which consequently leads to a lower water vapour pressure in the desiccant inside the absorber, reducing its capacity to attract water.

2.5.3 Water vapour pressure at the surface of LiCl

Many researchers have measured water vapour pressures over aqueous saturated salt solutions (Gokcen 1951). It depends on the temperature, the nature of the solute and its concentration, lists of vapour pressure at the surface of saturated LiCl solutions at different temperatures shown in Table 2.1.

Table 2.1 Vapor pressure at the surface of saturated LiCl solutions at different temperatures (Gokcen 1951)

Temperature (°C)	Average vapor pressure (mm Hg)
23.90	2.63
29.90	2.93
34.90	5.32
39.90	7.26
44.90	9.82
54.84	16.70

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CHAPTER 3

Adsorption vacuum evaporative cooling

Abstract

An adsorption vacuum evaporative cooling (AVEC) system, which takes advantages of vacuum evaporative cooling and vapor adsorption of adsorbent, has been demonstrated to be able to provide sufficient cooling capacity for specialty application such as personal microclimate cooling. The effects of several key parameters affecting the performance of an AVEC system including vacuum degree, continuous powered vacuum, amount of desiccants, evaporative/adsorption area ratio, and a reflective foil on top of water container for the purpose of minimizing internal heat transfer were investigated. Under typical operating condition with 100 g water, 30 g LiCl powder at a vacuum pressure of 28.8 In Hg at room temperature, a cooling capacity of 179 W/m² was achieved with water temperature maintained at 12–15 °C for at least 4 hours.

Keywords: evaporative cooling, adsorption, vacuum membrane cooling.

3.1 Introduction

Evaporative cooling, in which the latent heat of evaporation of a liquid (most commonly water) is harnessed to absorb heat from an object, has attracted great interests due to its unique advantages such as energy saving, cost-effectiveness and large cooling capacity. The most common current applications of evaporative cooling are cooling towers (Pascal Stabat 2003; Bhattacharyal et al. 2010). In a cooling tower, hot water is sprayed from the top of the tower downward to form small droplets that provides large surface for water evaporation and ambient air is blown from the bottom upward to carry the vapor away from water surface to maintain the driving force for evaporation. The relatively cold air stream also takes away heat as sensible heat. Cooling towers have been widely used in commercial refrigeration and other industrial process cooling water recycling (Hajidavalloo 2007; wang 2009; Liu et al. 2010).

Evaporative cooling have also been used directly in devices such as evaporative condensers for vapour-compression refrigeration systems (Hajidavalloo 2007; Nasr and Hassan 2009). In these studies, evaporative cooling pads were employed to enhance the efficiency of condensers of window-air-conditioners (Hajidavalloo 2007; Nasr and Hassan 2009).

Vacuum evaporative cooling, which is more commonly referred as vacuum cooling, is mostly used in food process industry (Da-Wen 2004; Olosunde et al. 2009; Ozturk 2009). It is a rapid cooling technology that achieves cooling through fast evaporation of the moisture on wet surfaces such as meat and vegetables via the application of vacuum, which lowers the boiling point of water and causes rapid evaporation. Vacuum cooling of fresh product has been around since the 1950's. It has also be used industrially to make such food products as

evaporated milk for chocolate milk, and tomato paste for ketchup (Da-Wen 2004; Olosunde et al. 2009). An increase in vacuum results in lowering the boiling point of the moisture, causing rapid evaporation.

Incorporating of liquid desiccant in evaporative cooling systems has also been proposed in recent years (Conde 2004; Lee and Hwang 2006). It has been getting more and more attention because of its unique advantages. First of all, it has great ability to capture moisture in the air or inside a device at a temperature higher than its dew point. Also, it only requires low grade energy for regeneration, thus incurring lower operation costs. And finally, it normally does not require compressor, condenser, chiller coils or other heavily insulated piping, which can remarkably save the maintenance costs.

All the aforementioned evaporative cooling technologies require continuous power supply, relatively heavy equipment, and large footprint and are therefore not suitable for specialty applications such as microclimate cooling where portability is a major concern (Nunneley 1970; Konz 1984; Chauhan 1999). For these applications, passive evaporative cooling that involve no mobile parts and requires no power supply is an attractive alternative. For instance, water soaked garments were investigated by Technique International and other organizations (Perez et al. 1994). They, however, could provide a very small cooling capacity due to the limited water evaporation and the small evaporation rate. Based on the same passive evaporative cooling concept, Rothmaier et al. (Rothmaier et al. 2008) proposed a three-layer laminated garment, which allowed water vapor to pass through but is water-proof at the outside layer to allow cooling by sweat evaporation. However, this type of passive evaporative cooling depends heavily on the ambient temperature and humidity. It is

very inefficient and unreliable in a hot and humid environment where personal cooling is needed the most.

In this work, we developed an innovative vacuum evaporative cooling technology that is enabled by continuous water vapour adsorption of desiccant rather than vacuum pump. Experiments have shown that this approach could provide four hours or longer continuous cooling without any power supply except in the initialization stage at the beginning. The effects of key parameters such as vacuum degree, desiccant/water ratio, adsorption/evaporation surface area ratio, and inclusion of a perforated reflective foil were also investigated. The knowledge we gained in this study provided valuable guidelines for the design and prototyping of functional cooling pads and cooling garments, which will be discussed in a separate paper (see Chapter 4).

3.2 Experimental setup and procedures

3.2.1 Experimental apparatus

A schematic diagram of the experimental apparatus used in this study is shown in Figure 1. The core of this apparatus is a vacuum desiccator (Pyrex 3120, Cole-Parmer), which is connected to a vacuum pump (WZ-07061-11, Cole-Parmer) that could generate maximum vacuum of 29 In Hg. The pump can deliver a free air capacity of 32.5L/min with an 115V AC power requirement. The desiccator was a 3800 ml vessel made of borosilicate glass (Pyrex) with a conic cover that ends in a cylindrical form where a Pyrex rugged stopcock is installed. The stopcock had a tube connection of 10 mm OD (outside diameter), which

allowed thick vacuum tubing to connect desiccator and pump. To maintain a tight contact between the detachable parts of the vacuum desiccator during an operation, vacuum grease was put on the edges of the vessel; and also on the stopcock and the cylindrical top part of the cover, opposite from each other. This helped ensuring a perfect vacuum inside the desiccator.

A glass vacuum valve was installed on the tubing connecting the vacuum pump and the desiccators. The desiccator was divided into two compartments, the upper space and the lower space, by a perforated plate of 19 cm diameter. The distance between the bottom of the desiccator and the lower edge of the perforated plate is 7.5 cm. In experiments, three 8.8 cm (ID) petri dishes (Fisherbrand) holding desiccant were located on top of the perforated plate and a 15.2 cm (ID) reusable glass petri dish (PYREX), which was used to hold water was located at the bottom of the desiccator. The distance between the water surface and the desiccant surface was 8.5 cm. The picture of the apparatus is shown in Figure 3.1 (b).

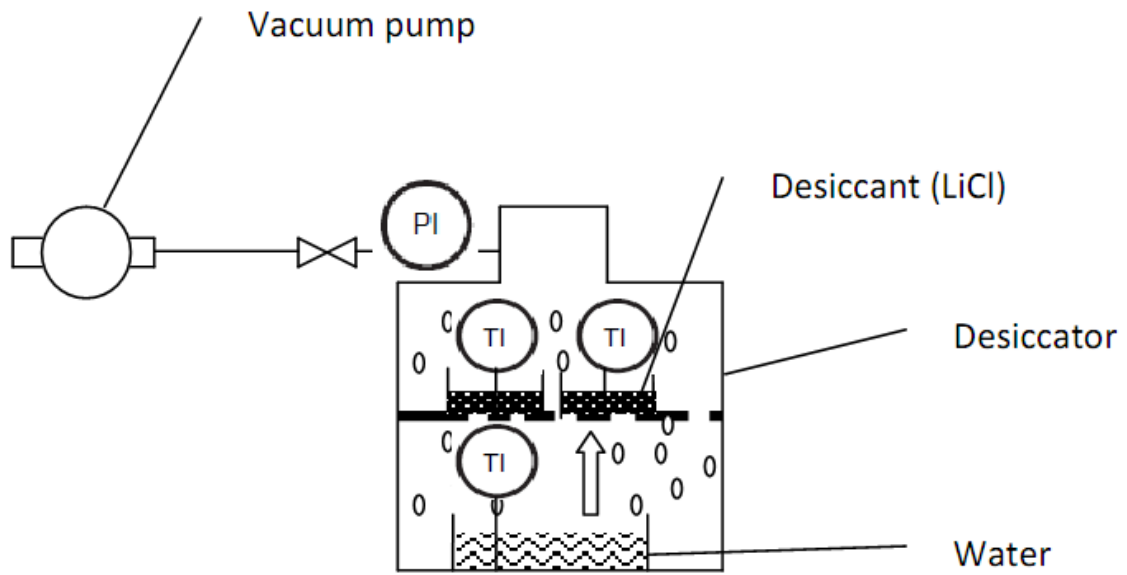


Figure 3.1 (a) A schematic diagram of the vacuum desiccators for adsorption vacuum evaporative cooling tests



Figure 3.1 (b) the vacuum desiccator systems for the adsorption vacuum evaporative cooling (AVEC) tests

3.2.2 Experimental procedures

In experiments, appropriate amount of water as specified in the text was put in the water dish, which was then put on the bottom of the desiccator. Then, the perforated plate was put in place and the three desiccant dishes were put on the perforated plate in such a way that they evenly occupied the upper surface of the plate. Desiccants of appropriate amount as specified in the text was weighed and evenly distributed on to the bottom of the three dishes right before the start of experiments. Four Oakton water-resistant pocket thermometer (WZ-90003-00, Cole-Parmer), one dish each, were then put vertically in the dishes in such a way that the sensor was either buried in the desiccants or submerged in the water. When everything was in place, the cover of the desiccator, which was connected to the vacuum pump, was put back to seal the desiccator. The vacuum pump was then turned on, which might remain on for the entire course of experiment or be turned off when a required vacuum pressure was reached, depending on the requirement of an individual experiment as specified. In addition, the water dish was completely covered using a piece of perforated aluminum film to reflect radiation away from the water dish in specified in the text. The aluminum film was perforated using a punching tool, with needles with a diameter of approximately 2 mm. The density of the holes was approximately 5 holes per square centimetre.

3.3 Results

3.3.1 The temperature profiles of a typical experiment

The temperature profiles of the water and the desiccant layer in a typical AVEC experiment are shown in Figure 3.2. The experiment can be divided into three stages, the initialization stage, the transit stage, and the pseudo-steady state stage. In the initialization stage, which took approximately 5 minutes, the desiccator was initialized using a vacuum pump to the required pressure of 28.8 in Hg. As shown in Figure 3.2, the water temperature decreased from the starting temperature of 23 °C to a much lower temperature of 21.2 °C in 5 minutes. The fast temperature drop of water was caused by water evaporation due to the combined effects of vacuuming and vapor adsorption. The vacuum pump was cut off when the vacuum degree reached 28.8 In Hg from the desiccators by turning off the valve and the experiment entered the transit stage. As shown in Figure 3.2, the water temperature continued to decrease from 21.2 °C to 15.3 °C in the transit stage. At this point, the experiment reached the pseudo steady state stage where the temperature of the water layer stabilized at 15.3 °C. During this pseudo steady state, for water layer, the heat gained from environment and desiccant layer were equal to the heat lost by water evaporation process.

The desiccant layer had a very different temperature profile than that of the water layer. In the approximately 5-minute initialization stage, the temperature of the desiccant layer increased sharply from the starting temperature of 23 °C to 26 °C. The fast temperature increase of the desiccant layer indicates that substantial amount of vapor was absorbed by the desiccant, which released adsorption heat. In the transit stage, the temperature of the desiccant layer continued to increase until it reached 29.7 °C and then decreased slowly. In

the pseudo steady stage, the desiccant temperature continued to decrease slowly, which indicate that the heat lost to the environment and water layer were larger than the heat gained from the adsorption process at this stage.

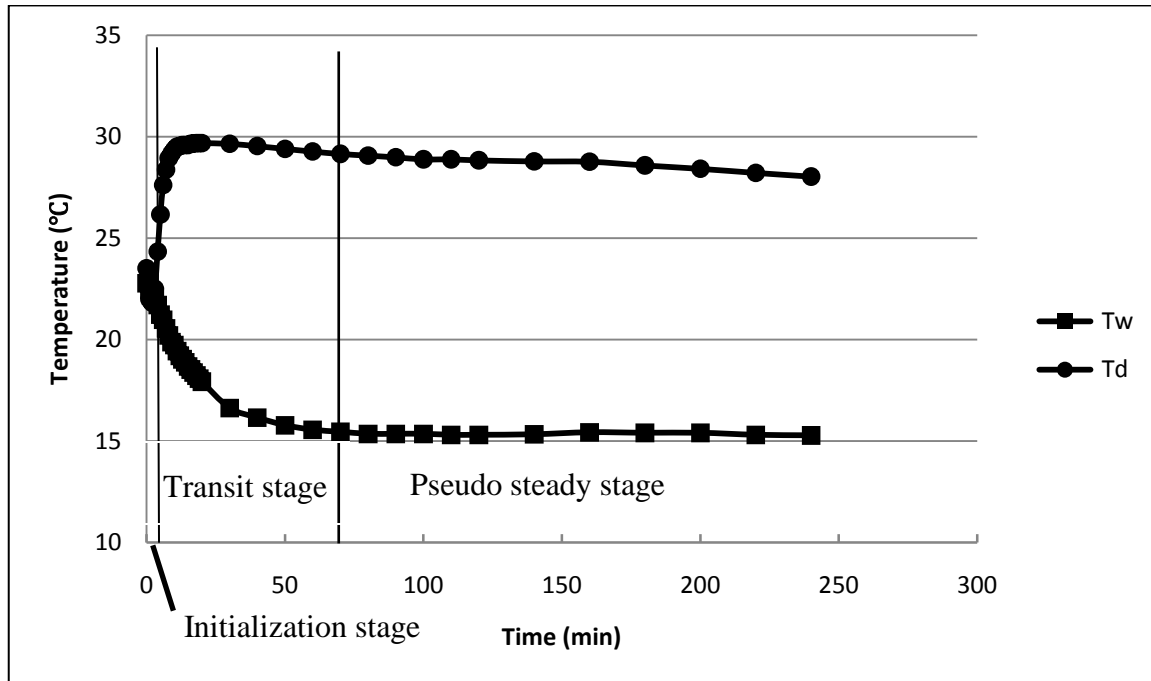


Figure 3.2 Temperature profiles of water (T_w) and desiccant layer (T_d) in a typical adsorption vacuum evaporative cooling (AVEC) experiment carried out with 100 g water, 30 g LiCl powder, at a vacuum pressure of 28.8 inHg at room temperature (Vacuum pump was switched off after the 5-minute initialization period)

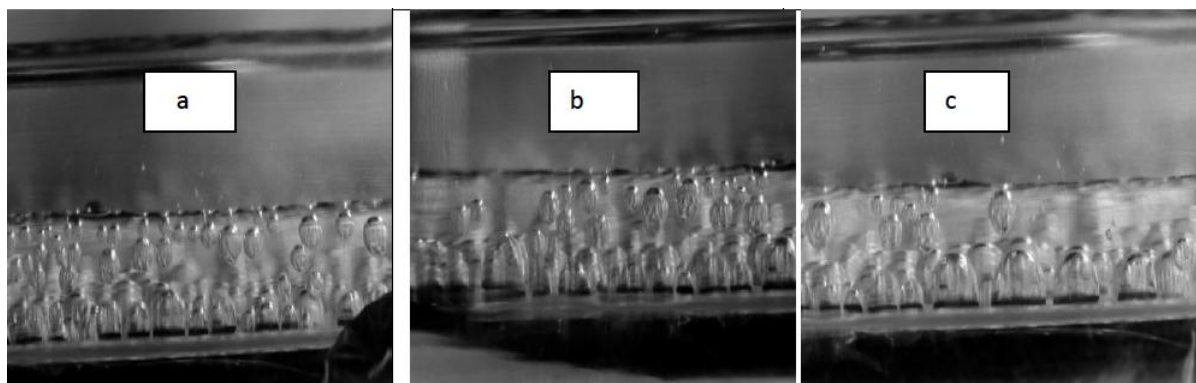


Figure 3.3 Water bubbling at different stages in a typical AVEC experiment carried out with 100 g water, 30 g LiCl powder, vacuum pressure of 28.8 in Hg at room temperature (23-24°C): (a) initialization stage (picture taken at 5 minutes); (b) transit stage (picture taken at 100 minutes) and (c) pseudo steady state stage (picture taken at 240 minutes)

Figure 3.3 shows the bubbling of water in the three stages. Strong bubbling was observed in the initialization stage, which could be attributed to the combined effects of degassing and water evaporation. Bubbling continued into the transition and pseudo steady state stage, however, at much less intensity. It was observed that in the later part of the transit stage and in the pseudo steady state, most bubbles occurred only at the bottom of the dishes but disappeared before they reached the water surface. This phenomenon could be tentatively explained by the hypothesis that the water temperature at the bottom of the dish was higher than that at the surface due to the conductive heat transfer between the water at the bottom with the ambience through the wall of the desiccator and that of the petri dish which held the water.

It is worth noting that the physical state of desiccant layer changed with the adsorption of water vapour process. It was solid powder at the initialization stage, and turned to solid/solution two-phase mixture during the transit stage and aqueous solution in the pseudo steady stage.

3.3.2 Influence of desiccant amount on water evaporation kinetics

Different amounts of solid desiccant have been tested while keeping the same amount of water of 100g for evaporation, vacuum degree of 28.8 In Hg, and initial water and desiccant temperature of around 25 °C. Five different desiccant masses were tested: 2.5g, 5g, 10g, 30g and 50g of LiCl at the same evaporation surface of 181cm² and adsorption surface of 182 cm².

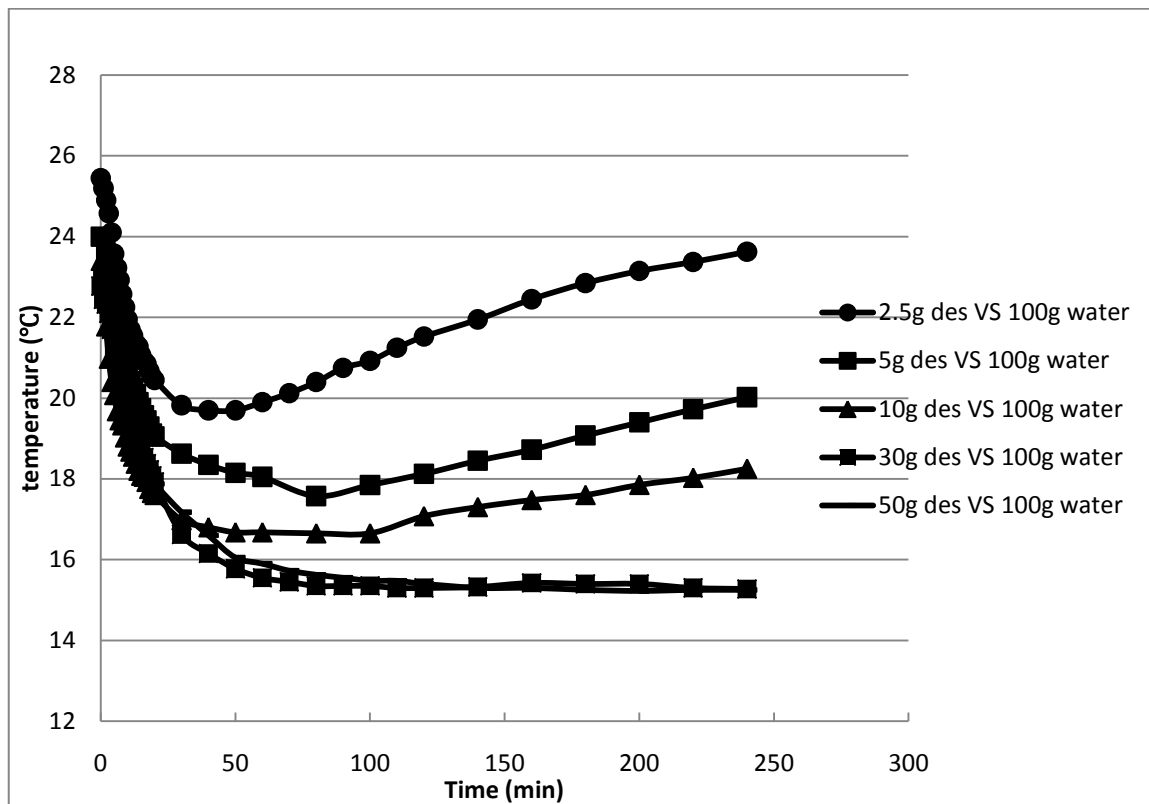


Figure 3.4 Water temperature profiles of AVEC experiments carried out at different desiccant masses with an initial water mass of 100 g, at a vacuum pressure of 28.8 In Hg and constant evaporation and adsorption surfaces. (Vacuum pump has been switched off after 5 minutes)

Table.3.1 Summary of the water temperature profiles shown in Figure 4.4

	T_{init} (°C)	T_{min} (°C)	T_{final} (°C)	ΔT (°C)
2.5g des : 100g water	25.45	19.70	23.63	5.75
5g des : 100g water	24	17.58	20.03	6.42
10g des : 100g water	23.4	16.65	18.25	6.75
30g des : 100g water	22.78	15.28	15.28	7.5
50g des : 100g water	22.9	15.25	15.25	7.65

As shown in Figure 3.4, the water side temperature profiles exhibit different patterns for these five different desiccant masses. The water temperature dropped during the water evaporation process, while the desiccant was heated up by absorbing vapor. As the temperature difference between the water side and desiccant side became larger, heat transfer took place inside of desiccator. When the heat transferred from desiccant side to water side was equal to the latent heat of evaporation from water side, it reached the pseudo steady state, therefore the water side temperature levelled off till the heat transferred from desiccant layer was larger than latent heat of evaporation, its temperature went up.

The lowest temperature (T_{min}), final temperature (T_{final}), and overall temperature differences (ΔT) are summarized in Table 3.1. The results show that the more desiccant mass in the system, the lower final water temperature it can reach. The lowest temperatures for those runs are significantly different except the runs with 30g and 50g of desiccant, which were similar to each other. It indicates the best temperature profile for this specific setup of tests. As those two runs perform both the best, 30g is the best desiccant amount for this specific setup in term of cost efficient.

All the five curves in Figure 3.4 show a similar curve shape which indicates a typical cooling process for water layer. At the initialization stage, the water temperature was as high as approximately 23-25 °C, therefore the difference of vapor partial pressure between the desiccant and the water layers was large, causing large evaporation fluxes from the water layer, which were adsorbed by the desiccant. And then desiccant layer was heated up due to the release of heat of vapor adsorption, resulting in the decrease of driving force due to the increase of vapour pressure at the desiccant surface and decrease of that at the water surface.adsorption. Therefore, the difference of vapor content between the water and the desiccant layers become smaller, resulting the decrease of evaporation rate, thus, the cooling effect in water layer decrease.

Meanwhile, after certain times, as the desiccant became more dilute with time, the evaporation rate decreased further more, then the water layer temperature begins to increase due to the heat transfer inside of the system, mainly from the warm desiccant layer.

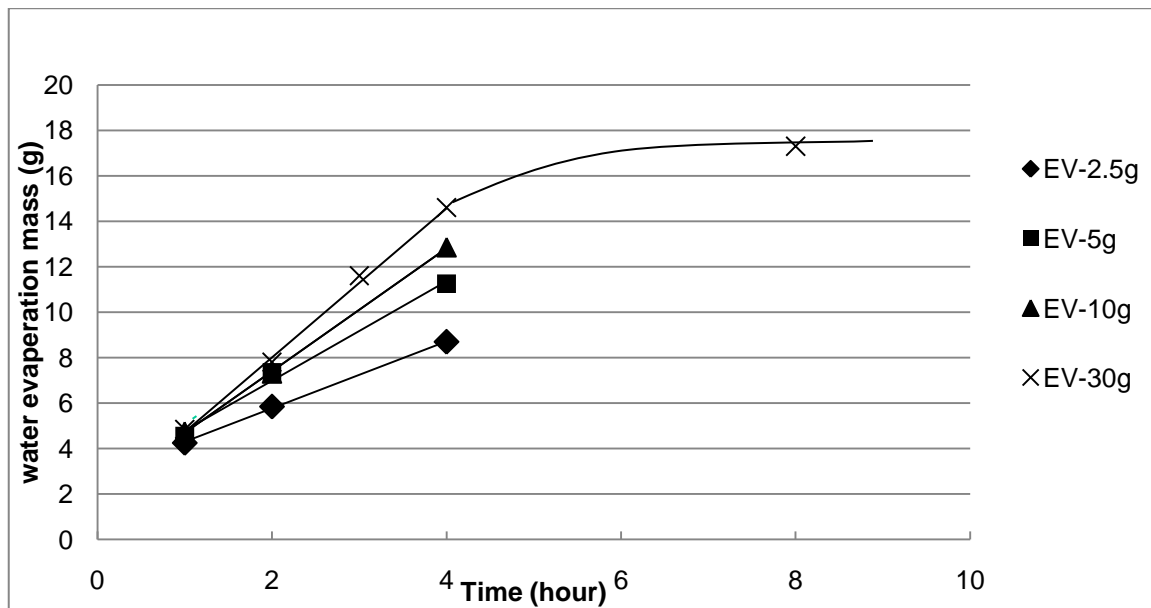


Figure 3.5 Water evaporation mass of AVEC experiments carried out at different desiccant masses with an initial water mass of 100 g, at a vacuum pressure of 28.8 In Hg

Water evaporation mass of AVEC experiments with different desiccant amount were measured at different time intervals, data has been presented in Figure 3.5. It shows a linear relationship for all the 4 different desiccant amount runs before 4-hour, but the speed of water evaporation slowed down afterwards, shown as the curves above for 30g desiccant runs.

As mentioned previously, the vacuum pump was shut off after the 5 minutes initialization period. The water evaporation rate, shown as the slope of each set of the data points before 4-hour run, were the same for each desiccant amount runs, which prove that the driving force for water evaporation is only the adsorption of water vapor by desiccant layer. While the evaporation speed decrease due to the decrease of vapor partial pressure difference between water and desiccant layers and also the gradually saturation of desiccant layer later on, the evaporation mass cannot keep a linear relationship with time after certain times, shown as the curve for 30g desiccant longer than 4 hour runs in Figure 3.5.

Table 3.2 Kinetics data of water evaporation

	2.5g desiccant	5g desiccant	10g desiccant	30g desiccant
Average flux during 4h (g/m ² /h)	83.6	123.7	153.1	204.5
Cooling capacity(w/m ²)	55.8	82.5	102.1	136.4
LiCl concentration at end of 4h	0.23	0.33	0.45	0.68

The fluxes for each desiccant amount run and the desiccant concentration at the end of 4-hour run have been listed in Table 3.2. 30g desiccant can deliver the largest evaporation flux (evaporation mass divided by the evaporation area then divided by time) which shows again that 30g desiccant is the best desiccant amount for this case investigated.

3.3.3 Influence of vacuum degree

Vacuum had an important role to play in the AVEC to facilitate mass transfer and enhance thermal insulation between the water layer and the desiccant layer. In order to investigate the best vacuum condition for the AVEC system, experiments with different initial vacuum degrees were carried out. Temperature profiles for 3 sets of vacuum degree tests for 4 hours has been shown in Figure 3.6.

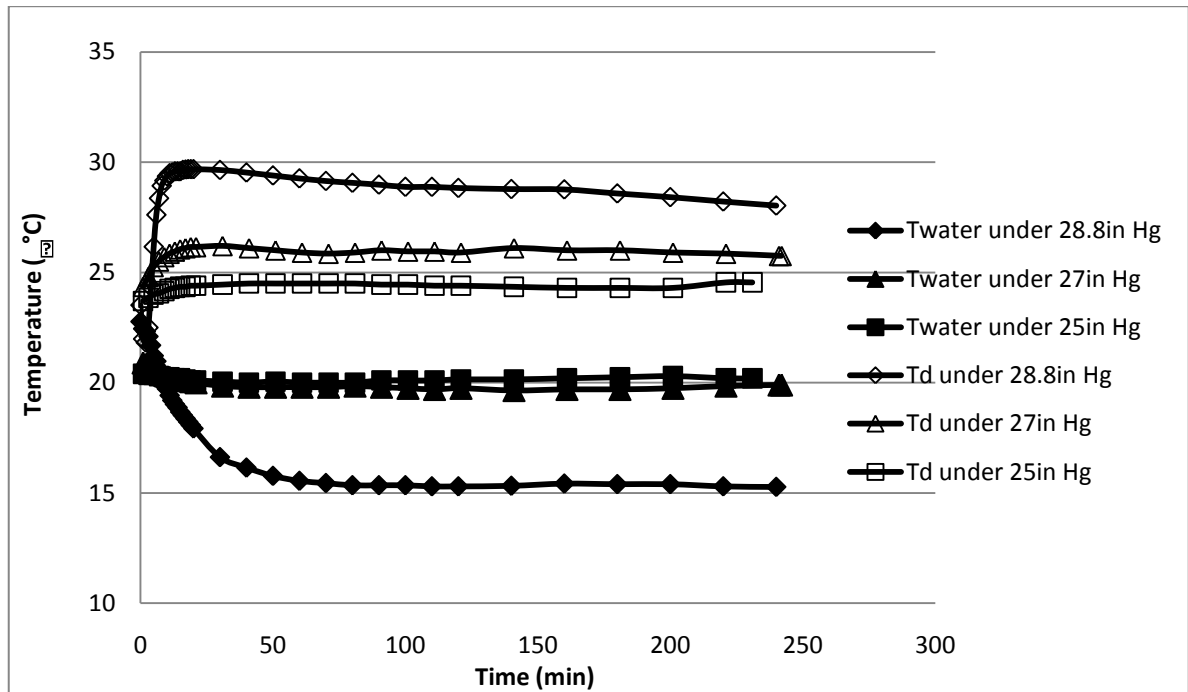


Figure 3.6 Water temperature profiles of AVEC experiments for 3 hours carried out at different vacuum degree with an initial water mass of 100 g, desiccant mass of 30g. (Vacuum pump has been switched off after 5 minutes)

As noticed from the Figure 3.6 above, the temperature profiles under different initial vacuum degree are not the similar shape anymore. As the 28.8 In Hg is the highest vacuum degree the system EW-07061 vacuum pump can reach at, the three sets of experiment were carried out under 25 In Hg, 27 In Hg and 28.8 In Hg, respectively. The 28.8 In Hg run has shown a typical evaporation and adsorption temperature profiles for water and desiccant layers, while the 27 In Hg run has a much worse data points which indicate poor evaporation in the system. The data with 25 In Hg initial vacuum degrees performed the worst as there is only slightly temperature drop at the first 30 min for water layer temperature and then stabilized for the rest two and half hour run.

In addition, it was observed that only when the vacuum was maintained at 28.8 In Hg that bubbling of water was maintained in the transit stage. Since water evaporation directly depends on the system pressure, when the system pressure reduced to the saturation pressure corresponding to the water layer temperature, water start to boil untill new equilibrium condition was achieved. Therefore the requirement for vacuum degree initialization stage is essential for effective AVEC.

3.3.4 Influence of AD/EV ratio (adsorption area versus evaporation area)

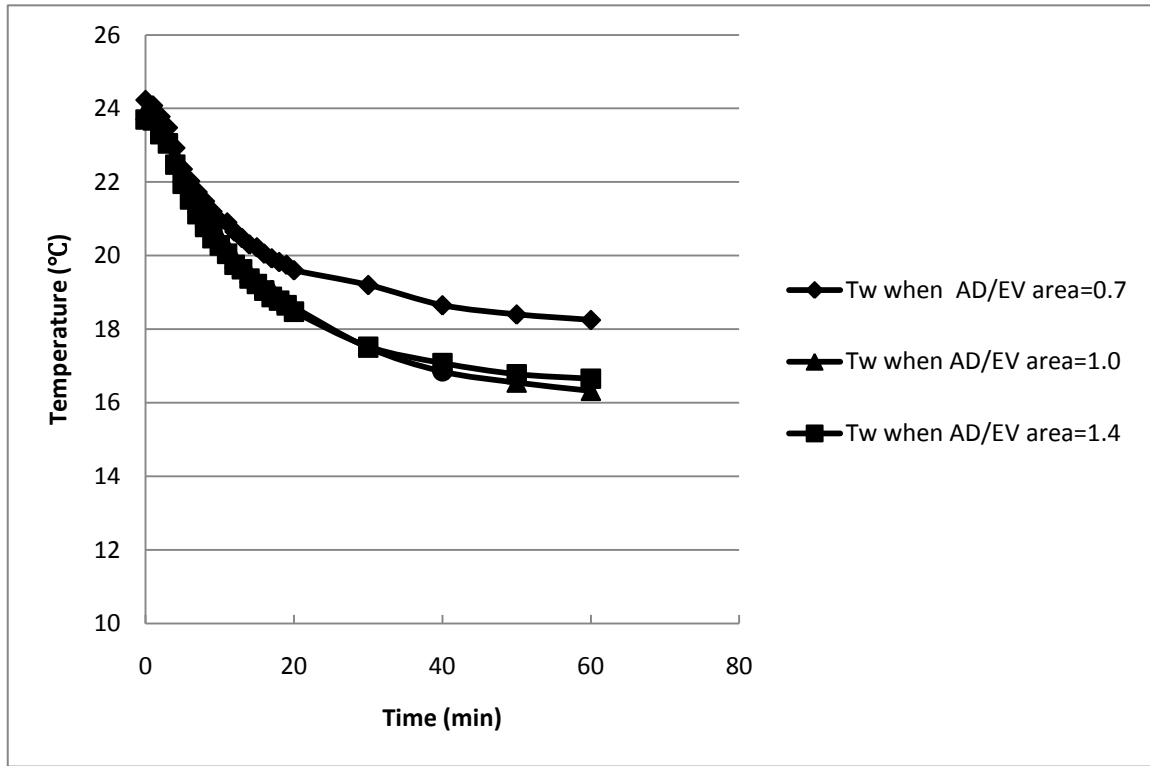


Figure 3.7 Water temperature profiles of AVEC experiments carried out at different adsorption area with an initial water mass of 100 g, desiccant mass of 30g, at a vacuum pressure of 28.8 In Hg. (Vacuum pump has been switched off after 5 minutes)

Table 3.3 Evaporation fluxes at different AD/EV

AD/EV	0.7	1	1.4
Flux(g/h/m ²)	248.6	262.7	296.6
Cooling capacity (w/m ²)	165.9	179	197.9

In this study, different adsorption over evaporation area ratios were investigated for better temperature profile for water layer. From Figure 3.6, the run with 1.0 and 1.4 AD/EV ratio have almost overlapped temperature profile while the AD/EV=0.7 run gives a slightly worse set of data for water layer. The flux was calculated by using evaporation mass divided by the evaporation area and then divided by the time, and the cooling capacity was calculated using the flux multiplied by latent heat of evaporation at corresponding temperature.

As shown in Table 3.3, the run with AD/EV=1.4 had the largest evaporation flux of $296.6 \text{ g h}^{-1}\text{m}^{-2}$. However, the run with AD/EV=1 resulted in the lowest water temperature at the end of the one hour run, which is shown in Figure 3.6. These observations could be tentatively explained by the hypothesis that the increase of adsorption area provided a larger desiccant surface area, facilitating the adsorption process. Meanwhile, the more adsorption in desiccant layer leading to the more adsorption heat release, resulting potentially more heat transfer from the warm side (desiccant layer) to the cold side (water layer) which affect the temperature profile.

In terms of the system design, it is always easier to have a same evaporation and adsorption area system than a large AD/EV ratio system, so we keep the same area for evaporation and adsorption in the design of AVEEC system.

3.3.5 Influence of aluminum foil

Internal heat transfer between the cold water layer and the hot desiccant layer may exist due to the close distance between these two objects that are of remarkably different temperatures inside an AVEC system. The internal heat transfer should be minimized because it would neutralize a portion of the latent heat of water evaporation, which should ideally be

compensated completely by adsorbing heat from the environment, i.e., the object to cool. Since a high vacuum degree was maintained throughout the process and the mass transfer occurred primarily by the means of vapour diffusion from the cold water surface to the hot desiccant surface, it was reasonable to assume that internal heat transfer from the hot desiccant to the cold water was primarily carried out by means of radiation. Therefore, minimizing internal heat transfer via radiation is essential to improve the cooling performance of the system,

We investigated the effects of reflective aluminum foil covering the whole desiccator (except for the lid in order to record data by reading the thermometers) and covered only to prevent internal radiation from the hot desiccant layer. Data are shown in Figure 3.8 and Figure 3.9.

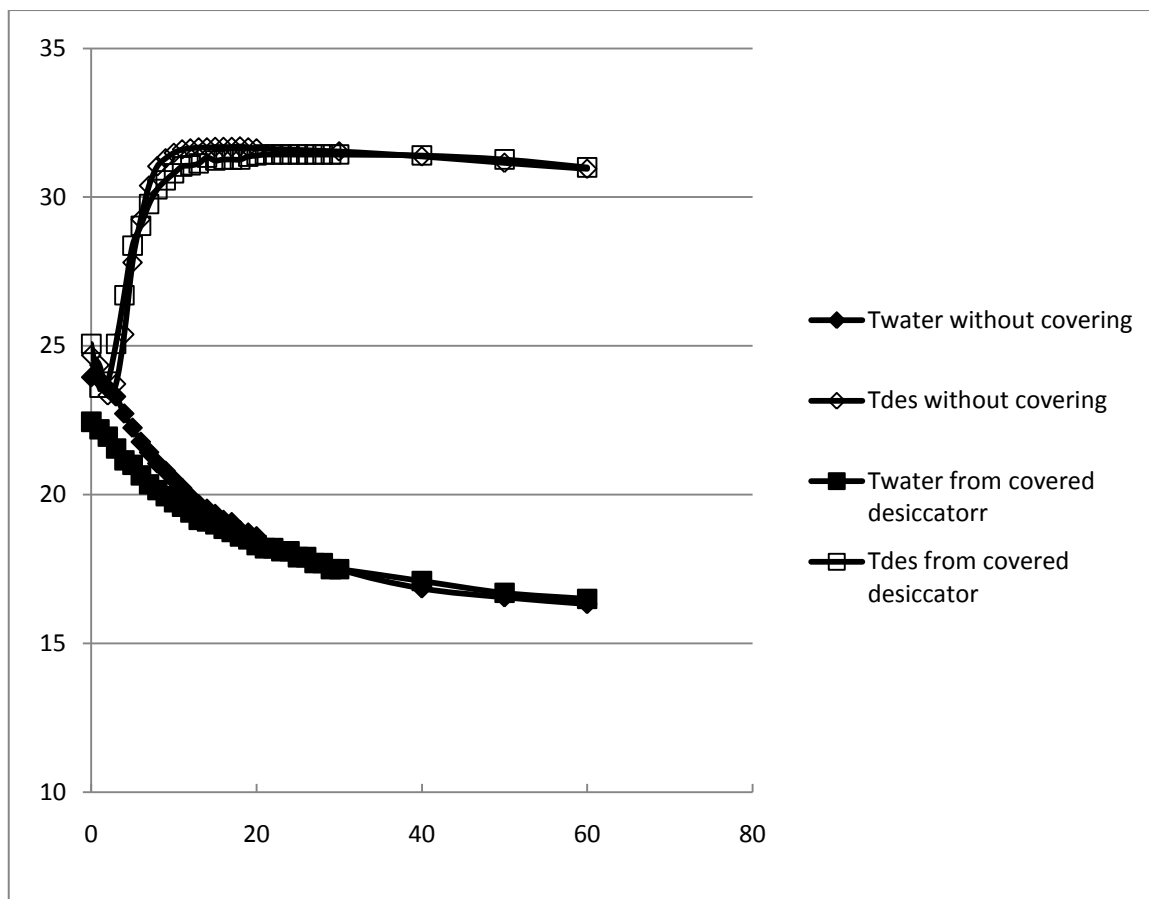


Figure 3.8 Temperature profiles of AVEC experiment carried out with 100 g water, 30 g LiCl powder, at a vacuum pressure of 28.8 In Hg at room temperature with and without aluminum foil covered around whole body of the desiccators for one hour. (Vacuum pump has been switched off after 5 minutes)

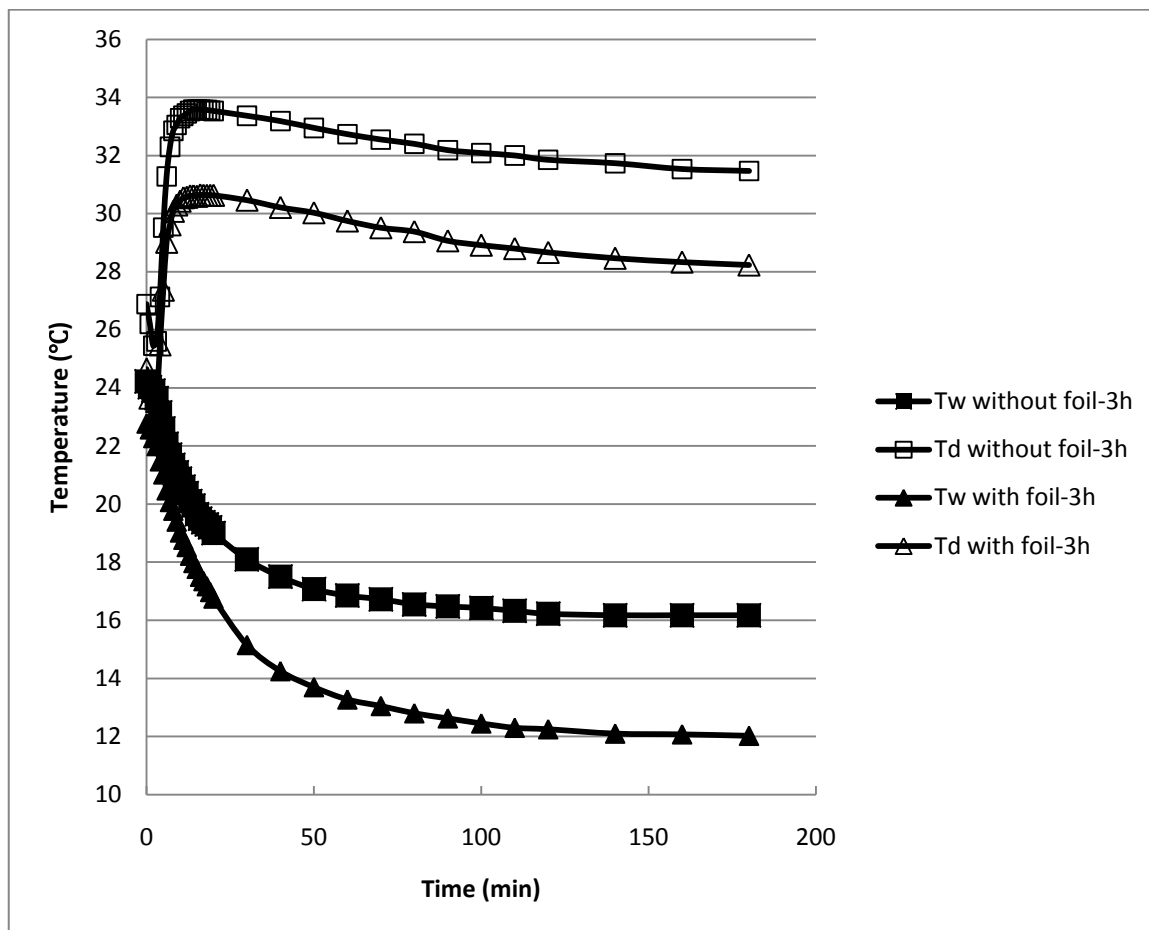


Figure 3.9 Temperature profiles of AVEC experiment carried out with 100 g water, 30 g LiCl powder, at a vacuum pressure of 28.8 In Hg at room temperature with and without aluminum foil covered only at water layer surface for three hour. (Vacuum pump has been switched off after 5 minutes)

As shown in Figure 3.8, there was no significant difference on both water and desiccant layers temperature profiles, indicating that the heat transfer by radiation from ambient environment to AVEC system was negligible. In contrast, the run with aluminum foil covered only at water layer performed much better than the run without foil, which brought the final temperature at water layer 4.2 °C lower comparing to the run without foil on the water layer at the end of 3-hour run.

This results show that radiation as a means of heat transfer had an important role in the internal heat transfer between the hot desiccant and cold water layers of the AVEC system. Aluminum foil covering the water layer can drastically improve the cooling performance of the system by reflecting the radiation coming from the desiccant layer away from the water layer.

3.4. Discussion

3.4.1 The adsorptive vacuum evaporative cooling

As aforementioned, Figure 3.2 shows the temperature profiles of the water layer and the desiccant layer of a typical AVEC process, which could be divided into three stages, the initialization stage, the transit stage, and the pseudo steady state stage.

It was clear that flashing take place in the initialization stage because a significant drop of water temperature. 2-3 °C was observed in association with strong bubbling in this period. It shown that desiccant started to absorb water vapour in the initialization stage, as indicated by the sharp increase of desiccant temperature in this period. And also at the starting point, the desiccant is in powder condition which gives full capacity to absorb water vapor.

As time passed, desiccants get wet and turned to solution. The concentration of desiccant solution became smaller and smaller, so did the driving force for absorbing water vapor. As a result, the rate of evaporation decreased and the temperature drop for water side became slower and slower.

Since we shut the vacuum pump off after 5 minutes of initialization, it's not the vacuum pump but the adsorption that maintained the vacuum in the desiccator.

In terms of heat transfer, the water dish was in contact with the plate and then desiccators. Heat could transfer from the ambient environment to the water and from the desiccant to the ambient environment via conduction. Since the desiccant was at a relatively high temperature, radiation always happened. So there was heat transfer inside of the system which might bring the water temperature up and desiccant temperature down.

Overall, it shows that the more desiccant put in, the bigger temperature differences between two layers at the end of 4-hour test, and also, we can find out that we are able to reach pseudo-steady state for 30g desiccant versus 100g water during 4-hour test, and that is therefore our fixed ratio for making cooling pads.

The energy balance on the liquid water in petri dish is given by the following governing equation, assuming one dimensional heat transport.

$$\frac{dQ_{cv}}{dt} = \dot{Q}_{sur} + \dot{Q}_d - \dot{Q}_{ev} \quad (3.1)$$

Where dQ_{cv}/dt is the rate of sensible heat accumulation in the control volume (i.e., water), \dot{Q}_d the rate of heat transfer from the hot desiccant, \dot{Q}_{sur} the rate of heat transfer from the surroundings and \dot{Q}_{ev} is the rate of latent heat removal, which is given by the following equation:

$$\dot{Q}_{ev} = m_{ev} \Delta H_{ev} \quad (3.2)$$

Where m_{ev} is the mass flowrate of water evaporation, and ΔH_{ev} is the specific latent heat of water evaporation.

At the beginning, water layer had a similar temperature as the ambient environment, so did the desiccant layer. Therefore $Q_{sur}=0$; $Q_d=0$, and as discussed above, the evaporation rate (Q_{ev}) was the largest. As a result, $\frac{dQ_{cv}}{dt} < 0$ and the control volume, i.e., the water, cooled down quickly.

In the transit phase, desiccants adsorbed more vapour and was heated up while water layer was cooled down, resulting the decrease of water evaporation and therefore the rate of heat removal from the water layer. At the same time, both heat transfer from the desiccant (Q_d) and that from the environment (Q_{sur}) increased with time due to the increase of desiccant temperature and the decrease of water temperature. When at a certain point of time the rate of heat removal by water evaporation was balanced by heat gaining via heat transfer from the environment and from the desiccant, i.e., when $Q_{sur} + Q_d = Q_{ev}$, the control volume was reached steady state and water temperature remained constant. This was how the water side temperature reached pseudo steady state.

Heat balance could also be used to explain the water temperature profiles of experiments carried out with different amount of desiccants as shown in Figure 3. 4. As shown in Figure 3.4, pseudo steady state of the water temperature profile was achieved only when 30 g or 50 g of desiccants were used in the experiments. For desiccant of 10 g or less, water temperature hit the lowest point and then started to increase. The less amount desiccant used, the sooner the lowest temperature reached and the higher the lowest water temperature. This seems to suggest that for these experiments, the rate of heat removal from water by evaporation was balanced by heat transfer from the desiccant and the environment at the point when the

lowest water temperature was reached. Then, since the desiccant solution was diluted quickly due to the small quantities of LiCl, the evaporation fluxes decreased rapidly, resulting in a scenario where the rate of latent heat removal via water evaporation was smaller than the heat transfer into the water from the desiccant and the environment. Consequently, water temperature increased after the lowest temperature point was passed.

3.4.2 The driving force

As all the experiments above were carried out with a 5 minutes vacuum pump initialization stage, which was followed by completely unpowered operation as vacuum pump was cut off. It was interesting to observe that the system vacuum was maintained at the same level throughout the process and approximately the same evaporation rate was maintained within 4 hours. These observations indicate that adsorption of water vapor by the LiCl powder layer was sufficient to maintain the vacuum in the system and the driving force of AVEC system was the partial vapour pressure difference between that at the water surface and that at the desiccant surface.

Vacuum played an important role in the AVEC system, for example, facilitating mass transfer of vapor from the water surface to the desiccant surface. It was not a surprise to observe that, as shown in Figure 3.6, the cooling effects was minimal when the vacuum degree was 25 or 27 In Hg while that it was quite remarkable at 28.8 In Hg. These data seem to suggest that a minimum vacuum degree was required for the AVEC system to work efficiently. This observation was coincident with the observation that water bubbling, which was an indicator of water boiling, was observed at a vacuum of 28.8 In Hg but was absent when it was 27 or 25 In Hg.

3.5 Conclusion

An adsorption vacuum evaporative cooling (AVEC) system was demonstrated to be working well without power supply after 5-minute initialization. Results show that the AVEC could deliver 179 W/m^2 cooling capacity while maintaining water temperature at 12 to 15 °C in an ambient temperature of approximately 24 °C. A high degree of vacuum was maintained throughout the process inside the AVEC system by the means of vapor adsorption of desiccants. A reflective aluminum foil greatly enhanced the cooling performance of AVEC by minimizing internal heat transfer via radiation. It was also concluded that a minimum degree of vacuum was required for achieving efficient cooling using AVEC.

3.6 Acknowledgment

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CHAPTER 4

Personal-portable cooling garment based on adsorption vacuum membrane evaporative cooling

Abstract

A personal-portable cooling garment based on the AVMEC concept was developed. It first was demonstrated with cooling pad that a cooling capacity of 277.4 W/m^2 could be achieved in an ambient environment of $37 \text{ }^\circ\text{C}$. Tests with human subjects wearing prototype cooling garments consisted of 12 cooling pads with an overall weight of 4 kg indicated that the garment could effectively maintain a core temperature 0.5 to $1.5 \text{ }^\circ\text{C}$ lower than the same subject wearing shorts only walking on a treadmill in an environment of $40 \text{ }^\circ\text{C}$ and 50% relative humidity (RH) within 100 minutes. Tests also show that the AVMEC cooling garment was more efficient than a commercial ice-pad vest when the subject wore NWBC suit and was working under the same conditions.

Keywords: evaporative cooling, vacuum membrane cooling, personal cooling, cooling garment.

4.1 Introduction

Several occupations such as firefighter, soldiers and oil refinery worker expose workers to extreme heat stress in their daily work environment. Heat stress management is important for these workers because body overheating may lead to serious problems, which could be fatal. The typical symptoms include a series of disorders such as discomfort and pain (heat rash and heat cramps), life-threatening conditions (heat exhaustion and heat stroke) and also dehydration (Nunneley 1970; Allan 1988; Bishop et al. 1991).

Different cooling technologies have been developed to help manage heat stress. These technologies can be classified according to different criteria. According to the scale of operation, cooling could be classified as space cooling and microclimate cooling (i.e., personal cooling garment). The subject of space cooling is the whole working space, which is much more costly than microclimate cooling, which cools the microenvironment directly surrounding the worker only (Chauhan 1999).

Ventilation, air cooling, fans, shielding, and insulation are the five major types of engineering controls used to reduce heat stress in hot work environments (E. bakaya-Kyahurwa). General ventilation is used to dilute hot air with cooler air. This technique clearly works better in cooler climates than in hot ones. A permanently installed ventilation system usually handles large areas or entire buildings. Portable or local exhaust systems may be more effective or practical in smaller areas. Air conditioning is a method of air cooling, but it is expensive to install and operate for large spaces. An alternative to air conditioning is the use of chillers to circulate cool water through heat exchangers over which air from the ventilation system is passed.

Another way to reduce heat stress is to increase the air flow (e.g. using fans, etc), in the work area (as long as the air temperature is less than the worker's skin temperature). Changes in air speed can help workers stay cooler by increasing both the convective heat exchange (the exchange between the skin surface and the surrounding air) and the rate of sweat evaporation. Because this method does not actually cool the air, any increases in air speed must impact the worker directly to be effective.

When sources of radiation, such as heating pipes are involved, both insulation and surface modifications can be used to achieve a substantial reduction in radiant heat. Instead of reducing radiation from the source, shielding can be used to interrupt the path between the source and the worker. Polished surfaces make the best barriers, and special glass or metal mesh surfaces can be used if visibility is a concern. Shields should be located in such a way that they do not interfere with air flow, unless they are also being used to reduce convective heating. The reflective surface of the shield should be kept clean to maintain its effectiveness.

Heat mitigation can be achieved by using power assists and tools that reduce the physical demands placed on a worker. Personal microclimate cooling is the most common method to help the heat mitigation. Various microclimate cooling technologies have been developed in the last few decades. They can be classified into three groups: fluid conditioned cooling garment, phase change material (PCM) cooling garment, and evaporative cooling garment (Chauhan et al. 2008).

Fluid conditioned cooling garments use chilled fluids, most commonly water and air, to cool one's body. Circulating air was the most popular approach in the earliest development (Bishop et al. 1991). By directing compressed air around the body from an air supply system,

both evaporative and convective cooling is improved. The greatest advantage of this approach is that air is readily available all the time. Its disadvantage, however, include the requirement of continuously power supply, and that the whole garment is relatively heavy and bulky due to the packed tubing.

More recently, it has been shown that water-cooled garments are the most effective personal cooling devices in commercial use at present (Chauhan 1999). Use of this equipment requires a battery-driven circulating pump, liquid-ice coolant, and a container (Cadarette et al. 2006). The great advantages of water cooled garments comparing to air cooled garment is that water has higher specific heat capacity, which is 4.181 kJ/kg·K at 25°C, comparing favourably to that of dry air, which is 1.005 kJ/kg·K at the same temperature. In addition water has a thermal conductivity of 0.58 W/m·K, superior to that of air, which is 0.024 W/m·K.

Phase changing materials (PCM) are those solids that ideally melt at the temperature range of 0 to 30°C, making them good candidates for personal cooling. PCM cooling garment is simple and portable as it requires no power supply and involves no mobile parts. However, heat of fusion of PCM is in general very small. For instance, that of ice, which is one of the largest heats of fusion of PCMs, is only 330kJ/kg at 25 °C. Consequently, they can provide only moderate cooling capacity in a very limited operation time. For instance, ice vests, the mostly available PCM cooling garment in the market, can last only 2 to 4 hours at moderate to heavy heat loads, and frequent replacement or recharging is necessary (Cadarette et al. 2006).

Evaporative cooling garments are a relatively new development. The early design of evaporative cooling garments were so called water soaked cooling garments, in which the

fabric is soaked up with water like sponge. It can absorb water fast and then release water slowly through evaporation. Most recently, Rothmaier et al. (2008) proposed a concept of three-layer laminates, which employed membrane technology and was based on body temperature driven evaporation of water out of skin-contacting water reservoir. The advantage of EVG is that it employs the large latent heat of water evaporation, which is around 2400 kJ/kg at 25°C more than seven times of the heat of ice fusion. However, a major problem with the conventional EVG is that its cooling performance depends heavily on the ambient humidity and temperature and is of low efficiency or even dysfunctional under humid and hot environment, where personal cooling is most needed.

4.2 The concept of adsorption vacuum membrane evaporative cooling

As shown in Figure 4.2, an AVMEV cooling pad is composed of three major components, a cooling core (water layer), a spacer, and an adsorption core (desiccant layer). The cooling core is a soft impermeable water bag with the side facing the spacer replaced with a semi-permeable membrane that allows only water vapour to pass through but retains liquid water. The membrane provides the barrier for liquid/vapour phase separation. The opposite side that faces the environment is the interface for heat exchange with the subject to be cooled. The spacer serves to create a space of vacuum that, after initializing with a vacuum pump, separates the cooling core from the adsorption core. The adsorption core is another soft impermeable bag with the side facing the spacer replaced with a piece of semi-permeable membrane. A fabric layer onto which desiccant powders were distributed evenly was

inserted into it. These three components, the cooling core, the spacer and the adsorption core were sandwiched with the spacer in the middle to form a cooling pad. A soft cloth was inserted between the spacer and the cooling core and between the spacer and the desiccant holder (the adsorption core) for the protection of the membranes. A piece of perforated aluminum foil was inserted between the protection cloth and the spacer facing the cooling core to reflect the radiation from the hot desiccant away from the cold water. An outer bag was used to bag the entire sandwich structure and was sealed air-tight using hot glue.

In such an AVMEC pad, cooling was achieved by water evaporation, which is facilitated by the vacuum that is created by initializing the structure with a high performance vacuum pump and maintained by continuous vapour adsorption/absorption by the desiccant on the other side of the spacer (in the adsorption core). The driving force for mass transfer is the vapour pressure difference between the water surface and the desiccant surface. The vacuum between the cooling and the adsorption cores serve two functions, facilitating water evaporation and enhancing the thermal insulation between the hot desiccant and the cold water, which are at close proximity in the sandwich structure.

The major advantages of the AVMEC technology include: 1) It takes advantages of the large latent heat of water evaporation, which is approximately 2400 kJ/kg, approximately 7 times of that of the heat of fusion of ice; 2) It is independent of ambient temperature and humidity; 3) It can provide large cooling capacity because vacuum is used to facilitate water evaporation; 4) internal heat transfer between the hot desiccant and the cold water is minimized; 5) vapour sorption rather than vacuum pump is used to maintain the vacuum and therefore no power supply or mobile parts are involved except in the initialization stage, which only takes as short as 5 minutes; and 6) water is nontoxic, cheap, and easily available

for recharge. As a result, a lightweight, man-portable, and efficient cooling garment that does not rely on ambient conditions for best performance and can provide relatively large cooling capacity for a long period of time could be developed.

In this chapter, we report the results of the proof of concept studies on a man-portable cooling garment using the innovative cutting edge AVMEC concept, which was based on AVMEC, which was based on the concept of adsorption vacuum evaporative cooling (AVEC) discussed in chapter 3 with the hydrophobic membranes included to provide the barrier for liquid/vapour phase separation. Experiments were carried out in two phases. First, cooling pads were fabricated and then tested in an incubator maintained at 37 °C. In the second phase, prototype cooling garments consisted of 12 cooling pads with a total active evaporation area of 0.3 m² was tested with human subjects. The results showed that the man-portable AVMEC cooling garment, which has a total weight of 4.0 to 5.0 kg, was capable of providing significant heat stress mitigation for subjects at a moderate working load in an environment of 40 °C and 50% RH wearing a NWBC suit or not.

4.3 Experiments setups and procedures

4.3.1 Cooling pad fabrication

The parts for assembling cooling pad have been presented in Figure 4.1. Part (a), is the cooling core (water bag) with membrane surface on top; part (b), the adsorption core (desiccant powder), sealed in a vacuum bag to prevent it from absorbing moisture from surrounding; part (c), the spacer, a single layer or multi-layer honeycomb covered with a

piece of perforated aluminum foil followed by a layer of cotton towel; part (d), is the outer bag with a connection tubing.

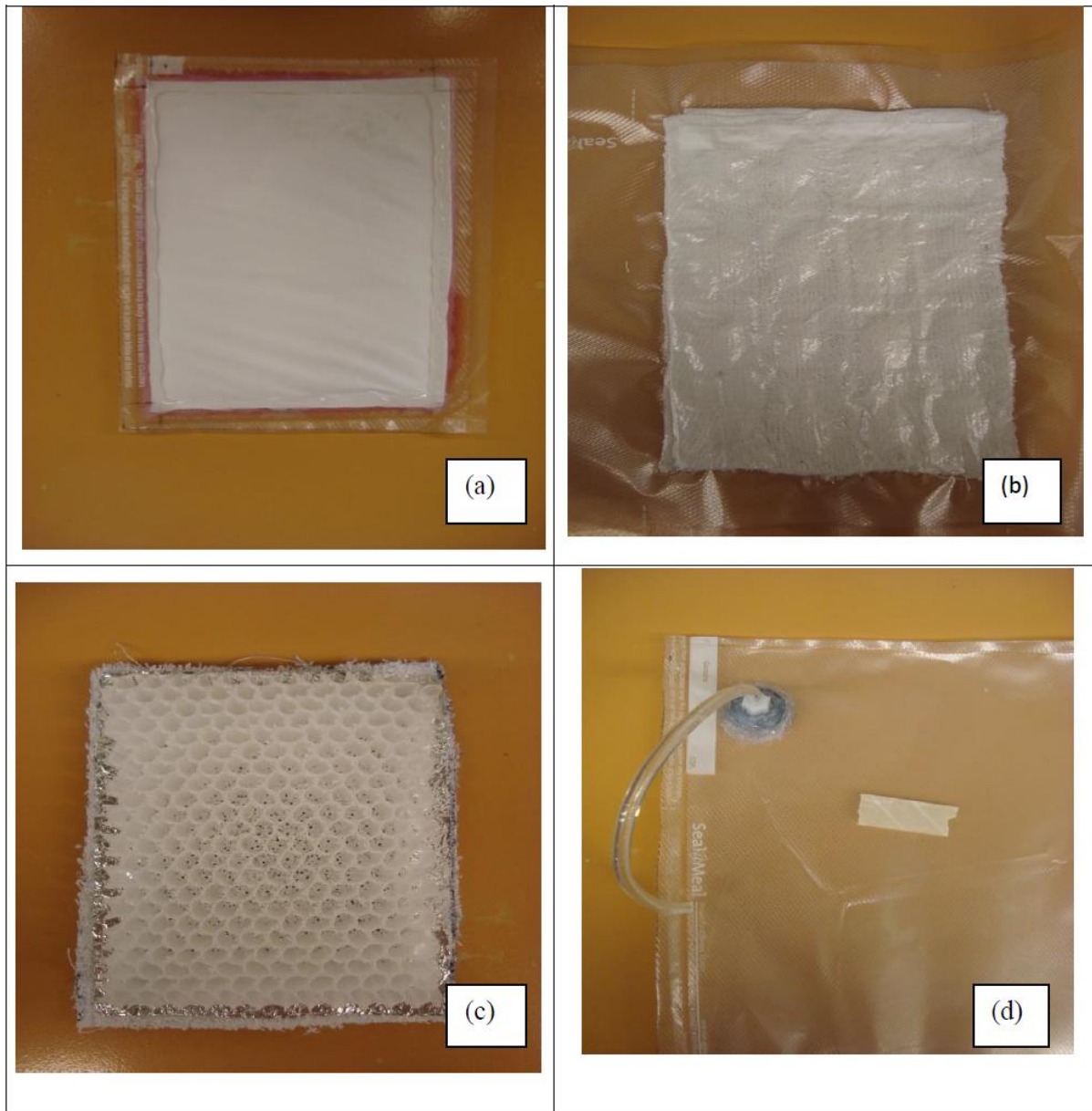


Figure 4.1 Parts for assembling a cooling pad. (a) the cooling core (water bag); (b) the adsorption core (desiccant holder); (c) the honeycomb spacer with perforated aluminum foil; (d) the outer bag

In this study, two different sizes of cooling pads were fabricated. The larger one has the size of 180×250 mm with the effective membrane area of 170×220 mm; and the smaller ones had a size of 140×190 mm with active membrane area of 170×120 mm. The honeycomb spacers were single-layer, double-layer, or triple-layer, corresponding to a total thickness of 8, 16, or 24 mm, respectively. The completed cooling pad was 25-30 mm thick.

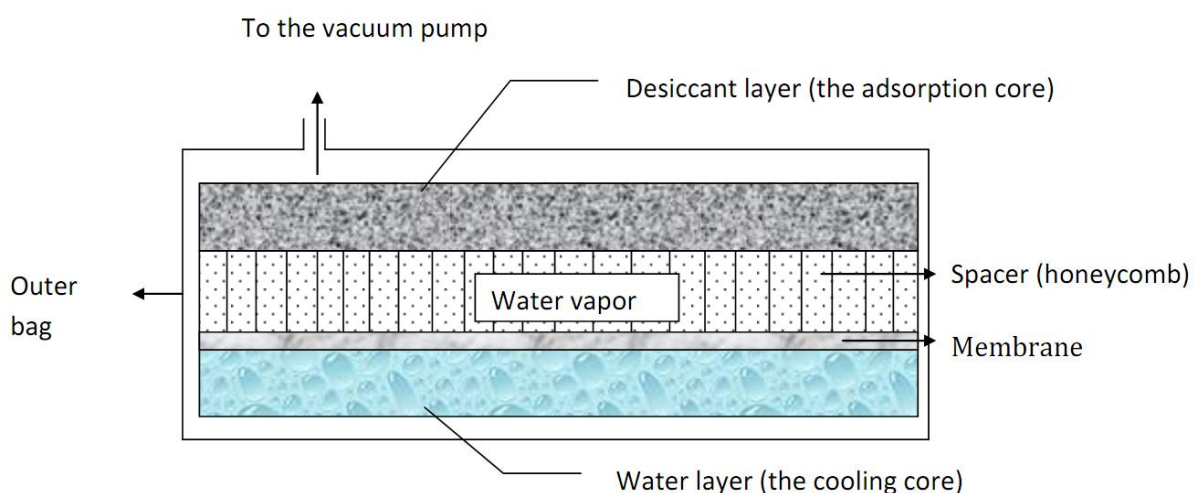


Figure 4.2 Schematic diagram shown the cross-sectional view of a cooling pad experiments setup and procedure

The schematic of cooling pad is shown in Figure 4.2. It contained three major parts: adsorption core, spacer and cooling core. In this study, LiCl was chosen to be the desiccant. LiCl powder was spread evenly onto a piece of cotton towel, which was covered by another piece of towel. Hot glue was used to glue these two pieces of towel together and also separate between each powder square to keep the powder in place. The spacer was made of polypropylene honeycomb with a density in the range of 3.5-20 lb/ft³. It had a very low thermal conductivity, very high moisture resistance and high chemical resistance (Plascore®). It was put between the two layers to prevent direct contact of the two. The

cooling core was made with the vacuum storage bag (Seal-a-Meal®), manufactured by Sunbeam Product, Inc. One side of the bag was replaced with a piece of Teflon (PTFE) unlaminated membrane (Sterlitech, US). The PTFE (polytetrafluoroethylene) had a pore size of 0.2 μm and a thickness in the range of 0.061-0.081mm. It had enough strength for fabrication and at the same time had the pore size to allow vapor pass through but not liquid water.

The outer bag which is made of the same type of commercialized vacuum storage bag (Seal-a-Meal®), was used to provide the vacuum environment for evaporation and thermal insulation when initialized with a vacuum pump. It has vinyl tubing which could connect the cooling pad with the vacuum pump for initialization. After the initialization period, the tubing was knotted to cut the cooling pad from the vacuum pump. As a result, the cooling pad operated detached from the vacuum pump from then on.

4.3.2 Cooling pad experiments procedure

In cooling pads experiments, an incubator (MaxQ 5000, GENEQ Inc) was used to maintain a temperature of 37°C. The cooling pad was first connected to the vacuum pump (WZ-07061-11, Cole-Parmer), which could deliver a maximum vacuum degree of 29 In Hg, a free air capacity of 32.5L/min with an 115V AC power requirement at room temperature. The pads were then put in a preheated incubator and the vacuum pump was turned on for about 5 minutes to achieve a vacuum degree of 28.8 In Hg (3.725 kPa). Then the tubing on the pad was closed by knotting and disconnected from the pump. The cooling pad was then put inside the incubator. Temperature profiles of the surface of the adsorption core and that of the cooling core were monitored by taping the probe of Oakton Temp-300 dual-input type K

thermocouple (Cole-Parmer) on to the surfaces of corresponding cores as soon as the pump was on. Water evaporation flux was estimated by weighing water loss at the end of each test, which was divided by the time of evaporation and the area of active membrane surface.

Based on previous results, 30g desiccant was used for building the cooling pad with an initial water mass of 100g, the highest vacuum degree achievable with the vacuum pump (approximately 28.8 In Hg) was chosen to be the cooling pad initialization vacuum degree. The effective membrane areas of the adsorption and the cooling cores were the same, which were 181 cm².

4.3.3 Human subject experiments setup and procedure

The AVMEC garment prototype was composed of 12 cooling pads attached to a regular XL T-shirt. As shown in Figure 4.3, there were two cooling pads of 170×220 mm effective membrane dimensions at the chest and the back part (i.e., the upper position), while four cooling pads of 170×120 mm effective membrane at the lower. Each cooling pad was attached to the T-shirt using four Velcro, so that it could be replaced with new one easily when necessary.

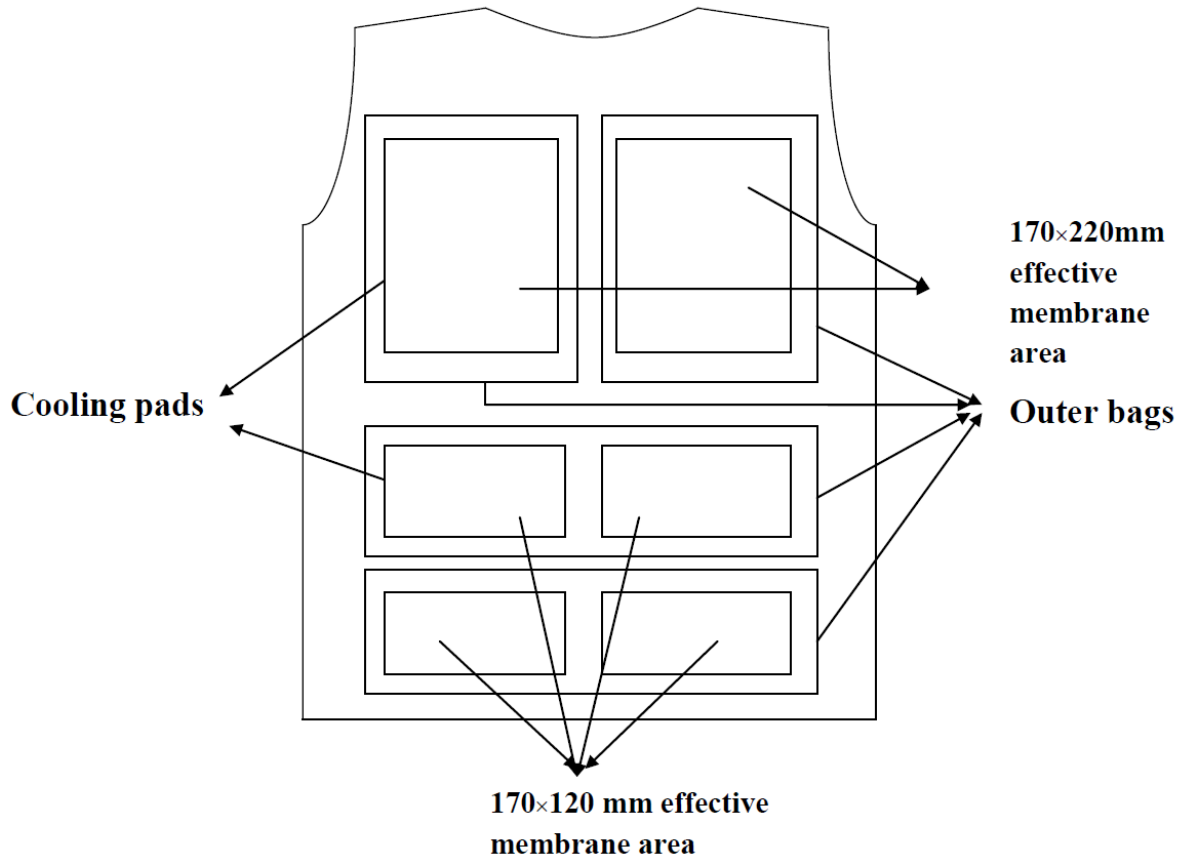


Figure 4.3 Schematic diagram of the inside view of the front cooling garment (the back is same)

All human subject tests were carried out with a 22 year old male of a weight of around 70 kg. Pictures of the subject dressed in the cooling garment are shown in Figure 4.4a, and that of the subject walking on a treadmill is shown in Figure 4.4b. During the tests, the subject performed a walking protocol on a treadmill at 3 miles per hour and 2% incline. The exercise was performed in a thermal chamber maintained at 40°C and 50% RH. Trials lasted for 90 minutes unless they were terminated for safety reasons. Core body (rectal) temperature and different parts of skin temperature were monitored.

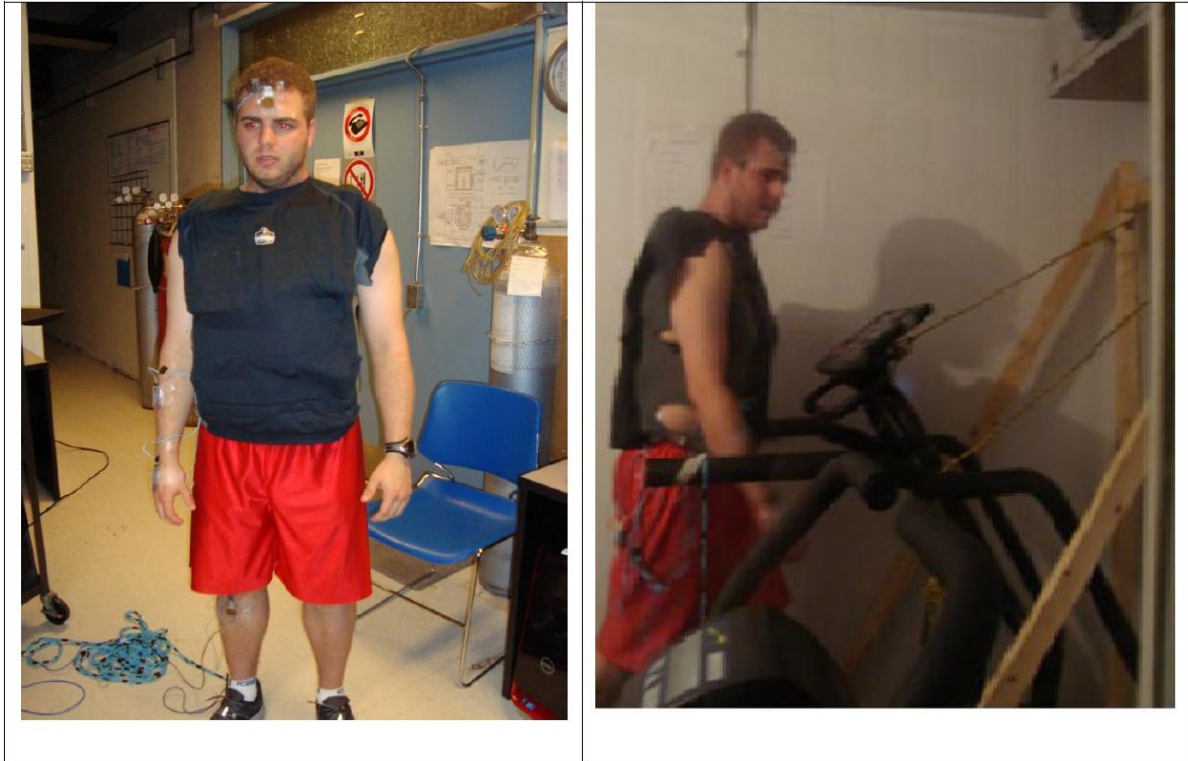


Figure 4.4 Human subject test a) test subject wearing cooling garment (left side); b) test subject walking in thermal chamber walking on a treadmill (right side).

Four different tests were conducted: 1) the subject wore a T-shirt and a pair of shorts only; 2) the subject wore the AVMEC garment on top of the T-shirt and shorts; 3) the subject wore a NWBC suit on top of a commercial ice vest plus the T-shirt and shorts; 4) the subject wore the NWBC suit on top of the AVMEC garment plus the T-shirt and shorts.

4.4 Results

4.4.1 Cooling Pad tests

4.4.1.1 Influence of continuous vacuum pump

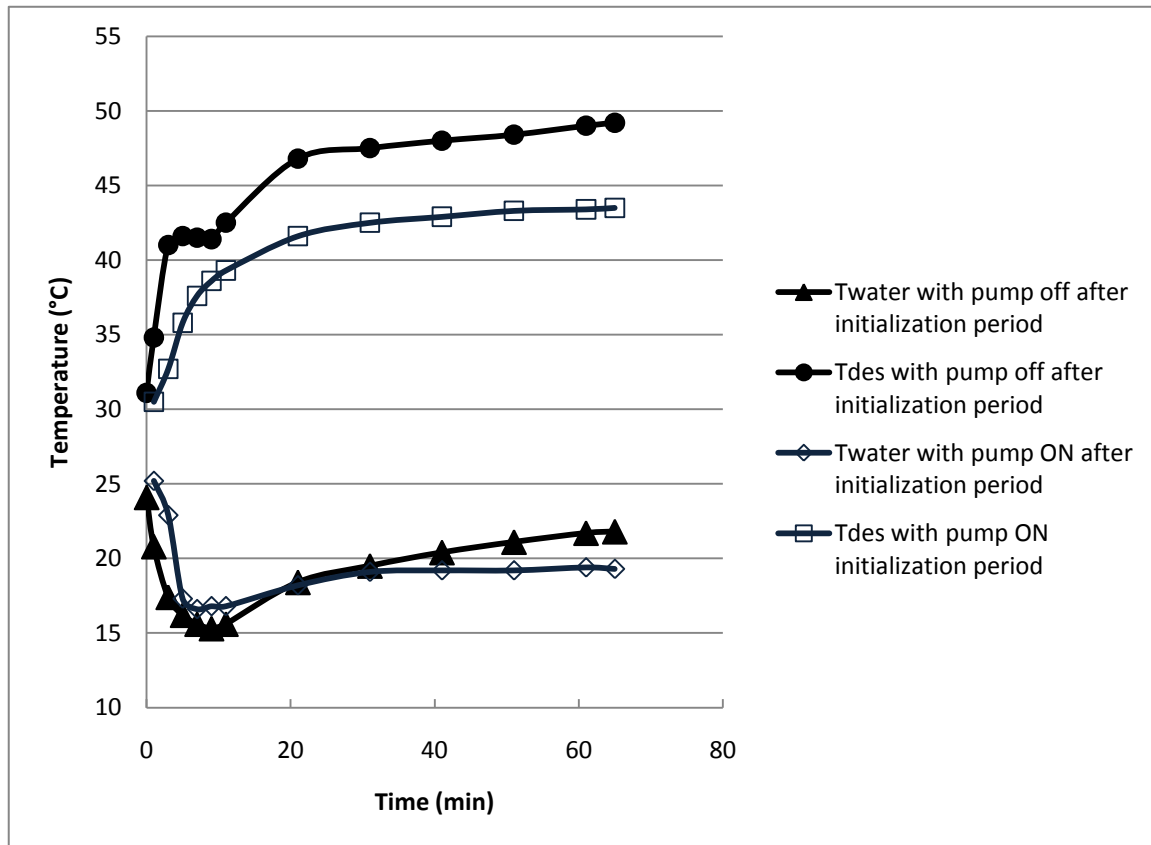


Figure 4.5 Temperature profiles of water and desiccant layers in a typical cooling pad with aluminum foil built in. Experiments were carried out with pump on (empty symbols) or pump off (solid symbols) after initialization period with 100 g water, 30 g LiCl powder, at a vacuum pressure of 28.8 In Hg (3.725 kPa) at 37 °C environment inside an incubator.

The temperature profiles comparison between vacuum pump on during the whole run and the one with pump off after an initialization period are shown in Figure 4.5. The temperature profiles have the same trend which can be divided into two stages, the initialization stage and

the transit stage, a pseudo-steady stage was not reached for any of these trials, apparently due to the fast heat transfer between the water in the cooling core and the environment, which was at 37 °C.

The initialization stage usually took 5 minutes to achieve 28.8in Hg vacuum degree. As shown in Figure 4.5, for pump-off test, the water temperature decreased from the starting temperature of 23°C to a much lower temperature of 16.9°C. This fast temperature drop of the cooling core was caused by water evaporation due to the combined effects of vacuuming and vapor adsorption. The temperature of the adsorption core increased significantly at the same time from the starting temperature of 33.5°C to 41.5°C. This was because of the release of adsorption heat. In the transit stage, vacuum pump was disconnected after knotting the tubing of cooling pad. As shown in Figure 4.5, the water temperature continued to decrease to 15.95 °C and then the water side was warmed up gradually and the surface reached 20.64 °C at the end of the one-hour run in the 37 °C incubator.

With respect to the pump-on test, during the initialization stage, the water temperature decreased from the starting temperature of 25.2°C to 16.7°C at the end of the 5-minute period, and then it hit the lowest temperature of 16.5 °C, after which water temperature started to increase. The temperature on the outer surface of the cooling core for pump-on test was 19.3°C at the end of the one-hour test, 1.34°C lower than that of the pump-off test.

It should be mentioned that the cooling core temperature reported was measured by taping the thermal sensor on the outer surface (i.e., the surface facing the ambience) of the cooling core, which was expected to be substantially higher than the water temperature inside the cooling core.

4.4.1.2 Cooling pad with a piece of perforated aluminum foil as a radiation reflector

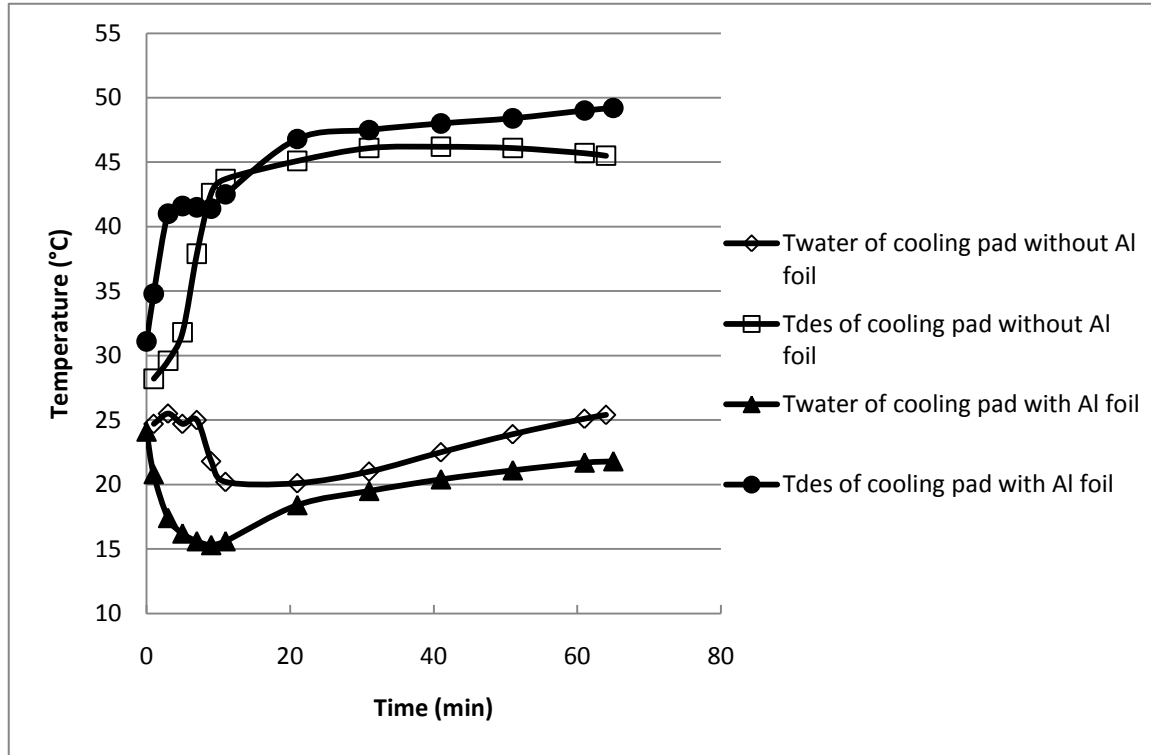


Figure 4.6 Temperature profiles of water and desiccant layer in a typical cooling pad with (solid symbols) and without aluminum foil (empty symbols) built in experiment carried out with pump off after initialization period with 100 g water, 30 g LiCl powder, at a vacuum pressure of 28.8 inHg at 37 °C . (Vacuum pump has been switched off after 5 minutes)

Table 4.1 Average evaporation fluxes of cooling pads with or without aluminum foil (corresponding to the tests shown in the Figure 4.6)

	No foil	Foil
Evaporation flux (g/h/m ²)	511.2	415.7
Cooling capacity (W/m ²)	341.1	277.4

As shown in Figure 4.6, the cooling pad with aluminum foil showed a significantly lower water layer temperature profile comparing to that without aluminum foil, while the two different cooling pads had similar desiccant layer temperature profiles. With-foil cooling pads were able to achieve a final temperature as low as 21.8 °C, while the without-foil pads had a final temperature of 25.4 °C for water side. Table 4.1 gives the water evaporation fluxes for the two tests under different conditions. The with-foil pads had less evaporation flux of 415.7 g/h/m² comparing to that of without-foil cooling pads, which were 511.2 g/h/m².

4.4.1.3 Influence of thickness of spacer

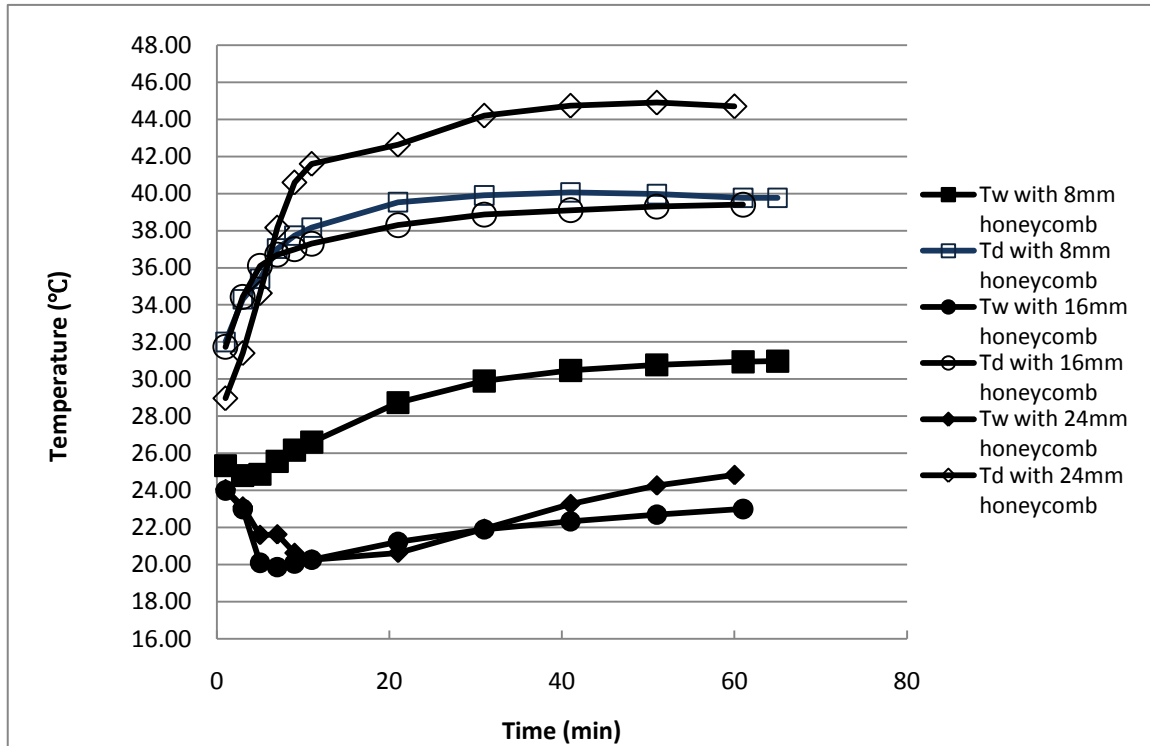


Figure 4.7 Temperature profiles of water and desiccant layer in a typical cooling pad with aluminum foil built in, at different thickness of spacer, pump off experiment with 100 g water, 30 g LiCl powder, at a vacuum pressure of 28.8 inHg at 37 °C. (Vacuum pump was switched off after 5 minutes)

As shown in Figure 4.7, different thicknesses of polypropylene honeycombs with 3.5-20 lb/ft³ density range were tested: 8mm, 16mm and 24mm. Among those data, the best temperature profile (lowest temperature of 19.87°C, final temperature of 23°C was achieved by using the honeycomb with a thickness of 16 mm. The honeycomb of 8 mm thickness performed obviously the worst, while the test with honeycomb of 24 mm thickness had slightly worse temperature profile comparing to that of the 16 mm honeycomb spacer.

4.4.2 Human subject test

4.4.2.1 Cooling performance of the AVMEC garment

In order to evaluate the performance of AVMEC cooling garment, control test data is need. We chose the same person to perform all the human subject tests to minimize the systematic error as the metabolic rate varies with different human subject. The control test data was taken under the same working load and within the same environment (controlled by a thermal chamber) except wearing shorts only instead of the AVMEC cooling garment.

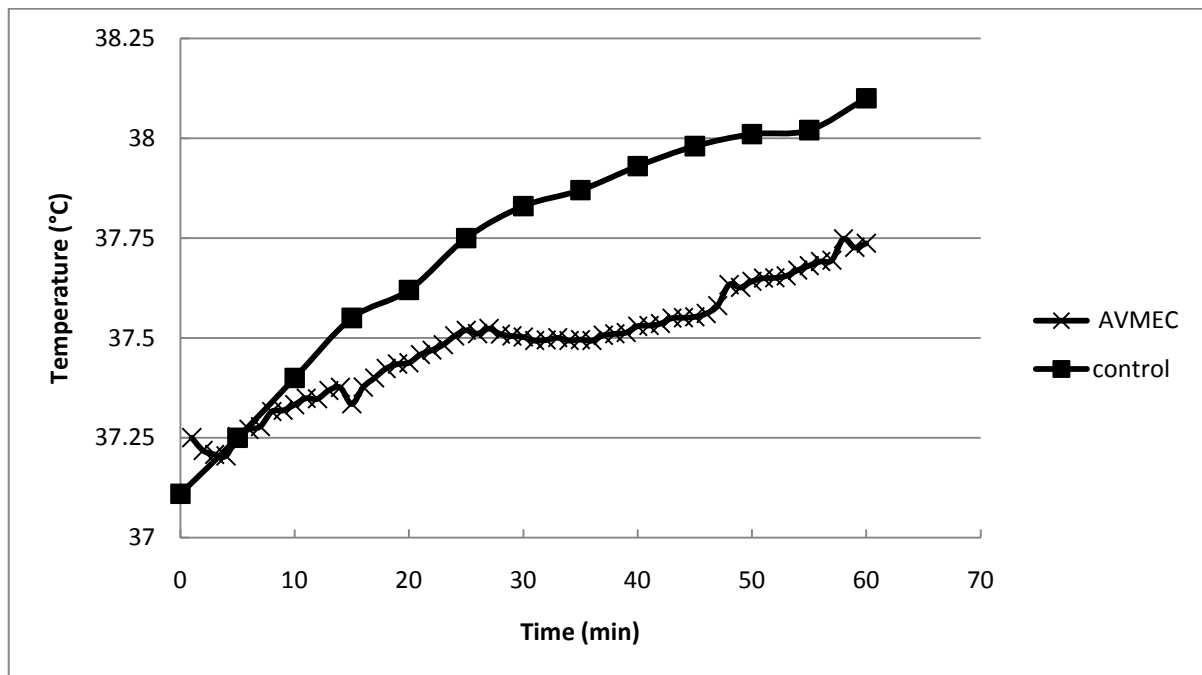


Figure 4.8 Comparison of human subject's control test data and cooling garment data for the same person

As shown in Figure 4.8, the core temperature of the human subject, who wore shorts only during the control tests, was 37.1 °C in the beginning and increased steeply to 37.8 at 30 minutes and continued to increase at a lower rate up to 38.4°C at the end of the 100-minute

test, which was very close to the limit of 38.5 °C beyond which the subject's health would be adversely affected. When the subject wore the cooling garment, the core temperature started from 37.25 °C and increased at a much low rate than that of the control to reach 37.5 °C at 30 minutes. The core temperature of the subject was 38.1 °C at the end of the 90-minute test. The different starting core temperatures of the subject wearing the cooling garment or not (the control test) were different because people's metabolic rate varies at different times. The results show that even though started at a core temperature 0.3 °C higher than that of the control, the subject core temperature was 0.2 °C lower at the end of the 90-minute test when wearing the cooling garment, which was quite significant in the core body temperature changing scale.

4.4.2.2 Cooling performance with NWBC suit

In order to have a better understanding to the performance of the AVMEC cooling garment, we chose the commercial ice-pad vest named Climatech CM2000 (Clima Tech Safety, US) for comparison. The weight of the vest was 4.1 kg which was similar to our AVMEC cooling garment. All the comparison tests in this section were performed by the same person. However, tests were carried out with wearing a nuclear warfare biological chemical protective suit (NWBC suit). Two different control data were collected, they are the subject wearing shorts and T-shirt only, or wearing shorts, T-shirts plus a NWBC suit.

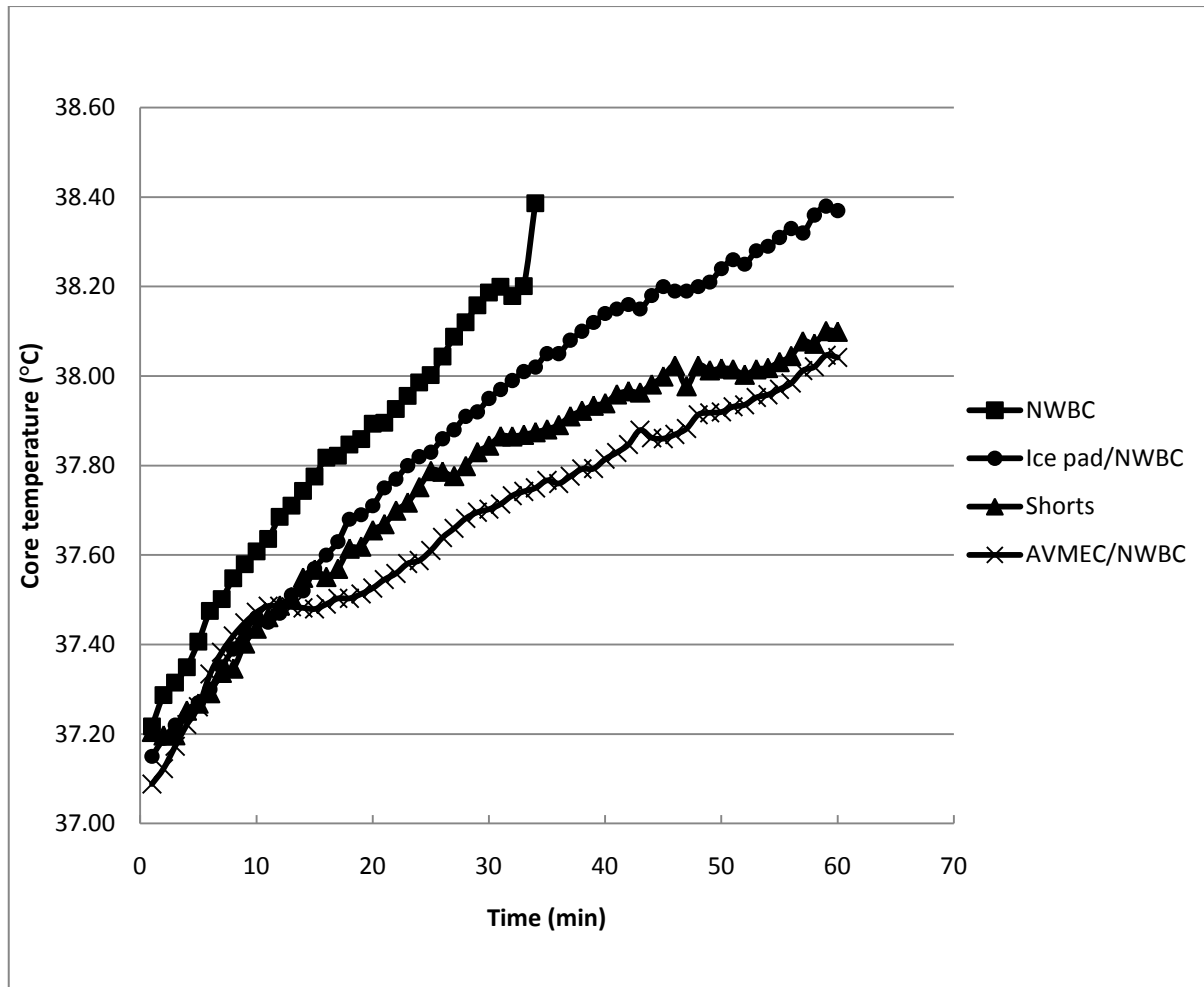


Figure 4.9 Comparison of ice pad vest data with AVMEC cooling garment data based on NWBC suits on tests

As shown in Figure 4.9, the NWBC suit trial only lasted for about 38 minutes as the core body temperature was approaching 38.5°C, the maximum core body temperature recommended to keep the test subject safe. Ice-pad vest data shows significant cooling effect. By the end of 38 min, it kept the core body temperature at 38°C, which was 0.4°C lower than that of the NWBC suit only test. By the end of the one-hour test, it still helped keep the core body temperature at 38.3°C. When the subject wore the AVMEC garment, the data from Fig 4.9 shows a much better core body temperature profile comparing to the ice-pad vest data. It

manages to maintain core body temperature as low as 38.05°C by the end of the one-hour test. That is 0.25°C lower than the ice-pad vest data. These results show that the AVMEC cooling pad is more efficient than the commercial ice-pad vest in maintaining the core body temperature during the one-hour test under the specific protocol.

4.5 Discussion

4.5.1 Unpowered adsorption vacuum membrane evaporative cooling

Conventional vacuum evaporative requires a vacuum pump to remove water vapour from water surface to maintain the driving force for evaporation. In AVMEC, however, vacuum pump is required only for the short initialization period in which vacuum is generated. The maintenance of vacuum in the process is achieved by vapour adsorption. The temperature profiles shown in Figure 4.5 indicate that continuous vacuum only generated marginal difference in comparison to that when vacuum pump was disconnected after the 5-minute initialization, which had a final water side temperature 1.3°C higher than the pump-on test. Correspondingly, the water vapor flux for pump-on test was 50.8 g/h·m² larger than that of the pump off tests. This difference between vacuum pump-on test and pump-off test can be tentatively explained by that the vacuum pump served as a supplement to desiccant adsorption to prevent any vapor accumulation on the surface of desiccant which assured the largest vapor partial pressure difference, resulted the largest evaporation rate.

Considering that the pump off cooling allows the cooling pad to work unpowered and hence be man-portable, and that the temperature profile was not very different comparing to the

pump-on test data, this compromise would be tolerable. Even with those compromise, the cooling pad could still maintain a temperature difference of 27.4 °C between the cooling core and the adsorption core at the end of an one-hour test with the water side surface temperature 15.2 °C lower than the ambient temperature (37 °C). The evaporation flux was 454.5 g/h/m² which corresponded to a cooling capacity of 303.2 W/m² if the internal heat transfer between the adsorption core and the cooling core was negligible.

These results also confirmed that the driving force of water evaporation in the cooling pad was the vapor pressure difference between the surface of water and that of desiccant. After the pump was disconnected from the cooling pad after 5 minutes of initialization, the water side temperature continued to be maintained in a relatively low level comparing to the an environment temperature of 37 °C, indicating continuous water evaporation throughout the entire trial. The vapor pressure difference between the two cores has driven evaporation during the whole process, resulting in temperature decrease at the water core due to the latent heat loss of evaporation. This cooling pad can provide cooling effect without any pre-cooling treatment or external energy supply except in the 5-minute initialization period. When the evaporation rate at the cooling core slowed down due to the decrease of the partial pressure difference, the desiccants in the adsorption core were approaching saturation, the performance of the cooling pad decreased gradually. The regeneration of desiccant and recharge of the cooling garment will be studied in the future.

4.5.2 Internal heat transfer between the cooling and adsorption cores

The results shown in Figure 4.6 indicate that adding an aluminum foil on top of the membrane of the water bag could significantly improve the temperature profile of the water in the whole course of the one-hour test while the water evaporation flux was significantly reduced in comparison to that without the aluminum foil. This is because the aluminum foil, by reflecting away the radiation from the hot desiccant, which was in the close proximity of the cold, significantly reduced the internal heat transfer between the two components of the cooling pad.

The reduced water evaporation when aluminum foil was added was because the minimized internal heat transfer reduced the water temperature, which led to smaller driving force for water evaporation and therefore less evaporation flux. This also implies that the larger water flux when no aluminum foil was included did not represent elevated cooling capacity because part of the heat absorbed by water was transferred from the desiccant by radiation.

Heat is transferred between a hot subject and a cold one by means of radiation, convection and conduction. Since a vacuum space separated the hot desiccant and the cold water, the conduction is limited to that through the matrix of the spacer and the convection became one direction from the water to the desiccant, in alignment with the direction of mass transfer. Therefore, the primary means of heat transfer in the cooling pad was radiation, which was minimized when the perforated aluminum foil was included. The 3.6°C difference between the with foil and without foil data for final temperature of cooling core (shown in Fig. 4.6) has proved that the radiation is essential for the internal heat transfer.

4.5.3 Effects of the thickness of spacer

Another means of heat transfer is conduction. It occurs by direct contact of objects with a temperature difference. In this study, there are two sources of heat from the water perspective, the environment and the adsorption core. We would like to maximize the heat transfer between the water and the environment but minimize the internal heat transfer between the cooling core and the adsorption core. As conduction is highly dependent on material conductivity (Flouris and Cheung 2006), we choose polypropylene honeycomb from PLASCORE which has very low thermal conductivity. Also according to the structure of honeycomb, it formed by stacked polypropylene tubing which has fairly thin wall, so it has small contact surface facing both desiccant side and water side.

Minimizing the contact surface is not enough. The distance between two layers may also affect the temperature profile at water layer. A few tests were carried out in an incubator at 37°C with honeycomb of different thicknesses. By increasing the thickness of honeycomb, it increased the distance between the two cores, which can possible help to slow down the conduction through honeycomb, and therefore, delivered better water side temperature profile. Sets of data have been shown in Figure 4.7. Because of the conduction through the contact of honeycomb material, the shorter distance for conduction the better heat transfer, resulting the worse temperature profile on water side. The fact that the honeycomb with 24 mm thickness performed slight worse comparing to the 16 mm one can be explained by the slower mass transfer rate due to the slower adsorption rate at desiccant side, causing by the longer distance for vapour to pass through inside of spacer.

In order to keep the best water side temperature profile, the spacer with 16 mm thickness has been chosen to be the right cooling pad spacer for building the unpowered man-portable cooling garment.

4.5.4 Feasibility of the AVMEC garment for personal cooling

The energy expenditure is 290 kilocalories per hour for a 70 kg man in active exercise. Assuming that 70% of the energy expenditure is converted to heat, this leads to a metabolic heat production rate of 236 W (Guyton and Hall 2000). Assuming the total effective cooling area of a human body is 1 m², the desired cooling capacity is over 236 W/ m². As shown in Table 4.1, a cooling capacity of 277 W/M² was achieved with a typical cooling pad, indicating that this technology could provide sufficient cooling capacity for personal cooling. The subject test results as shown in Figure 4.7 confirmed it. Indeed, Figure 4.8 shows that AVMEC garment has better performance than the commercial ice vest garment when doing the same setting experiments, while the AVMEC has the same weight and similar structure (vest) with commercial ice vest garment.

4.6 Conclusions

A personal-portable cooling garment based on AVMEC technology was developed. Cooling pads were shown to have a cooling capacity of 277.4 W/m². Experiments regarding the influence of continuous vacuum, the influence of adding the aluminum foil at water side and the influence of spacer thickness were tested. It was proven that continuous vacuuming after initialization period is not necessary. Adding a layer of aluminum foil drastically improved the water side temperature profile as it reflected away most of the radiation from desiccant side. A spacer of 16 mm was chosen to be the proper spacer thickness, as it could slow down

the heat transfer through conduction by the wall of the honeycomb while ensuring the mass transfer to be fast enough. The human subject test showed that the AVMEC garment could achieve a cooling capacity of 272 W/m^2 , which was above the design requirement of a personal cooling garment.

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CHAPTER 5

Conclusion and recommendation

This concept has been implemented under controlled conditions in a desiccator for concept proven tests then prototype cooling pads tests, and eventually human subject tests. Lithium Chloride and PTFE20 (Teflon with a $0.2\mu\text{m}$ pore size) were selected as the desiccant and membrane for building the prototype based on previous researcher on the project.

Based on sets of concept proven tests, within the desiccator, the best desiccant amount was 30g with an initial water mass of 100 g, at a vacuum pressure of 28.8 In Hg, as it can achieve the average flux of 136.4g/h/m^2 during 4 hour run at $25\text{ }^\circ\text{C}$. And 28.8 In Hg which is the highest vacuum degree the EW-07061 pump can achieve was chosen to be the desired vacuum for AEVEC system simply because it can generate the largest vapor pressure difference between the surfaces of water side and desiccant side.

The influence of adsorption area has been studied as well, the run with AD/EV (adsorption area versus evaporation area) =1 has the lowest final temperature at the end of one hour run. But the run with AD/EV=1.4 has the largest evaporation flux. This can be explained that a larger desiccant surface area can facilitate the adsorption process by eliminating the vapor content accumulation at the desiccant surface, while the more vapor absorbed in desiccant layer leading to the more adsorption heat release, resulting potentially more heat transfer from the warm side (desiccant layer) to the cold side(water later) which affect the temperature profile. Thus, the AD/EV=1 is the best choice, and it can achieve a flux of 262.7g/h/m^2 for one hour run.

The role of vacuum and desiccant in the heat and mass transfer process has been carefully studied. Vacuum was found to be crucial for the initial temperature drop during initialization period as well as for internal insulation between the desiccant and water layers due to minimizing convection. Mass transport is governed by the difference of vapor pressure at the surfaces of two layers. Once vapor starts to be absorbed in the desiccant layer, desiccant-water temperature difference sets out. The desiccant sustains the evaporative process in this manner while maintaining the water temperature at a low value. This was confirmed by the previous researcher about the fact that continuous vacuum affects little the cooling performance (a flux and a capacity decrease of 4g/h/m^2 and 2W/m^2 respectively). Applying an ongoing vacuum did not favour vapour's migration rather than the desiccant, which proves that the desiccant is the essential driving force for water evaporation.

The influence of a layer of aluminum foil for reflecting radiation on the water side was also investigated in terms of eliminating heat transfer from inside of system. A big difference at final water side temperature has shown from tests. There are $4.2\text{ }^\circ\text{C}$ differences between the final temperature for a 3-hour with-foil and without-foil trials, on contrast, there is a 56g/h/m^2 flux reduction for with-foil test and 38W/m^2 in energy lose from evaporation process respectively. The reason here for the run with a lower evaporation is that the aluminum foil has partly block the water evaporation surface resulting a smaller effective evaporation area, and the run with less evaporation mass has the lower final water side temperature is because the aluminum foil reflects most of the radiation from the hot subject (desiccant side). It also proves that radiation is the major heat transfer source inside of system. To achieve a better cooling performance overall, it is worth to compromise and adding the layer of aluminum foil can efficiently improve the result.

The cooling pads tests were successful in providing a cooling capacity of 277.4 W/m^2 within the 37°C ambient environment (incubator). As for the aimed criteria of 236 W of cooling, a garment covering one square meter of body surface, the cooling pack can achieve a potential 277 W heat removal. The cooling packs are now fairly small but scaled up to the size of torso garments, with appropriate amounts of liquid water under adequate vacuum level, cooling can be assumed to reach fairly good levels.

The overall experiments and results obtained can confirm that the concept of a three layer desiccant facilitated vacuum evaporative cooling was validated. Further investigation by subsequent fellow collaborators has showed that clamping the tubing (with a screw clamp) resolved the air leakage from that entry point. Thus air tightness can be achieved inside the designed garment. Considering the importance of insulation for preventing backflow of heat at the water layer, an alternative allowing ongoing vacuum at low cost (using miniature vacuum pumps) should be compared to the performance of an airtight garment in terms of cooling usage duration and capacity provided.

One of the next phases to pursue in assessing the cooling effect of this novel cooling pack is to perform human testing, with subjects wearing garments based on the developed three-layer cooling prototype. Measuring the thermoregulatory responses of these subjects under heat stress will be a definitive way of quantifying the cooling performance of the cooling packs.

This new cooling concept can also be applied to small portable coolers and heat sinks for vehicles.

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APPENDIX

Theoretical approach

A simply theoretical approach for AEVEC system based on thermodynamic principles has been studied. The latent heat of water evaporation at 28.8 In Hg pressure, at room temperature is 2400KJ/Kg. The amount of cooling capacity is equal to the amount of heat removal from evaporation, which can be calculated as

$$Q=mC_{\text{subject}}\Delta T=\Delta m_{\text{evaporation}}h_{\text{latent}}$$

The basic equations of the mass and heat transfer which occurs in the systems using evaporative cooling are listed below, the main assumption are:

1. The heat exchange between the desiccators and the surrounding is negligible;
2. The specific heats of the fluids are assumed to be constant;
3. The mass and heat transfers take place only in the direction normal to the flow;
4. The water and vapor flows are uniformly distributed in the water dish and desiccant dish perpendicular to the flow;

Evaporation process mass transfer

The two main mass transfer mechanisms are convection and diffusion, those two often occur simultaneously and with heat transfer. As the system is operated under vacuum, we assume that convection is negligible.

The basic relation for molecular diffusion is Fick's Law:

$$J_A = -D_{AB} * \frac{dC_A}{dz}$$

Where

J_A Molar flux density (mol A/m²S)

C_A Molar concentration (mol A/m³)

D_{AB} Molar diffusivity (m²/s)

The diffusion coefficient depends on three key factors: pressure, temperature and composition of medium. For this system, it operates not at normal pressure and normal boiling point for water to evaporate, in that case, the pressure and temperature correction equation need to be applied:

$$D_{AB,T_2,P_2} = D_{AB,T_1,P_1} \frac{P_1}{P_2} \left(\frac{T_2}{T_1} \right)^{1.5} \frac{\Omega_{T1}}{\Omega_{T2}}$$

Where

P_1 atmospheric pressure, 1 atm

P_2 operating pressure, 29.9-28.8=1.1 InHg = 0.0374atm

T_1 room temperature, 298.15 K

T_2 operating temperature for water side, 288.15 K

Ω_{T_1, T_2} collision integral based on Lennard-Jones potential

$$(KT_1/e = 0.36831 \rightarrow \Omega_{T_1} = 2.4; KT_2/e = 0.35599 \rightarrow \Omega_{T_2} = 2.474)$$

Therefore,

$$D_{AB, T_2, P_2} = 6.23 * \frac{10^{-4} \text{m}^2}{\text{S}}$$

By getting the system diffusion coefficient, dC_A can be solved by

$$C_A = \frac{\rho_A}{M_A} = \frac{P_{vap}}{RT}$$

Assuming water vapour is ideal gas,

P_{vap} Saturation vapour pressure at system condition, pa

T System operating temperature, K

R Ideal gas constant, $8.314 \text{ J} \cdot \text{K}^{-1} \cdot \text{mol}^{-1}$

Vapour pressure at the water surface can be calculated by following equation

$$P = \exp(20.386 - 5132/T)$$

Where,

P Vapor pressure in mmHg

T The temperature in Kelvin

Therefore, choosing the final pseudo steady state temperature at water sides which is 15°C as the water side temperature, P is then calculated as 13.02 mmHg which is also 1736 Pa. And from literature, the average vapor pressure of water at 30°C at the surface of saturated Lithium Chloride is 2.93mmHg, which can be converted to 390.6 Pa(Gokcen 1951). There is 8.5 cm distance between the surface of water and surface of desiccant

Therefore,

$$\frac{dC_A}{dZ} = -\frac{6.704\text{mol}}{\text{m}^4}$$

Then, the theoretical evaporation flux can be calculated as

$$J = -D \frac{dC_A}{dZ} = D_{AB,T_2,P_2} \frac{dC_A}{dZ} = \frac{0.00418\text{mol}}{\text{m}^2\text{s}} = \frac{270.86\text{g}}{\text{m}^2\text{h}}$$

Table 3.4 Experimental data of fluxes and cooling capacities for different amount of desiccant with 100g water tests for one hour based on $1.81 \times 10^{-2} \text{m}^2$ evaporation area

1 hour		
Desiccant (g)	Flux (g h ⁻¹ m ⁻²)	Q (W/m ⁻²)
2.5	234.8	156.663
5	251.4	167.722
10	262.4	175.094
30	268.0	178.78