



Department of Architecture & Built Environment

**Investigation of a Novel Liquid Desiccant Wheel
Cooling System**

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Degree of Doctor of Philosophy**

Abstract

The energy consumption in cooling and forced ventilation, especially in a hot and humid climate, is widely significant and concerning. Nevertheless, the growing demand for air conditioning even in temperate climates is causing a substantial increase in energy usage and resulting in a rise in greenhouse gasses emission. It has driven the exploration of new air conditioning technologies, such as a hybrid desiccant cooling system, to save energy and reduce greenhouse gasses emission. One of the methods of a hybrid desiccant cooling system is to utilize the advantages of a wheel mechanism with liquid desiccant into one integrated system. This prototype has been developed to employ more efficient dehumidification as well as better cooling capabilities. The in-house experiment and computational simulation were used to evaluate the performance of this prototype. Based on the analysis, the coefficient of performance (*COP*) of this system is higher than the conventional system, and the potential can be explored for more improvement. Based on the results, the *COP* of the system obtained is 7.547. Then in terms of economic and environmental analysis, the system also shows a tremendous impact. In economic analysis, it shows that it only required 2.72 years to earn its investment payback. In environmental analysis, the Eco-indicator 95 tools have been used, and it shows around 0.269 points which relatively low and would have minimal impact on the environment. Even though the capital and maintenance cost of the novel liquid desiccant wheel cooling system is much higher than the conventional HVAC system, but its running cost is relatively lower. Thus, the novel wheel liquid desiccant cooling system is cost-effective and environmentally friendly.

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Nomenclature

Symbol	Description, Units
A	Cross sectional area, m ²
C_p	Specific heat capacity, kJ/kg K
C_{Pa}	Specific heat of dry air, kJ/kg K
C_{Pv}	Specific heat of water vapour, kJ/kg K
C_{Pw}	Specific heat of dry air, kJ/kg K
C_{pf}	Specific heat of wall material, kJ/kg K
C_{pd}	Specific heat of desiccant, kJ/kg K
$C_{P,sol}$	Heat capacity of desiccant solution, kJ/kg K
CC	Cooling capacity, W
COP_{cool}	Coefficient of performance of cooling section
COP_{heat}	Coefficient of performance of heating section
D	Wheel diameter, mm
E_{heat}	Energy consumption from heating section, kJ/kg
E_{cool}	Energy consumption from cooling section, kJ/kg
g	Gravitational acceleration, m/s ²
h	Enthalpy, kJ/kg
$\dot{m}$	Mass flow rate, kg/s
$\dot{m}_{sol}$	Mass flow rate of desiccant solution, kg/s
P	Pressure, N/m ²

p_{fan}	Power consumed by fans, W
p_{motor}	Power consumed by motor, W
P_{Vi}	Vapour pressure on the desiccant surface, Pa
Q	Latent heat of vaporization, kJ/kg _{water}
RH	Relative humidity, %
Re	Reynolds number
t	Time, s
T	Temperature, °C
T_{ai}	Temperature of air in cell no i, K
T_{Si}	Temperature of desiccant in cell no i , K
V	Air velocity, m/s
v_{Hi}	Specific volume of humid air, m ³ /kg
W	Electrical energy consumed by the system, W

Greek Symbol

β	Perimeter of square slot, m
μ	Dynamic viscosity of the fluid , Pa·s or N·s/m ² or kg/(m·s)

Subscripts

a	Air
i	Iteration
sol	Solution

<i>S</i>	Desiccant wall
<i>wheel</i>	Wheel desiccant
<i>motor</i>	Gear motor
<i>cool</i>	Cooling section
heat	Heating section

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Chapter 1

Introduction

1.1 Background

Cooling is essential for space conditioning of most buildings in warm, humid climates. Removal of moisture from the air represents a considerable portion of the air conditioning load. Conventionally, air conditioning systems have to lower the air temperature below its dew point to accomplish dehumidification. This act will result in wet cooling coil surface that leads to the growth of mould and bacteria, which could lead to unwanted health issues and bad indoor air quality in conditioned spaces (Ge et al., 2011). After that, the air is reheated to reach the desired comfort level. Following this traditional process consumes extra energy and increases peak electricity demands along with the growth of cooling loads in modern buildings with low thermal inertia, the peak electricity demand increases during the day. It causes problems concerning the energy supply in some countries, as the demand for air conditioning coincides with the maximum demand of industry and households. The inertia of energy plants leads to high consumption of coal, oil, gas, and nuclear energy by energy companies to ensure the provision of electricity (Gerrit Höfker et al. 2014). It means that a conventional air conditioning system has a severe issue in terms of inefficient removal of latent loads. However, they are very effective in removing sensible load within conditioned spaces.

These disadvantages of conventional air conditioning can be resolved by a desiccant cooling system (Shahb Alizadeh, 2008). Solar air conditioning has the highest technical levels whose advantages lie in its great match with seasons and weather. The greater the solar radiation, the

more massive refrigeration output the solar radiation could produce (Wang R.Z, 2009). Desiccant cooling systems offer a solution to meet the humidity and temperature requirements of buildings via decoupling latent and sensible loads. The desiccant cooling system is also considered as a suitable option to replace conventional air conditioning where humidity control is the primary concern, such as supermarkets, greenhouses, and clean rooms (Jagdeep Singh 2013). This humidity control can be achieved by using desiccants to humidify the air. Desiccant materials have a strong attraction to the presence of water vapour in the air. The driving force for this absorption of water vapour is due to the vapour pressure difference between the air and desiccant material (Sanjeev Jain et al., 2011). There are three types of thermally activated desiccant cooling technologies – solid desiccant, liquid desiccant, and hybrid desiccant. Classification of the desiccant-based air dehumidification and cooling systems are shown in Figure 1.1 (Enteria & Mizutani, 2011).

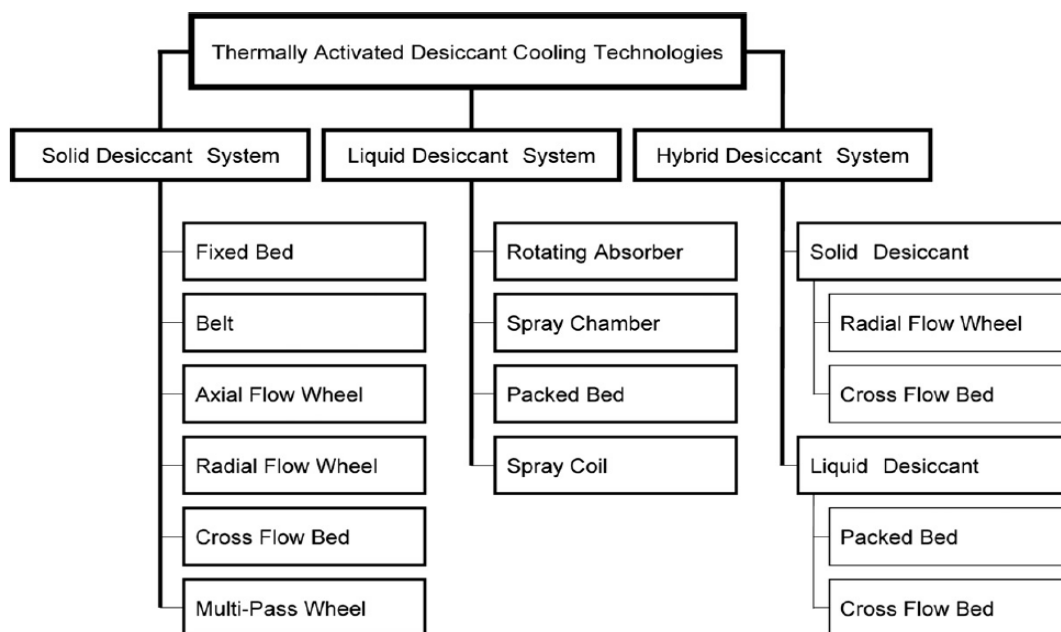


Figure 1.1 Classification of the thermally activated desiccant cooling technologies (Enteria & Mizutani, 2011)

The main components of a liquid desiccant dehumidification system are the dehumidifier and the regenerator. The desiccant material that has been in contact with the air absorbs the moisture from the air and dilute it in the dehumidification process. After that, the desiccant must be regenerated to the right level of concentration so that it can reabsorb moisture in dehumidification (Yanggao Yin et al., 2007). The minimum temperature required for regeneration depends on the nature of the desiccant material. It varies from 50 °C for lithium chloride to 60 °C for silica gel (Bourdoukan et al., 2008). The desiccant can be regenerated by low-grade heat with low temperatures, such as solar energy and industrial exhaust gas. For the economic aspect, we selected solar energy, such as solar thermal, because it is one of the most abundant energy sources that causes no environmental pollution. (VV Tyagia et al., 2012). One way to collect solar thermal energy is by using a solar thermal collector. Solar thermal energy collectors are special kind of heat exchangers that convert solar radiation into thermal energy through a transport medium or moving fluid. It absorbs the incoming solar radiation and converts it into heat energy. This heat energy is transferred through a fluid medium (usually air, water, or oil) for useful purpose/applications and this case, regenerating desiccant material.

Liquid desiccant air dehumidification had gained much progress recently because of the coherent qualities of this technology (Zhang LZ, 2008 & Grossman G, 2004). It is highly efficient, and it has no liquid water condensations. Packed beds, in which the liquid desiccant and the airstream exchanged heat and moisture indirect contact, were the traditional equipment used to realise air dehumidification. However, since the liquid desiccant and the process air were in a direct contact, small liquid desiccant droplets could be carried over by the processed air to an indoor environment, which was rather harmful to occupants' health, building structure, and furniture. Crossover problems had become the major problems for liquid desiccant air dehumidification, especially in engineering applications. A selectively permeable membrane had been combined with the liquid desiccant to form a new kind of liquid desiccant air

dehumidification system (Zhang L.Z, 2012). It was called a membrane-based liquid desiccant air dehumidification system. The most apparent difference between this technology and a traditional one was that the traditional packing columns, either the dehumidifiers or the regenerators, were replaced by membrane-based modules. In these modules, semi-permeable membranes were used to separate the desiccant solution and the process air from each other. The membranes were able to prevent the solution from crossing over to the process air, while selectively permitting the transport of heat and moisture between the solution stream and the air stream (Si Min Huang, 2013). Despite being a relatively new technology, there were many new developments in this area in the past ten years. These developments were focused on three aspects, which are membrane materials, membrane module store place packed beds, and system design and applications.

The solar liquid desiccant air conditioning system has the following advantages compared to the solid desiccant system (Xie X.Y., 2008):

1. Because of the fluidness of the liquid desiccants, it is easier to remove the latent heat to perform the isothermal dehumidification.
2. It has more potential for storing chemical energy.
3. It can improve air quality by absorbing dust, bacteria, and so on.

However, the existing liquid desiccants, such as lithium bromide (LiBr), lithium chloride (LiCl), and calcium chloride (CaCl₂), are corrosive to the metals, so the popularisation of the solar liquid desiccant air conditioning system has been limited. Thus, it is critical to find more suitable desiccants to substitute the existing desiccants. Ionic liquids (ILs) are molten salt at room temperature, consisting of organic cation and inorganic anion (Deng Y.Q, 2006). Compared to some organic desiccants, ILs are generally non-combustible, have good thermal

and chemical stability, and do not evaporate under normal conditions. Compared to the salt substances, ILs have good fluidity over a wide temperature range and have no or very low corrosion to metals. Until now, it has been found that [Emim]BF₄ is a possible desiccant (Luo YM 2011). However, because of the diversity of ILs and relatively fewer literature about using them in the field of dehumidification, it isn't easy to find suitable ILs.

A comprehensive literature review was carried out, and a patent search was conducted for liquid desiccant systems design, structure, and operation. No wheel-based liquid desiccant dehumidification system has been presented up to date. Thus, the unique, innovative proposed unit would blend the compact and simple design of solid desiccant wheels in addition to the advantages provided by a liquid desiccant.

1.2 Aims and Objectives of the Project

Based on the desiccant dehumidification process described in the previous section, the aim of this project is:

1. To obtain a better and higher coefficient of performance (COP) than the conventional vapour compression cycle.
2. By making it simple, cheap, and environmentally friendly without compromising the system's efficiency
3. Having a good heat and mass exchange structure and able to reduce liquid desiccant carry-over in the air stream.

The proposed novel liquid desiccant wheel cooling system can meet the above characteristics. This system will integrate with the attributes of both solid and liquid desiccant systems. The wheel is made from a material that can execute heat and mass transfer within the system. The size of the system can be reduced considerably by eliminating the need for bulky components, such as a packed bed or membrane, and replacing them with simple and compact absorption wheel. The system also excludes the need for vapour compression cycle, as the proposed system only needs a supply of tap water and dew point evaporative cooler. Nevertheless, the advantages provided by the solid and liquid desiccant system can still be retained by using the proposed novel wheel liquid desiccant cooling system.

The objectives of the proposed system are shown as follows:

1. Conducting extensive literature studies on desiccant cooling and integrate the best features of the rotor wheel and liquid desiccant into a hybrid system.
2. Optimising wheel sizing and predicting the final output of the system by using the developed mathematical model.
3. Conducting the experimental work to study the physical and chemical behaviours of the system by using full-scale laboratory testing.
4. Analysing the specific experimental results such as humidity and compare it with the simulation results that were done earlier.
5. Conducting the economic and environmental analysis so that the system is viable to the real engineering application.

1.3 Structure of the Thesis

The research involved in this thesis was divided into different chapters and sections. The thesis report comprises of seven chapters, and some other relevant information has been provided in appendices. A brief outline of the thesis structure is given below.

Chapter 1:

Presents an introduction and background of the research, defining the problems, aims, and objectives of the study. The chapter concludes with the structure of the thesis.

Chapter 2:

This chapter contains a review of the technologies applied to this research. It involved the mechanism of both liquid and solid desiccant cooling technologies. Different technological arrangements of desiccant have been reviewed, and the advantages of technologies over the conventional compression system have been highlighted.

Chapter 3:

This chapter discusses the theoretical background, wheel material selection and evaluation process. The potential desiccant selection has also been discussed thoroughly in this chapter based on literature studies. This chapter also discusses the choice of the body framework of the system and its suitability for this experiment.

Chapter 4:

This chapter presents the results of the simulation for the novel wheel liquid desiccant cooling system. The simulation investigated the proper sizing for the rotor wheel and the effect of various parameters on the airflow around the wheel. The obtained simulation results were

significant and crucial for deciding the proper design for the novel liquid desiccant wheel cooling system.

Chapter 5:

This chapter describes the prototype design and experimental rig setup. The chapter also describes important experimental facility components, such as data logging and instrumentation. Details of air airflow measurements air channels, such as in pipe and wheel, are also discussed. Various parameters measured during the experimental work are also shown in this chapter.

Chapter 6:

Mainly focuses on the experimental results obtained from various experimental investigations carried out on the novel liquid desiccant wheel cooling system. The chapter also presents the test results obtained from the complete prototype setup. All significant results are discussed in this chapter. The obtained experimental results also compared with the one from simulation results for validation purposes.

Chapter 7:

This chapter presents a general discussion of the results, including issues relating to the economic and environmental evaluations to the system.

Chapter 8:

The chapter brings all key conclusions from this study and relates it to the objectives that were set earlier. This chapter ends with a recommendation for further research.

1.4 Summary

This chapter presented an overview of various desiccant cooling systems. The most prevalent methods of air conditioning is those operating on harmful refrigerant material and fossil fuels, which account for a large proportion of the world energy demand. These methods of air conditioning are associated with large emissions of greenhouses gasses with severe environmental consequences, such as global warming and the destruction of the natural ecological system. Solid and liquid desiccant cooling technologies are considered to reduce these emissions and the severity of environmental degradation. The research work undertaken is aimed at minimising the number of emissions to the environment through the provision of sustainable indoor thermal comfort using a combination of various methods of desiccant cooling technology. This chapter also described the novel liquid desiccant wheel cooling system's concept, as well as the aims and objectives of this research project and thesis structure.

Chapter 2

Literature Review of Previous Research on Desiccant Dehumidification Cooling System

2.1 Introduction

This chapter presents a comprehensive review of desiccant dehumidification cooling system. The initial section of this chapter focuses on the basic principle of desiccant dehumidification cooling system, which includes a solid and liquid desiccant. However, for this review, the section will give more emphasis on the liquid desiccant dehumidification cooling system. We will discuss in-depth the mechanism involved in solid and liquid desiccant dehumidification technology used by researchers and engineers all around the world. Besides, we will look into the advantages of both systems and their application for cooling purposes. A piece of sufficient knowledge on these subjects is necessary in order to understand how to design a high-performance and efficient desiccant dehumidification cooling system.

2.2 Desiccant Dehumidification Cooling System

A desiccant is a hygroscopic material capable of attracting and capturing moisture in either liquid or gases until vapour pressure is in equilibrium with the surrounding air. The desiccant adsorption or absorption of moisture occurs because of the difference in water pressure between the surface of the desiccant and the surrounding air. If the desiccant's surface vapour pressure is lower than that of the surrounding air, the desiccant attracts moisture. The desiccant releases moisture if its surface vapour pressure is higher than that of the surrounding

air, which is a process called reactivation or regeneration. It is necessary to drive out the adsorbed/absorbed water vapour from desiccant (regeneration) to enable the desiccant to absorb water vapour in the next cycle. Regeneration of desiccant is usually achieved by the use of thermal energy supplied by electricity, natural gas, waste heat, or the sun, which causes the surface vapour pressure of the desiccant to become higher than that of the surrounding air. However, if the vapour pressure remains high, the desiccant's moisture removal rate is decreased. As a result, the desiccant must be cooled to reduce the vapour pressure and enable the desiccant material to absorb moisture once again. The regeneration temperature of a desiccant material depends on the nature of desiccant used, as different desiccant materials have different regeneration temperatures.

The process of removing moisture from the surrounding air is termed as sorption. Sorption occurs in two manners, depending on whether the desiccant undergoes a chemical change during the process. Adsorption does not change the physical or chemical nature of the desiccant, except by the addition of the mass of water vapour. Absorption, on the other hand, changes the physical or chemical nature of the desiccant (Neti and Wolfe, 2000).

As adsorption is a spontaneous and exothermic process, heat is generated during the process. The generated heat is equal to the latent heat of evaporation plus an additional amount of heat due to change in the surface energy of the desiccant material. This additional heat is called the integral of wetting and the total heat generated during adsorption is called the heat sorption.

Desiccant dehumidification systems can be generally classified as either solid or liquid desiccant systems, and there are many different materials of both types. Silica gel, zeolites, activated alumina, synthetic polymers, lithium bromide, and lithium chloride are examples of

some commercially available desiccant materials that are used solely or combined. However, the use of particular desiccant material depends on the application.

The desiccant materials are used in a different technological arrangement, either as standalone systems or in conjunction with conventional air conditioning systems. A typical configuration employs a slowly rotating wheel of 8–10 revolutions per hour impregnated with the desiccant material, with part of it exposed to the incoming or primary air stream while the rest of it is being regenerated (Wang R.Z., 2009).

Liquid desiccant system arrangement often employs a spray tower in which the aqueous liquid is sprayed into air stream or onto contact surfaces to absorb moisture from a primary air stream. Like solid desiccants, they need to be regenerated after becoming saturated with water vapour. The operation principles described here apply to both liquid and solid systems. Figure 2.1 shows the three process of desiccant systems, which are: the dehumidification process, the cooling process and the regeneration process.

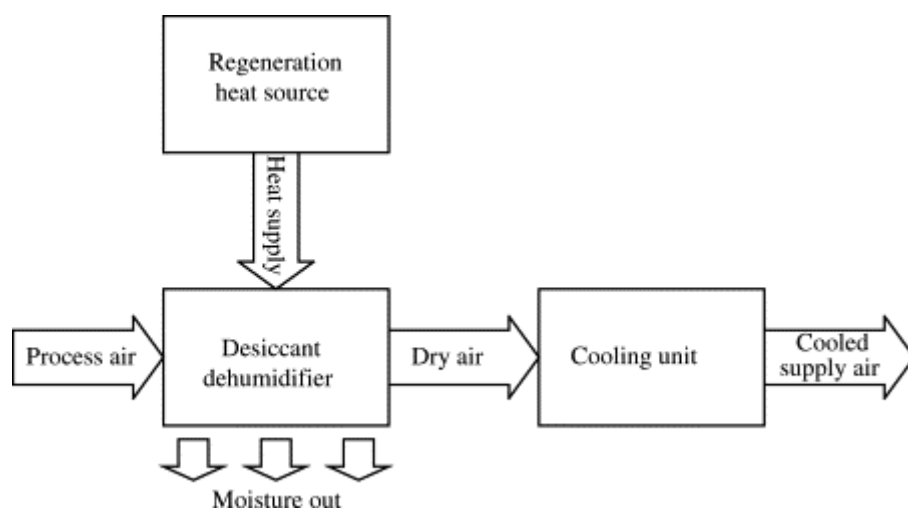


Figure 2.1 Principle of desiccant cooling (Wang et al., 2006)

The consumption of energy by the desiccant system is dependent on the movement of desiccant material through the whole cycle. As the removal of moisture uses relatively little energy, the cooling and the regeneration costs can be high. The energy needed for the cooling process depends on the desiccant material used in the system and on the difference in temperature and after regeneration.

These variables make up the total regeneration energy of a desiccant system (ASHRAE, 1993).

These variables are:

- The heat necessary to raise the desiccant's temperature high enough to make its surface vapour pressure higher than of the surrounding air.
- The heat required to vaporise its moisture content.
- The small amount of heat from desorption of water from the desiccant.

2.2.1 Advantages of Using Desiccant System

When desiccant systems are employed instead of typical air conditioning systems, a significant amount of savings is achieved not only in the initial equipment costs, but also maintenance and energy use. Air quality is also improved through the removal of contaminants and pollutants for primary air stream when desiccant systems are employed (Wang et al., 2009). Desiccant systems are quiet, long lasting, cheap to maintain, and generally environmentally friendly. In addition to the advantage of desiccant systems, a study by Kabeel, 2007 summarized the benefits of using desiccant systems, as follows:

- Desiccant systems consume less electrical energy and permit the use of solar energy and waste energy for its regeneration process.
- It is efficient when the latent heat load is larger than the sensible load.

- It can be used to condition the internal environment of buildings and operate without the use of harmful refrigerant, as it is a clean technology.
- The control of humidity achieved is better than when using vapour compression systems.
- In some cases, the cost of energy to regenerate the desiccant is less when compared with the cost of energy used to dehumidify air by cooling its below-dew point temperature.
- High ventilation achieved by desiccant systems is likely to improve the indoor quality.
- Desiccants have the capability of removing airborne pollutants.

2.2.2 Regions Using Desiccant Systems

Desiccant dehumidification and cooling systems are a suitable option in regions with hot and humid climatic conditions for sustainable thermal comfort. Desiccant-assisted dehumidification and cooling systems perform excellently in these regions because of the large temperature and humidity difference between indoor and outdoor environment. These climatic conditions are typically found in southern Europe, South Asia, South America, and parts of West Africa.

Miller et al. (2002) successfully installed and evaluated the performance of a desiccant-based air conditioning system in Florida, United States. Their investigation shows that the system is capable of delivering a high volume of ventilation air to school in a very humid climate while maintaining very comfortable classroom conditions.

Shahab Alizadeh (2008) conducted an experiment on a prototype solar collector/regeneration liquid desiccant system. He concluded that the system performed satisfactorily under the Australian summer conditions.

A review on the application of desiccant dehumidification and cooling systems conducted by Mavroudaki et al. (2002) shows that technology have been used successfully in northern Europe. They concluded that there were potential to utilise this technology in some parts of southern and most of central Europe. The study summarised the energy savings achieved by using a solar-powered single-stage desiccant cooling system in some European countries.

La et al. (2011) experimentally investigated and theoretically analysed a two-stage solar hybrid desiccant air conditioning system. They concluded that under Beijing (temperate), Shanghai (humid), and Hong Kong (extremely humid) weather conditions, the system can remove about 57%, 69%, and 55% of the seasonal moisture load, thereby reducing electrical power consumption by about 31%, 34%, and 22%, respectively.

Several desiccant dehumidification and cooling systems have been successfully developed and installed for different application in Europe, Australia, the United States, and China, as reported by Dai et al. (2010). There is a large amount of solar energy in these regions. Recently, technologies to improve the efficiency of desiccant dehumidification systems, as well as the utilisation of low-grade heat for regeneration have become the primary concern of research.

2.3 Solid Desiccant Cooling

Solid-based desiccant system uses solid desiccant materials that can hold off water vapour. There are several kinds of solid desiccant materials; they are silica gel, titanium silicates, calcium chloride, activated aluminas, zeolite, molecular sieve, lithium chloride, organic-based desiccants, polymers, as well as compound and composite desiccants. (Abdulrahman et al., 2013)

2.3.1 Basic Principle

This type of desiccant cooling system used solid-based desiccant materials in controlling air moisture content. Air moisture control mechanism can come through by absorption or adsorption. It is a solid, inorganic, crystalline desiccant that is impregnated in inert materials and used in a unit with a honeycomb type heat-wheel. They are widely used for commercial HVAC applications. Solid desiccant systems are available in single-wheel and dual-wheel packaged units. The single wheel units only include a desiccant wheel, cooling coil, and reheat coil. Fresh air from outside is dehumidified by the desiccant wheel. Then, a cooling coil reduces the air temperature to the desired dew point, and the reheat coil raises the air to the desired dry bulb temperature. The dual-wheel units use a two desiccant-based wheel and a sensible-only heat wheel along with the conventional chilled water or direct expansion coil. During the cooling mode, the sensible-only wheel is used as a post-cooler or to reheat the over-cooled dehumidified air. Heat is required to regenerate or reactivate the desiccant material by removing the absorbed water vapour and discharge it outdoors using an air stream. Natural gas and solar energy are usually used to provide the heat for regeneration in single-wheel units. Sometimes, a combination of exhaust air and gas heat may be used on dual-wheel systems.

2.3.2 Developments in Solid Desiccant Cooling

Early research that was conducted by engineers and researchers were mostly focused on the development of solid-based desiccant cooling. There are three major types of solid desiccant, which are a desiccant wheel, desiccant bed, and modified design.

2.3.2.1 Desiccant Wheel

Figure 2.2 shows the Air Handling Unit (AHU) system, which is composed of the two-wheel type or called the Munters Cycle (Halliday et al., 2000). It is an open heat-driven cycle, which comprises a desiccant wheel parallel to a thermal wheel, with a supplementary cooling coil on supply air and evaporative cooler in the return air stream before the thermal wheel. A regeneration coil located in the return air stream subsequently drives the whole cycle. The regeneration coil is usually a low or medium temperature hot water coil, although steam can be used. The author creates the model utilising meteorological data to compare the actual performance of the case study. It was found that during summertime, air temperature from the system can reach 23 °C, with 70% gas energy saving achieved annually.

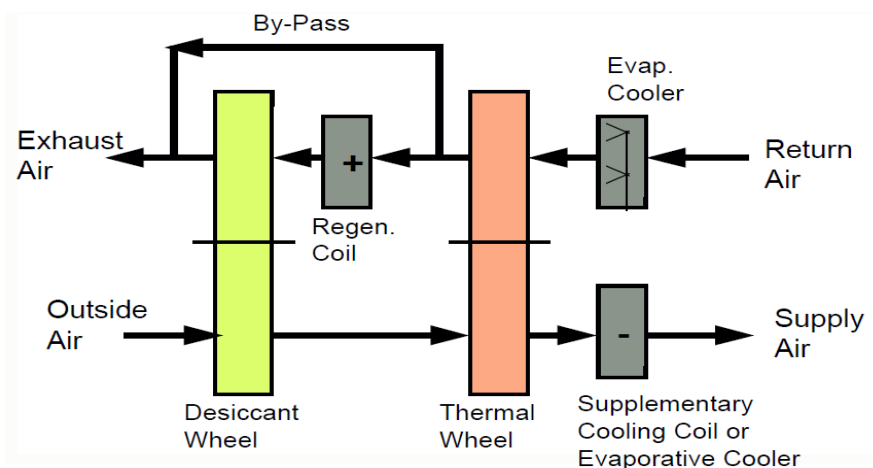


Figure 2.2 Air Handling Unit (AHU) configuration (Halliday et al., 2000)

The application of the desiccant wheel as the air dehumidifier has several elements to be considered. Kang and Maclaine-cross (1989) showed that the performance of the desiccant-based cooling and dehumidification system relies much on the desiccant material in moisture sorption capacity. Kodama et al. (2001) showed that an optimal-speed rotating desiccant wheel can lead to a high sorption rate. The optimal speed increases with increasing regeneration air flow rate, decreasing desiccant wheel depth, and decreasing bulk density of the rotor. The optimal wheel speed decreases with higher humidity and lower regeneration temperature. It also showed that the sorption rate is relative to the air relative humidity. Zhang and Niu (2002) showed that the sorption performance of the desiccant wheel depends on the wheel rotational speed and the number of transfer units. Hence, it was suggested that the desiccant wheel should have a number of transfer units of 2.5. Subramanyan et al. (2004) showed that increasing the air flow rate reduces the specific cooling load (difference in the enthalpy of the outdoor air and of the processed air). The cooling load rises because of the amount of air mass flow rate. In addition, he also showed that increasing the air flow rate increases the supply of air moisture content.

2.3.2.2 Desiccant Bed

Apart from the desiccant wheel type, the fixed-bed solid desiccant cooling system is another type of desiccant cooling system. The advantage of the fix-bed system is the sorption process, which can be done through isothermal means. It means that the temperature of the air, after passing the desiccant material, is not increased. Hence, the sorption process is increased because of the removal of the heat of sorption. Yuan et al. (2008) proposed a cross-cooled compact solid desiccant dehumidifier. The objective of the design is to make the dehumidification process cooler due to the heat exchange between the dehumidified air and the

secondary air. It was shown in the result that the performance of the design is better than without secondary cooling. Figure 2.3 shows the cross-flow fixed-bed-type air dehumidifier with cooler. Majumdar and Worek (1989) investigated the open-cycle cooled-bed desiccant cooling system using the cooled-bed dehumidifier. It shows that the system performance is more sensitive to the regeneration and indoor temperature and to the outdoor humidity ratio than to the indoor humidity ratio and outdoor temperature. It shows that the cooled-bed system can be regenerated at a low temperature of 50 °C. Zhang et al. (2003) studied the dehumidification performances with 10 kinds of solid desiccants, as the wall materials for honeycomb-type adsorbent beds are compared. He found that the *COPs* of various solid desiccants tend to remain unchanged with a higher regeneration temperature under humid operating conditions, but they decrease with regeneration temperature increasing under dry operating conditions. He also concluded that among various desiccants, the silica gels type B, 3A, and RD always perform better than other candidates for air dehumidification under most operating conditions, when low-grade waste heat is used as the driving energy. Henning (2007) showed the application of a fixed-bed through a cyclic operation called evaporative cooled sportive heat exchanger (ECOS).

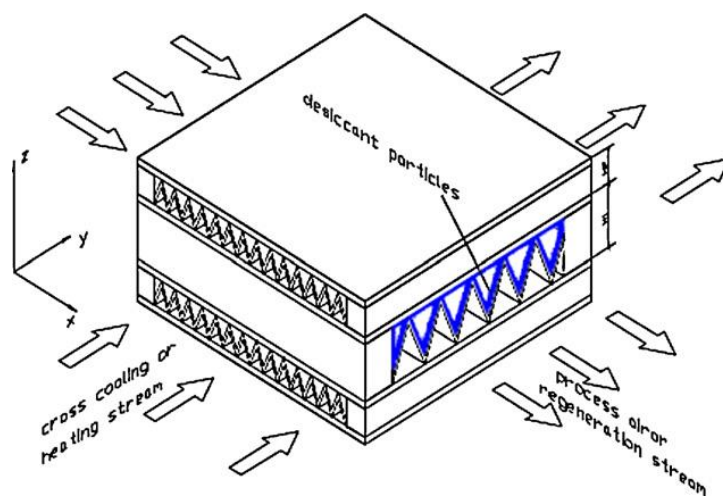


Figure 2.3 Desiccant coated cross-cooled compact dehumidifier (Yuan et al., 2008)

2.3.2.3 Modified Design

Then we came to a part where all designs discussed earlier were modified to meet specific criteria. In most designs, the operation of the solid desiccant cooling system is through a dehumidification-humidification process. With such a process, air dehumidification is done at very low humidity content for the application of evaporative cooling. With such process, the required regeneration temperature is increased. Thereof, constant dehumidification process is another option to avoid the deep dehumidification, which will be applicable to a hot and humid climate. Enteria et al. (2010) presented the constant humidity air cooling cycle of the desiccant as illustrated in Figure 2.4. Ando et al. (2005) showed the double-stage dehumidification process. In the double-stage dehumidification process, two desiccant wheels are employed as shown in Figure 2.5. The primary purpose of the double-stage dehumidification is to reduce the air moisture content in the case of humid air with lower regeneration temperature requirements.

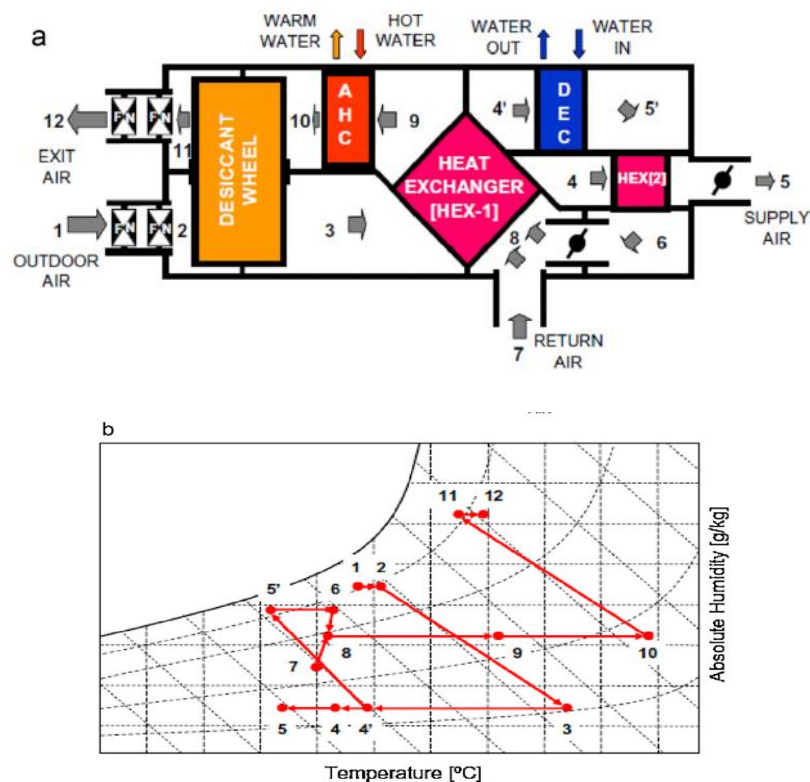


Figure 2.4 Constant humidity supply air desiccant cooling system (Enteria et al., 2010)

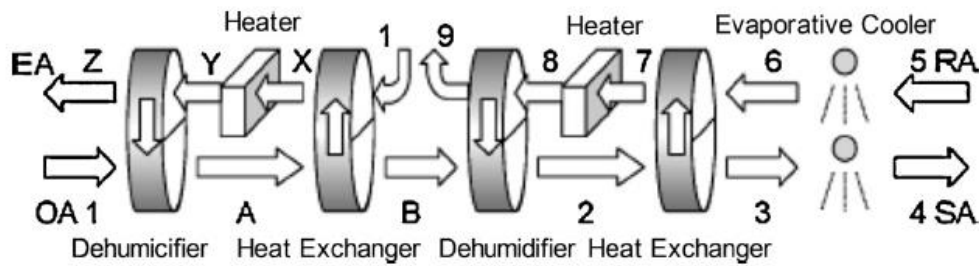


Figure 2.5 Double-stage dehumidification desiccant cooling system (Ando et al., 2005)

2.4 Liquid Desiccant Cooling

The liquid desiccant cooling system uses liquid desiccant in controlling air moisture content. The process of air moisture reduction is through an absorption process. For removal of the air moisture content, the air is passed in the same way as in the air cooling done in the solid desiccant cooling system. There are several types of liquid desiccants; they are calcium chloride, lithium chloride, lithium bromide, tri-ethylene glycol, and a mixture of 50% calcium chloride and 50% lithium chloride, and so on (Abdulrahman et al., 2013). These liquid desiccants have general properties, but their requirements are not fully answered by any single desiccant. These requirements include low vapour pressure, low crystallisation point, high density, low viscosity, low regeneration temperature, and low cost.

2.4.1 Basic Principle

Air is dehumidified when in contact with strong liquid desiccants, and subsequently, it will provide sensible cooling to the dehumidified air before being sent to the conditioned space. When the solution is weakened by the absorption of moisture, it is sent directly to the regeneration process to release the moisture using external heat resources. This process is

called regenerating the desiccant. The most straightforward liquid desiccant system configuration, shown in Figure 2.6, consists of a conditioner, regenerator, a heat exchanger to heat the solution, and another heat exchanger to cool the solution (Rutherford, 2000). The solution attracts moisture from the air in the conditioner, which weakens the solution. The weak solution is sent through the heat exchanger for heating and then sprayed into the regenerator. In the regenerator, the air carries away the water vapour from the solution. After this process, the solution is sent to the conditioner across the heat exchanger for cooling. Liquid desiccant dehumidification often requires two desiccant air-contact devices: absorbers and regenerators. A liquid desiccant absorber/regenerator system has three configurations, which are packed tower, spray chamber, and spray coil arrangement (Enteria N. et al., 2010).

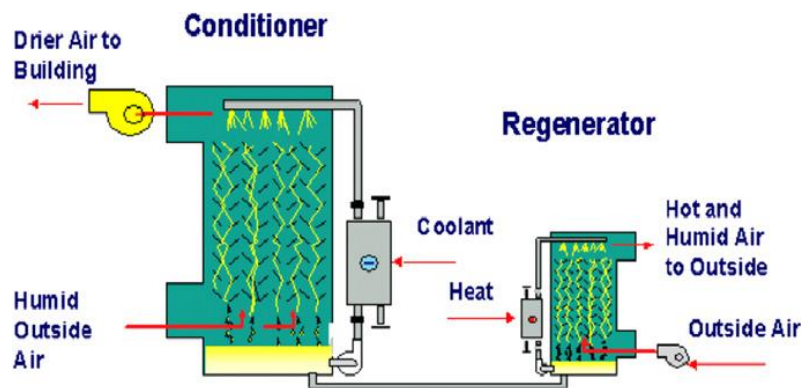


Figure 2.6 Liquid desiccant system (Rutherford, 2000)

2.4.2 Development in Liquid Desiccant Cooling

The development of the liquid desiccant cooling has taken the limelight in by several authors and engineers in recent years compared to solid desiccant cooling. This is because it can offer great advantages, such as lower regeneration temperature and capability for easier storage. The following are the recent developments in liquid desiccant cooling.

2.4.2.1 Packed Bed

Many researchers have investigated the packed-bed or tower-type absorber. In packed towers, the packing materials may be categorised as random or structured, with the former without regular geometric form and placed randomly in the packing tower, and opposite structured packing material, with fixed geometric form. Pall, rosette, ladder rings, ceramic saddles, and sandy layers are among the many random packing materials in cooling tower systems. Alternatively, a wavy plate, grid, silk net, and honeycomb packing materials are considered as structured packing materials and have been used in many cooling tower systems. The liquid desiccant solution is distributed into a thin film by the packing materials. Figure 2.7 shows the structural packing materials used in cooling towers.

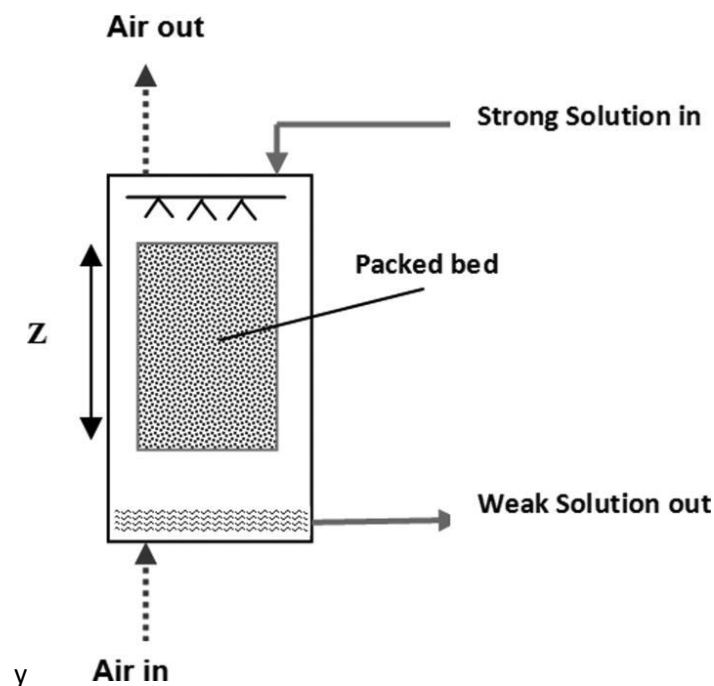


Figure 2.7 Structured packing materials (Fateme Eshandari Nia et al., 2011)

2.4.2.2 Spray Chambers

In the spray-type liquid desiccant arrangement, a large surface area for heat and mass transfer is obtained by distributing the liquid desiccant solution into small droplets by the help of nozzles on the spray chamber. Although the initial cost of the spray chamber arrangement can be low and has the advantage of being compact in size, but its absorption effectiveness is considered low. Other common drawbacks of the spray tower arrangement include possibilities of liquid carryover and large pressure drops on liquid side, as explained by Jain & Bansal (2007).

2.4.2.3 Wet Wall Column

In a wetted wall or falling film column, a vertical tube or plate is employed, over which the liquid desiccant solution is allowed to flow by gravity. The liquid flows as a thin film over the vertical surface represented by tubes or plates. As air flows through this column, it comes into direct contact with the surface of the following liquid film. The wetted wall columns are considered to have a low pressure drop and low initial cost. They can also provide high contact area per unit volume, as reported by Jain & Bansal (2007).

2.4.2.4 Single-Stage Liquid Dehumidification System

The single-stage liquid desiccant can be heated by an electric heater, which is used to simulate the solar collector. The layout of the experimental setup is given in Figure 2.8. Sanjeev Jain et al. (2011) used paper as indirect-contact air desiccant that allows moisture transfer, triggered by a difference in partial water vapour pressure between the desiccant and the air. He studied a wide range of desiccant flow rates, concentrations, temperatures, and ambient

conditions. Nevertheless, his studies were limited by moisture transfer characteristics of the surface in the contactor.

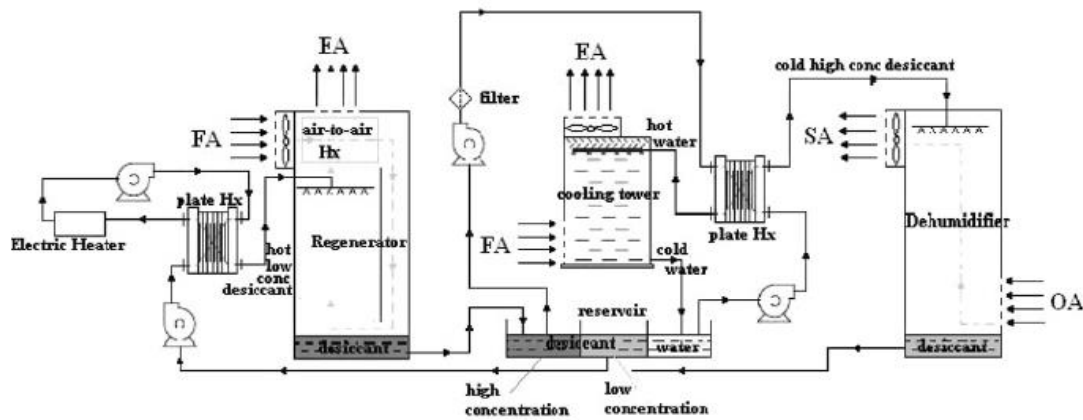


Figure 2.8 Liquid desiccant cooling by (Sanjeev Jain et al., 2011)

Croffoot & Harrison (2012) installed liquid desiccant air conditioning that uses low-flow parallel plate heat and mass exchanger as both the conditioner and regenerator (Figure 2.9). Desiccant flows in a thin wick down the outside of the plates, while cooling/heating water is circulated through the plates internally. The low-flow design eliminates carryover of desiccant into the air stream and improves the storage capacity because of the higher air-to-desiccant mass-flow ratio. The system was tested for the duration of the summer in Ontario, Canada. The average latent cooling power was 13.2 kW and average total cooling power was 12.3kW. Over the days of operation, a thermal *COP* of 0.47 was observed. Solar collector efficiency was 56% and solar energy was able to provide 63% of the heat to drive the liquid desiccant air conditioning. Performance is expected to improve as the weather gets hotter and more humid. Over the winter between October until March, the solar array was operated with a dry cooler. 18,800 kWh were collected with an average efficiency of 61%.

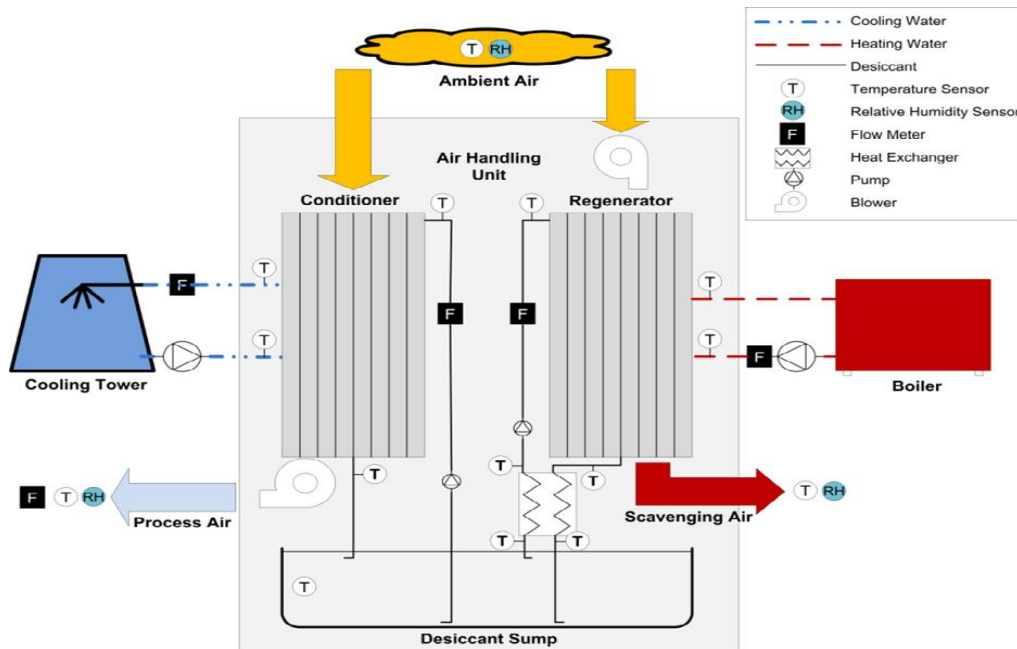


Figure 2.9 Schematic of liquid desiccant air conditioning equipment and instrumentation used by Croffoot & Harrison (2012)

There was an interesting study by Audhah et al. (2011) in utilising single-stage liquid desiccant cooling system. The authors conducted research on the feasibility of using a solar-powered liquid desiccant system to meet fresh water and cooling needs at minimum cost (Figure 2.10). Parabolic solar concentrators will be the heat source for regenerating the liquid desiccant. The water condensate will be captured from the air leaving the regenerator. The regenerator exit humidified air is directed to a cooling coil submerged in cold sea water where condensation occurs and distilled water is collected. They find out that heat sink temperature is a critical optimisation parameter and dictates the appropriate regeneration temperature within the constraints of operation of the system to deliver its outputs of fresh water and dry cool air stream.

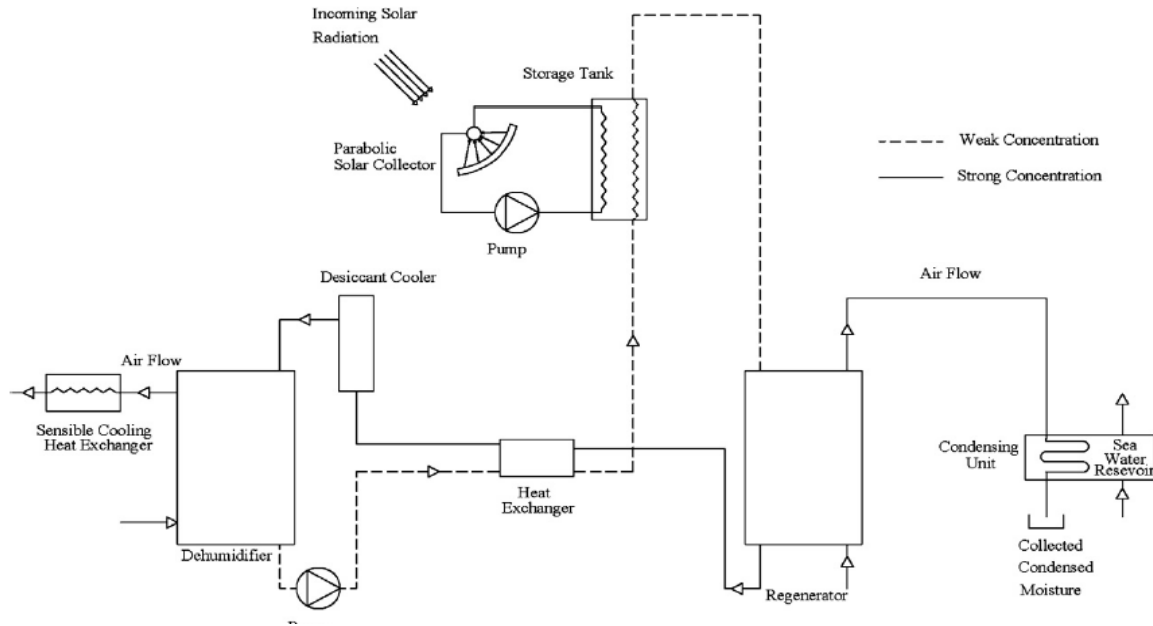


Figure 2.10 Schematic diagram of the liquid desiccant system (Audhah et al., 2011)

Kessling et al. (1998) have developed a liquid desiccant cooling system as shown in Figure 2.11. The difference of his system with others is the dehumidified air is cooled by simple adiabatic evaporative coolers in a Pennington or ventilation cycle, or more efficiently, by indirect evaporative coolers. The dehumidified air (2) is mixed (3) with indoor air of state (5) cooled in an indirect evaporative cooler (4) and supplied to the room. A part of the air flow (6) is humidified in contact with a wet heat exchanger surface and provides evaporative cooling (7). If heat supply is available, the diluted liquid desiccant can be concentrated in the regenerator.

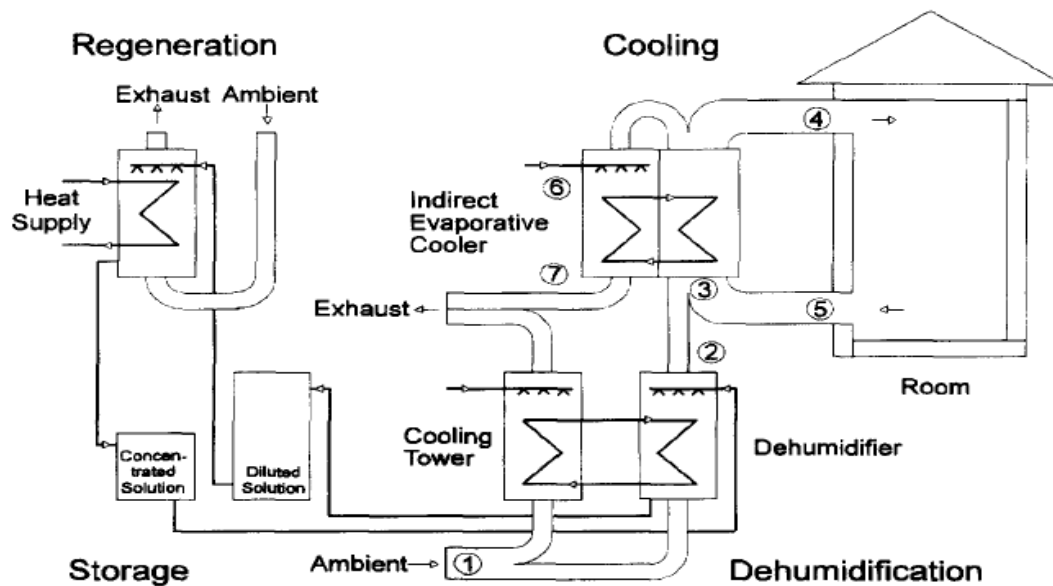


Figure 2.11 Schematic of an open-cycle liquid desiccant cooling system (Kessling et al., 1998)

2.4.2.5 Multi-stage Liquid Dehumidification System

Xiong et al. (2009) investigated the two-stage hybrid desiccant cooling system using lithium bromide as the primary dehumidifier assisted by calcium chloride powered by solar energy (Figure 2.12). The results show the improvement of the energy storage capacity compared to the ordinary liquid desiccant system, also, better dehumidification performance due to the effect of pre-dehumidification. The performance of the system is higher than the ordinary liquid desiccant by 15.8% and 32% ($TCOP$ and COP). The cost of the liquid desiccant sections decreased by 5% as long as there is an abundant supply of solar energy source. Meanwhile, Li et al. (2005) introduced the design of the liquid desiccant total heat exchanger shown in Figure 2.13. The simulated EER (energy efficiency ratio) for the air processor is 6.3 to 7.3 in summer condition.

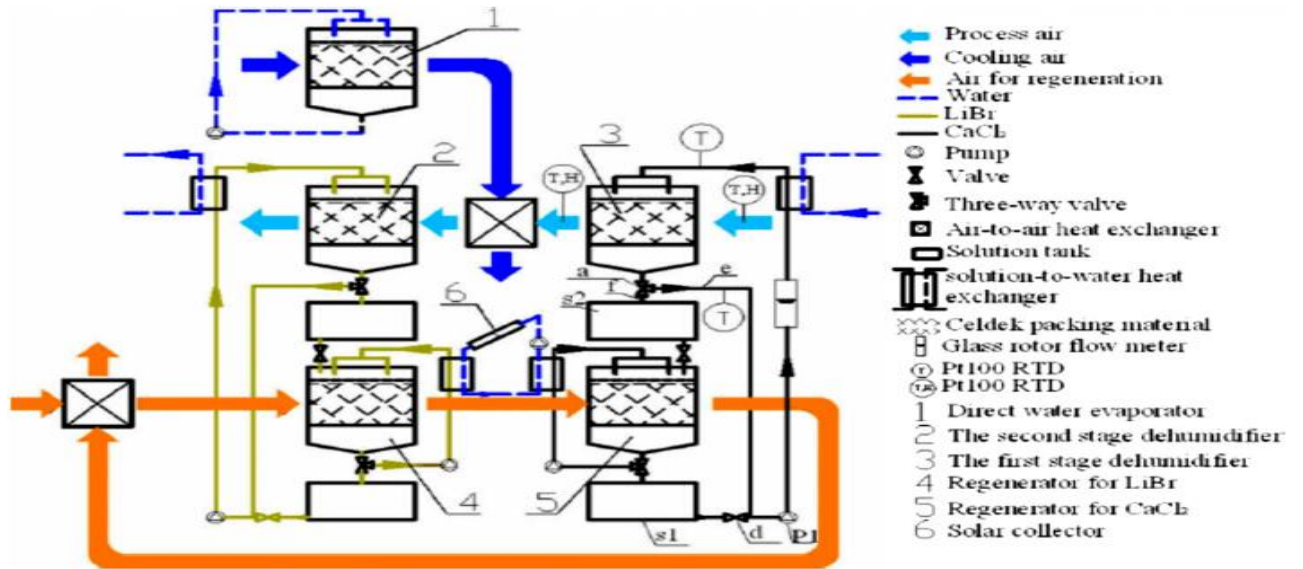


Figure 2.12. Solar two-stage liquid desiccant based hybrid system (Xiong et al., (2009))

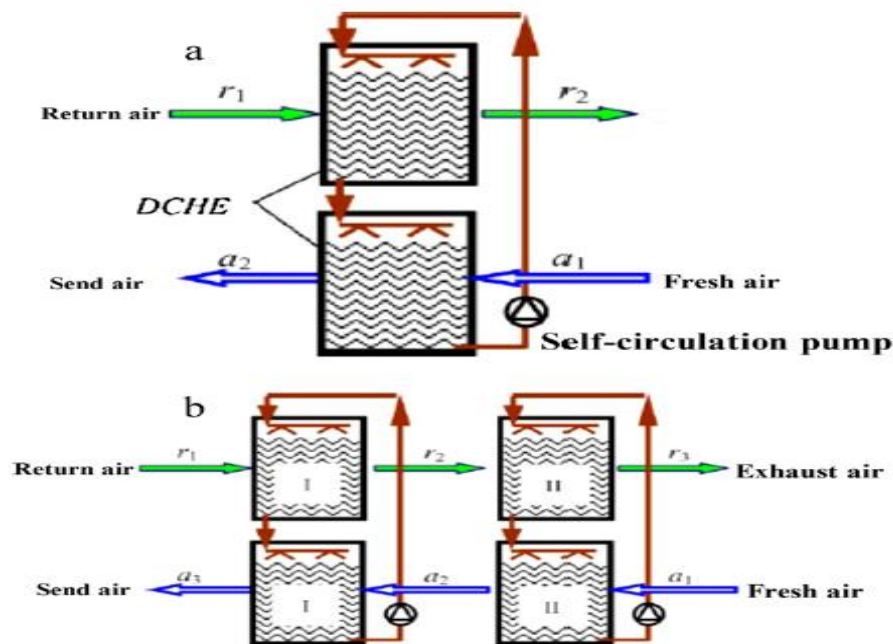


Figure 2.13 Liquid desiccant total heat exchanger: (a) single stage and (b) double stage (Li et al., (2005))

2.4.2.6 Hybrid Liquid Desiccant Cooling System

A hybrid liquid-desiccant air conditioning system is a combination of multiple mechanism of liquid desiccant dehumidification process and cooling unit in removing the latent load and meeting the sensible cooling load. The main classification of the hybrid liquid desiccant cooling system is based on the cooling units used in the reduction of the dehumidified air temperature. Using the hybrid system, the dehumidifier can absorb enough moisture with a smaller dehumidifier and by lowering the required regeneration temperature needed by the desiccant cooling system from (70– 80) °C to (50–60) °C (Dennis Schlepp K.S., 1984).

Al-Farayedhi et al. (2002) combined a 5-ton-capacity vapour compression unit with a packed dehumidifier and regenerator as shown in Figure 2.14. A gauze-type structure was used as packing towers with the following dimensions: height = 2.6 m; diameter = 50 cm; and thickness = 0.5 cm. Calcium chloride was used as a liquid desiccant to dehumidify the air inside the dehumidifier. The results showed that the hybrid cooling system provided higher *COP* than the conventional system. The *COPs* of the hybrid desiccant system for different modes of regeneration process were as follows: desiccant heating = 1.164, air heating = 1.616, and desiccant and air heating = 1.4221, and the *COP* of the vapour compression unit was 0.989.

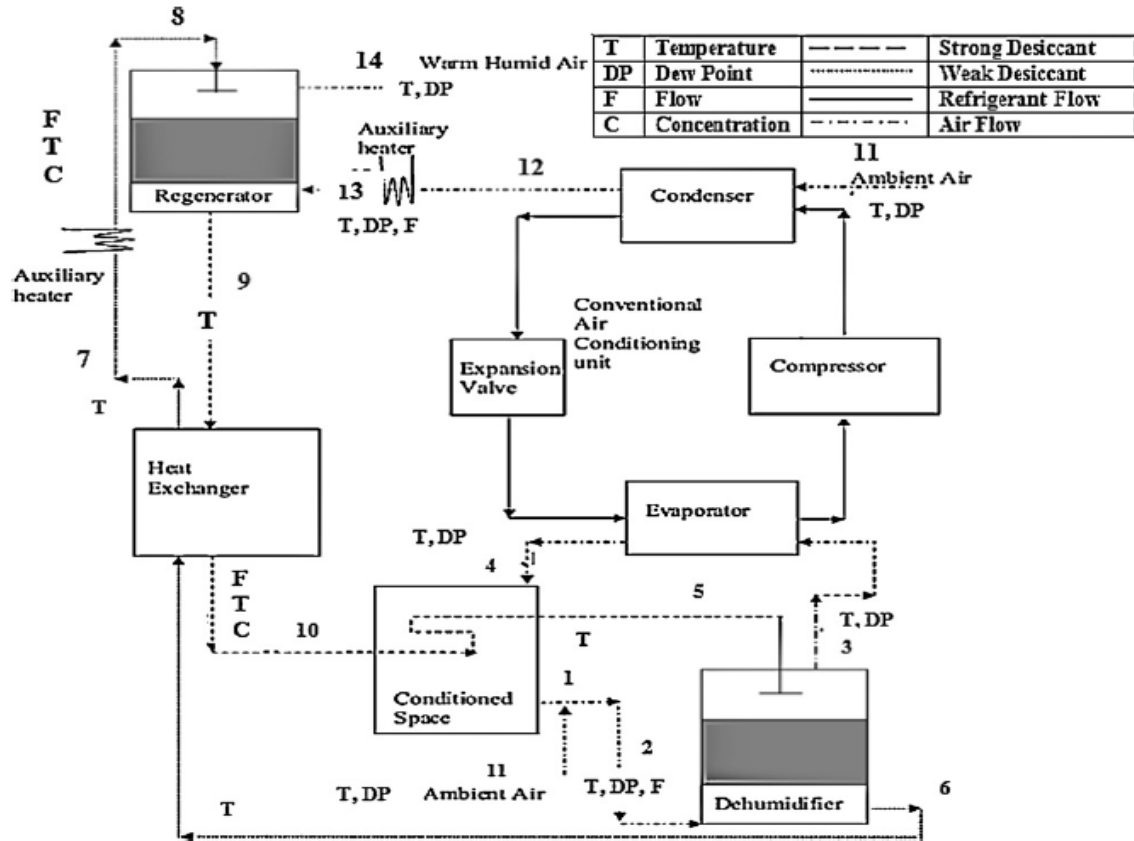


Figure 2.14 Experimental set-ups for hybrid cooling system (Al-Farayedhi et al., 2002)

Dai et al. (2001) established a hybrid air conditioning system consisting of a vapour-compression unit, desiccant dehumidification, and evaporative cooling, as shown in Figure 2.15. The desiccant dehumidification unit consisted of packing honeycomb paper. The *COP* of the system increased by approximately 23.1% compared with the *COP* of the conventional vapour system when the dehumidification section was considered. It increased by 15.3% when the dehumidification and evaporative sections were taken into account.

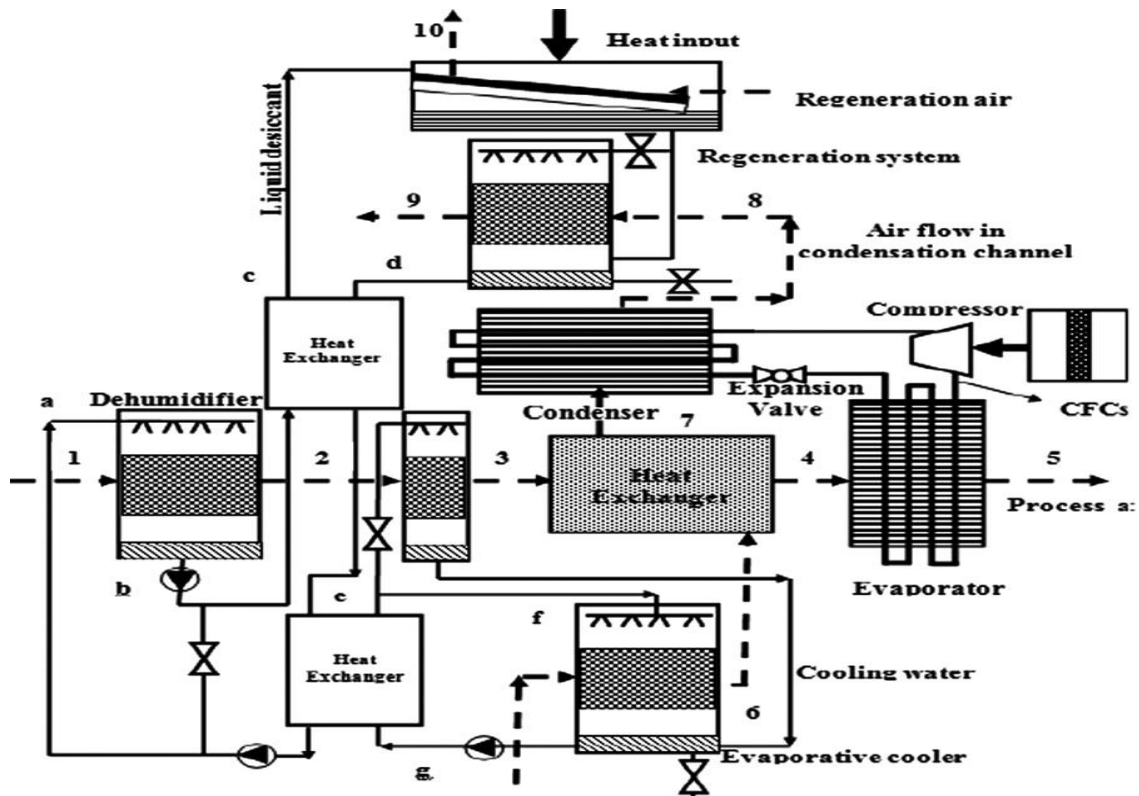


Figure 2.15 Schematic of the hybrid cooling system (Dai et al., 2001)

Zhang et al. (2010) integrated the heat pump system with a liquid desiccant dehumidification unit. The system provided both cooling and dehumidification in the summer mode, and heating and humidification in the winter mode. As shown in Figure 2.16, two major units represent the system: the sensible heat removal unit and latent heat removal unit. Two refrigerant cycles correspond to two heat pumps in the hybrid air conditioning system: one is used to deal with the sensible return air, whereas the other deals with pre-cooling the liquid desiccant on the dehumidification side and reheats the liquid desiccant on the return side. The figure shows the psychometrics of the process (a) in winter and (b) in summer. The results show that the *COP* is 30–40% higher than that of the heat pump integrated with an electric heater.

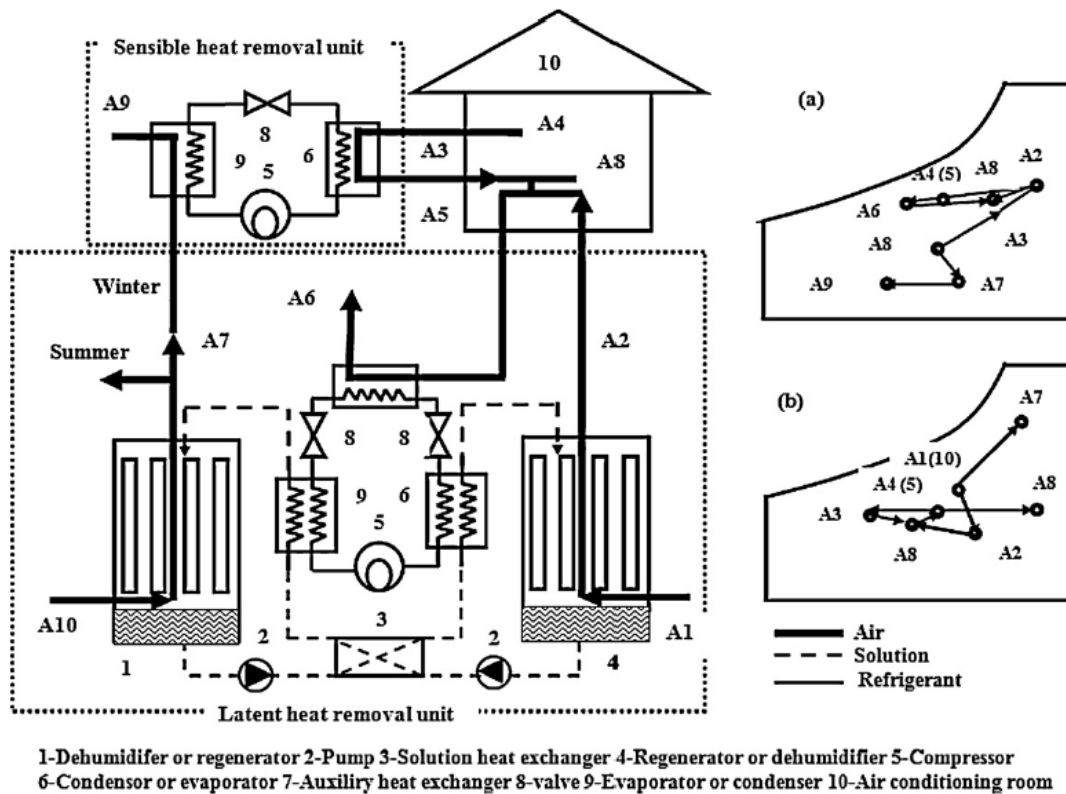


Figure 2.16 Schematic diagram of hybrid air conditioning with psychometric chart for (a) winter (b) summer (Zhang et al., 2010)

Bergero and Chiari (2010) examined the performance of an air dehumidification system integrated with a vapour compression inverse cycle. As shown in Figure 2.17, air is dehumidified in an air-solution membrane contractor and cooled by a vapour compression cycle. Lithium chloride, as a solution, is cooled by a vapour compression unit and regenerated in another membrane. They studied the steady-state behaviour of the system on different significant climatic parameters. The results show 60% energy savings at high latent heat load compared with a traditional direct expansion air conditioning system.

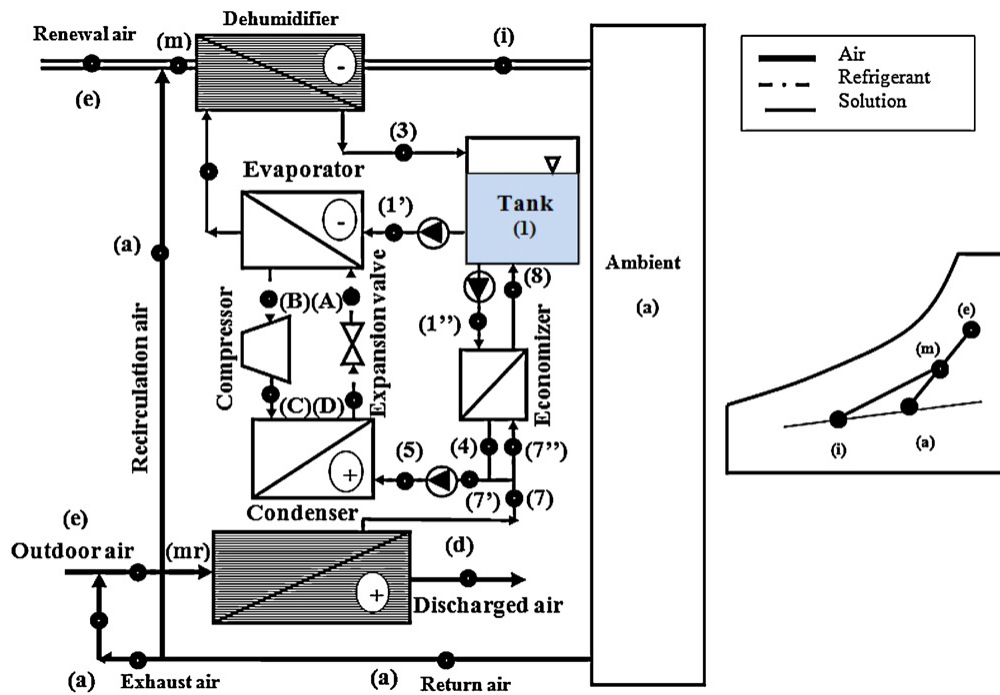


Figure 2.17 Layout of hybrid air conditioning system (Bergero and Chiari, 2010)

2.5 Comparison of Solid and Liquid Desiccants Dehumidification Systems

Currently, available desiccant systems can be generally classified as solid and liquid desiccant dehumidification systems, each with its own advantages and disadvantages. They all attract and hold moisture through adsorption when solid desiccant is used or absorption when liquid desiccant is used. The main differences between solid and liquid dehumidification systems are summarised as follows:

- As liquid desiccant is generally caustic, the equipment used in the system should be anti-corrosive, a problem which solid systems do not have.
- Dirt and dust carried in the primary or supply air, return air, and generation air is a problem with both solid and liquid desiccant systems, as it can block the pores in the honeycomb matrix of solid desiccant and can pollute the liquid desiccant solution.
- The vapour pressure of some liquid desiccants is small compared to solid desiccants, as found with pure triethylene glycol; this condition causes some of the glycol to evaporate

into the supply air stream, which makes it unacceptable for use in the air conditioning of an occupied building (Nelson & Goswami, 2001). Liquid desiccants have the potential to be harmful if they are airborne.

- The higher pressure drop in the air supply for solid desiccant systems needs more energy to increase the pressure loss in the supply and regeneration air, resulting in lower *COP*.
- The *COP* of solid desiccant systems is lower than that of liquid desiccant systems. The higher regeneration temperature of solid desiccant systems requires additional energy, which increases the system's input energy.
- Adsorption cooling unit are attractive as they can be operated at temperature level where liquid absorption systems cannot work (Dieng & Wang, 2001).
- Solid absorbents have relatively lower sorption capacity than liquid absorbents. For instance, a typical molecular sieve absorbent can hold only 17% of its dry mass in water when air flow condition is at 21 °C and 20% relative humidity (ASHRAE, 2001), while in contrast, lithium chloride absorbent can hold 130% of its mass under the same airflow temperature and relative humidity conditions.

On the whole, because solid desiccant systems are compact, less subject to corrosion and require less maintenance, the majority of systems used for commercial buildings use solid desiccant.

2.6 Membrane in Liquid Desiccant Air Dehumidification

As mentioned before, to prevent desiccant solution droplets from crossover, semi-permeable membranes could be used to realise air dehumidification. This was called membrane-based liquid desiccant air dehumidification technology. A good membrane should be characterised by the following (R. Namvar et al., 2012):

- Low vapour diffusion resistance in order to increase the rate of moisture transfer.
- High modulus of elasticity in order to reduce the deflections in the membrane, and consequently reduce the flow misdistribution that may occur in the flow channels.
- High liquid penetration pressure, so that no desiccant solution leaks into the air channels.

The selectively permeable membranes could be made into parallel-plates or hollow fibres. The air flowed on one side and the solution flowed on the other side of membranes. Heat and moisture were exchanged through the membrane effectively. Other harmful gases or liquid solution were prevented from permeating through the membranes.

2.6.1 Development of Membrane Technology into Dehumidifier/Regenerator

The parallel-plates or hollow fibre membranes were packed in plastic shells to form parallel-plates membrane modules or hollow fibre membrane modules, respectively. Both the parallel-plates and the hollow fibre membrane types had their own advantages and disadvantages. The former were simpler and easier to construct. The pressure losses in the channels were relatively less. The latter were more complex and difficult to construct, especially the two ends sealing problems. The pressure losses in the channels were higher. However, the packing densities of the hollow fibre membrane modules were larger and the effectiveness was higher.

A parallel-plates membrane module was shown in Figure 2.18. It was formed by the membranes stacked together. Equal spacing was kept between neighbouring membranes to create flow channels. The solution stream and the air stream flowed alternatively through the parallel channels, in a crossflow arrangement, for easiness of duct sealing. Other structures were not easy for sealing the liquid fluids. The process air was dehumidified by the liquid

desiccant while absorbing moisture from the air through the membranes. The parallel-plates membrane modules had been used as air-to-air heat and mass exchangers before. Similarly, because of the simple structure and easy fabrication of the parallel-plates membranes, this module was used in liquid desiccant air dehumidification (Huang S.M. et al., 2012).

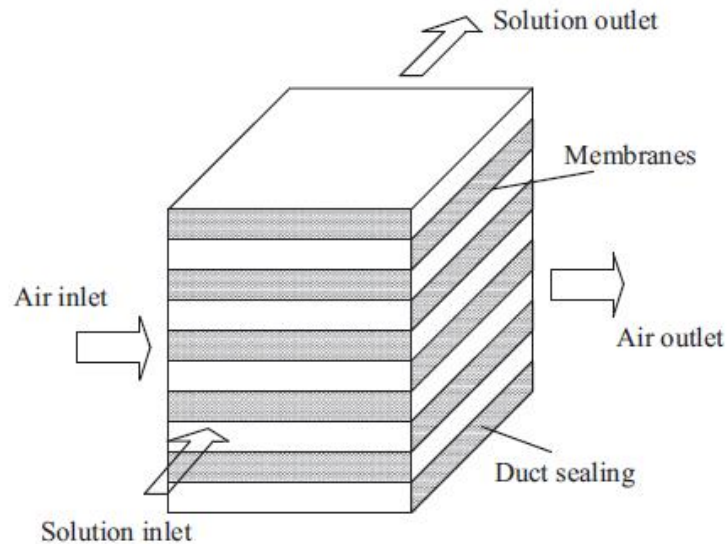


Figure 2.18 Structure of a parallel-plate membrane module (Huang S.M. et al., 2012)

Ahmed et al. (2013) proposed a membrane liquid desiccant air conditioning system that uses flat-plate liquid-to-air membrane energy exchangers (LAMEEs) to serve as the dehumidifier and regenerator, as shown in Figure 2.19. The proposed system is modelled using TRNSYS software. The performance of the system is investigated under different climate conditions (i.e., outdoor temperature and relative humidity), design conditions (i.e., number of transfer units and solution heat exchanger effectiveness), and operating conditions (i.e., liquid desiccant flow rate and temperature of solution entering the regenerator/dehumidifier). The air and desiccant solution streams in the LAMEEs are separated by a semi-permeable membrane (e.g., polyethylene, polypropylene) that allows water vapour transmission, but does not allow the transmission of any liquid water (R. Naymar et al, 2012). Therefore, no liquid desiccant droplets are entrained in the air streams due to the indirect-contact between air and liquid

desiccant. Membrane properties have a significant role in the heat and mass transfer in the LAMEEs, especially for the mass transfer.

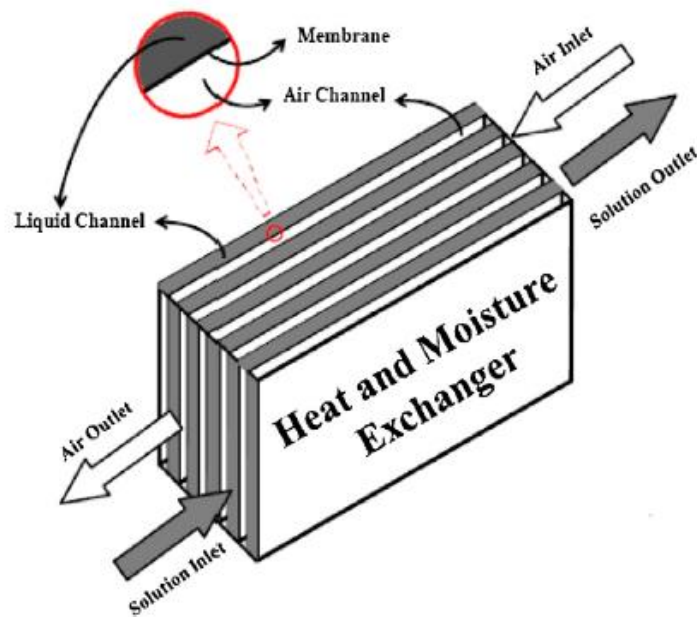


Figure 2.19 Schematic of a counter-flow LAMEE (Ahmaed et al (2013))

Bergero and Chiari (2010) have conducted a study on two crossflow membrane contactors in which the air and solution flows were separated by means of plane membranes in the hybrid air conditioning plant. Figure 2.20 shows a sketch of the membrane contactor; the dimensions of the basic module considered in the simulations were reported: a total air-solution exchange surface of about 1.5 m^2 is obtained by superimposing 30 membranes. The membranes are made of PVDF (polyvinylidene fluoride) and fixed to polyester support. In order to evaluate the performances of the dehumidifier and regenerator, a finite-volume FORTRAN computer code is used. The mass and energy conservation equations for both air and solution flows are numerically solved. Experimental results showed that the computer code was able to calculate the performances of the contactor reasonably accurately. The results obtained indicate that the proposed hybrid air conditioning system is able to achieve significant energy savings in

comparison with a traditional direct-expansion system in summertime conditions and that in particular operating conditions these savings may even exceed 50%.

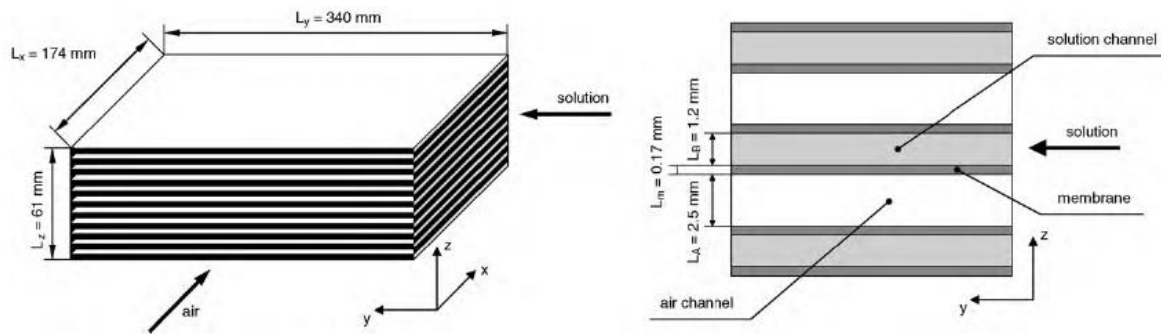


Figure 2.20 Dimensions and cross-section of the crossflow membrane contactor (Bergero and Chiari, 2010)

As mentioned earlier, the parallel-plate membrane module was simple in structure and easy to fabricate. The packing density of the module was about $500 \text{ m}^2/\text{m}^3$, which was not large enough. However, the hollow fibre membrane module, as shown in Figure 2.21, was more attractive because its packing density was higher and the heat and mass transfer capability was larger. As seen, the concept was similar to a parallel flow tube-and-shell heat exchanger where the two separate fluids flow on the respective sides of the membrane. For air-liquid contacting hollow fibre membrane contactor, the airstream flowed either outside (shell side) or inside (tube side) the tubes. However, in most engineering applications, the solution flowed in the tube side, while the air flowed in the shell side in a counter flow arrangement. The benefit of this flowing configuration was that the pressure drop for the airflow in the shell side was less and the performance was better (Al-Farayedhi et al., 2002). The mathematical models of the fluid flow and mass transfer in the gas-liquid hollow fibre membrane contactor were established conjugate heat and mass transfer (Zhang L.Z. et al., 2012).

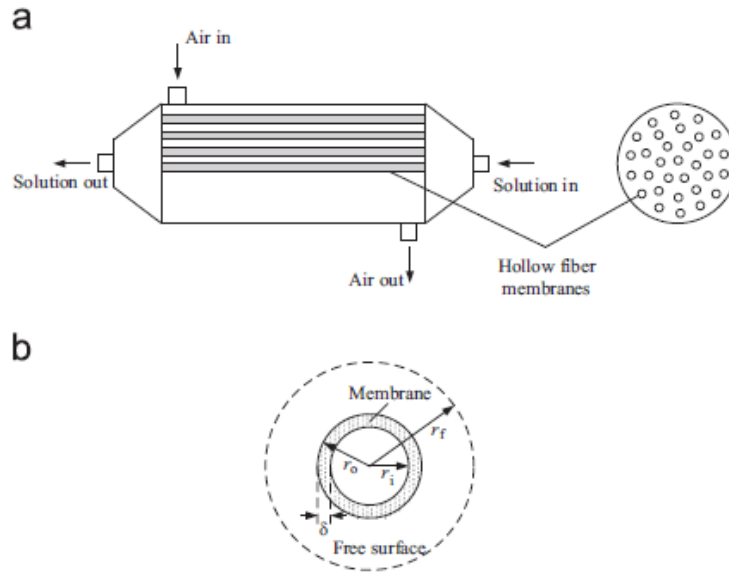


Figure 2.21 Schematic of the hollow fibre membrane module and the cell with a free surface model. (a) The shell and the tube structure; (b) the cell with a free surface (Zhang L.Z. et al., 2012)

As we discussed earlier, counter flow was the most commonly used arrangement for membrane contactors. However, this structure had a potential disadvantage of rather large pressure drop losses. To address this problem, the crossflow hollow fibre membrane module, as shown in Figure 2.22, was proposed for liquid desiccant air dehumidification (Zhang L.Z. et al., 2013). The concept was similar to a crossflow tube-and-shell heat exchanger. The solution stream flowed in tube side, while the air stream flowed across the fibre bank in the shell side in a crossflow arrangement. The fluid flow and heat mass transfer in the inner tubes were actually similar to those in the common metal-formed round channels. The fluid flow and heat and mass transfer in the shell side were more complex than those in the inner tubes. In order to solve the fluid flow and heat transfer problems in tube-and-shell heat exchangers, two distinct approaches were available to describe the fibre-to-fibre interactions: free surface model and periodic unit cell model.

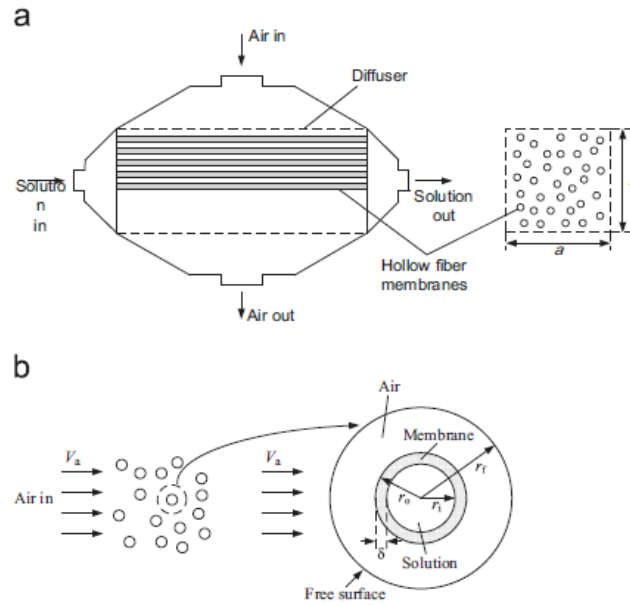


Figure 2.22. Schematic of a crossflow hollow fibre membrane contactor, a free surface cell selected as the calculation domain (a) the shell and the tube structure; (b) the free surface cell (Zhang L.Z. et al., 2013).

2.7 Summary

The chapter presented a thorough literature review on desiccant dehumidification cooling technologies. These review topics are directly related to this research work conducted in this thesis, and significant knowledge and understanding of these technologies are essential to the progress of the proposed research work.

In the earlier parts of the chapter, desiccant dehumidification and cooling were introduced, giving the various geographical regions employing desiccant systems and the numerous advantages of utilising such methods. Both liquid and solid desiccant systems have been discussed with a different technological arrangement of both fully explained, and their basic operational differences highlighted. Besides, hybrid desiccant dehumidification and cooling

systems, which employ better technical structure, have also been described. Solid desiccant systems are also attractive in comparison to the liquid desiccant systems because of their compact nature and less complication in design and operation. Solid desiccant systems are less subject to corrosion, and usually, require less maintenance than liquid desiccant. Nevertheless, liquid desiccant systems offer a generally higher overall performance, and better dehumidification and regeneration rate.

The review shows that the desiccant cooling system is a complicated technology, but yet simple enough to improve the performance of the system. Some ways to enhance the COPs of the system is to design an efficient dehumidifier. Those ways were such as multi-stage dehumidifier, double output (such as water condensation), and combining with conventional vapour compression unit. This review can be a perfect guideline on how to fabricate a new type of desiccant cooling system. Because of the advantages of liquid desiccant cooling, it is clear that liquid desiccant will be continued to be future research instead of solid desiccant. Nevertheless, we could not merely ignore other advantages of the solid desiccant wheel, whereby it is simple and compact. That is why we will try to implement the solid desiccant mechanism into our design that will be incorporated with liquid desiccant as the main working fluid.

Chapter 3

Theoretical Background and System Description

3.1 Introduction

This chapter will discuss the fundamental principle of the system and also detail the selection of materials and components used in this project. The wheel material selection and evaluation process of the research has proved to be of great importance. Desiccant materials selection has become a pivotal factor in determining the performance of the overall system. The efficiency of dehumidification and refrigeration systems is strongly dependent on the chosen working materials and components. The most suitable materials for the desiccant wheel were chosen based on some given criteria. Structural flexibility, cost, and commercial availability are examples of many additional considerations that will be addressed throughout this project. In addition, selected liquid adsorbent should have suitable material compatibility and not present any toxicity or environmental concerns.

3.2 Research Methodology

Figure 3.1 shows the flow chart for the research methodology that was applied novel liquid desiccant wheel cooling system. This figure shows how to make a good research flow and necessary method to accomplish your research aim and objectives. First, a preliminary study was conducted by identifying problems that occurred in desiccant cooling technology. From there, the aim and objectives of the research developed so that a novel liquid desiccant wheel

cooling technology can be created. Then the appropriate structure of the study can be developed by identifying a critical component needed to complete the research.

Secondly, an extensive literature review on desiccant cooling technology has been done at this stage. The basic principle of the available systems was conducted to get the main idea of how such system work. One of the focus of the literature is the theoretical background, methods, and tool that has been used by the previous researcher. This useful knowledge gave an excellent insight into how to develop a new desiccant cooling system.

Thirdly, a mathematical model, was developed based on multiple inputs from the previous literature and applied it to the proposed system with proper adjustment. Heat and mass transfer modelling was used on the wheel rotor. It used moisture and energy balance equations that accurately predict the output of the system. To help to simulate the system that involved a lot of data iteration, a thermodynamical software called Engineering Equation Solver (EES) was used. From the software itself, it can suggest the proper sizing of the wheel to make it compact as possible. It is excellent software where researchers can encode their modelling to the software's worksheet.

Fourthly, based on the information gathered at the modelling stage, the design of the system can be established. One of the most important is the dimensions of the wheel where it will determine the size of the final system later on. This information can be translated to detail drawings that show all the necessary components and workflow. Then, the design was sent to the workshop for the fabrication process by the technical team.

Fifthly, after the fabrication and inspection process was done accordingly, the test rig was ready for the experimental stage. To help in completing the data collection process, the latest data acquisition (DAQ) system such as EKH4 and DT80 equipment. This equipment helps to collect environmental data needed for further analysis of the system. For determining the COP of the system, EES will help to calculate the exact value using the relevant formulas.

Sixthly, after simulation and experimental work were done, the novel liquid desiccant wheel cooling system undergo the environmental and economic impact assessment. The former was done by using Eco-indicator 95 tools which was done by the previous researcher, and it is one of the best methods to assess the environmental impact. Then the latter was evaluated by using payback period analysis that can predict the break-even period for the system.

Lastly, the conclusion and recommendation would be made at the final stage. This stage examines whether all the completed work successfully met the objectives that have been set earlier. Later on, a set of recommendations was proposed so that this work can be improved and continue for future work.

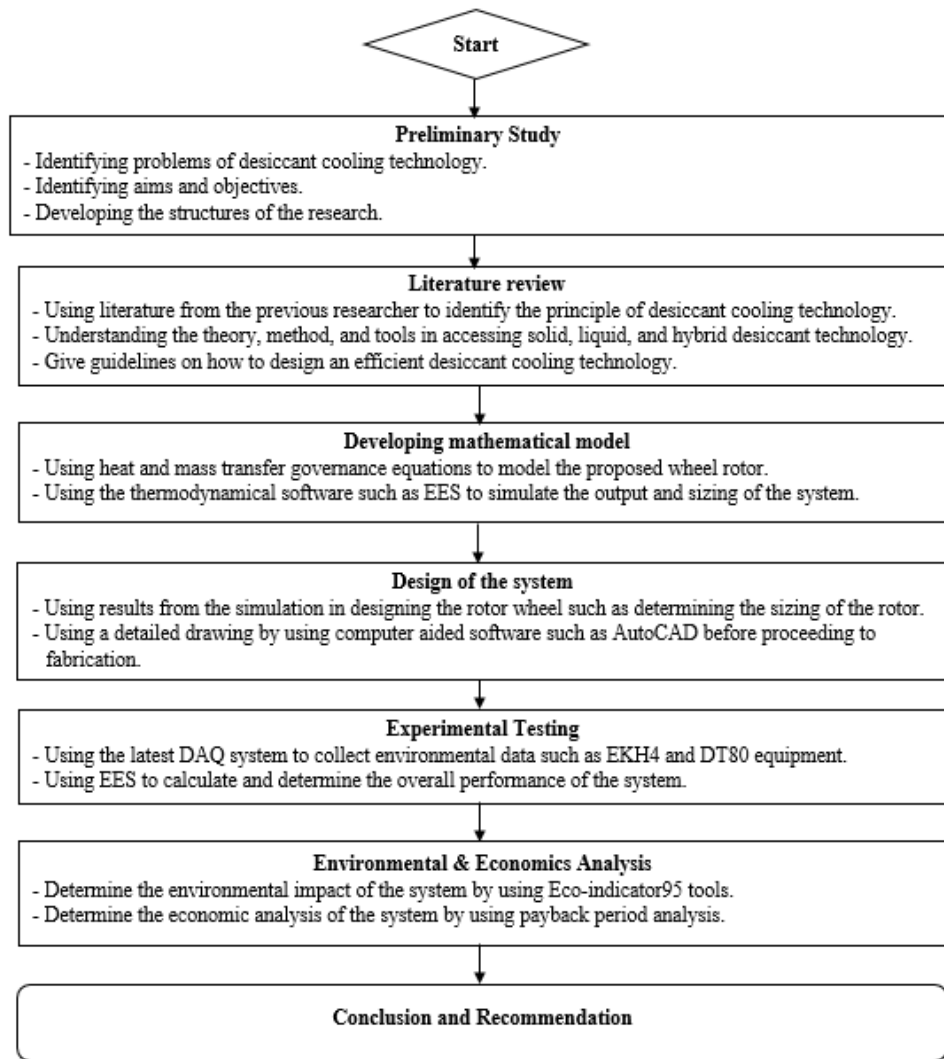


Figure 3.1 Research methodology flow chart of the novel liquid desiccant wheel cooling system

3.3 Description on Novel Liquid Desiccant Wheel Cooling System

Figure 3.2 shows the schematic diagram of the whole hybrid wheel liquid desiccant cooling system that can be installed in a commercial and residential building. The system will be installed to facilities where fresh air (FA) will be supplied by outside air. The dehumidification and regeneration systems are combined within a unit box, as shown in the black dash line. The wheel will be immersed in potassium formate enclosed in a tank, as a single motor will drive both wheels in the dehumidification and regeneration section. It will

ensure a uniform rotation and coupling mode for both wheels, besides, to reduce electric power consumption by employing a single motor. Heating of potassium formate will be done by heat waste or renewable heat such as solar thermal where flat plate solar collector can be used.

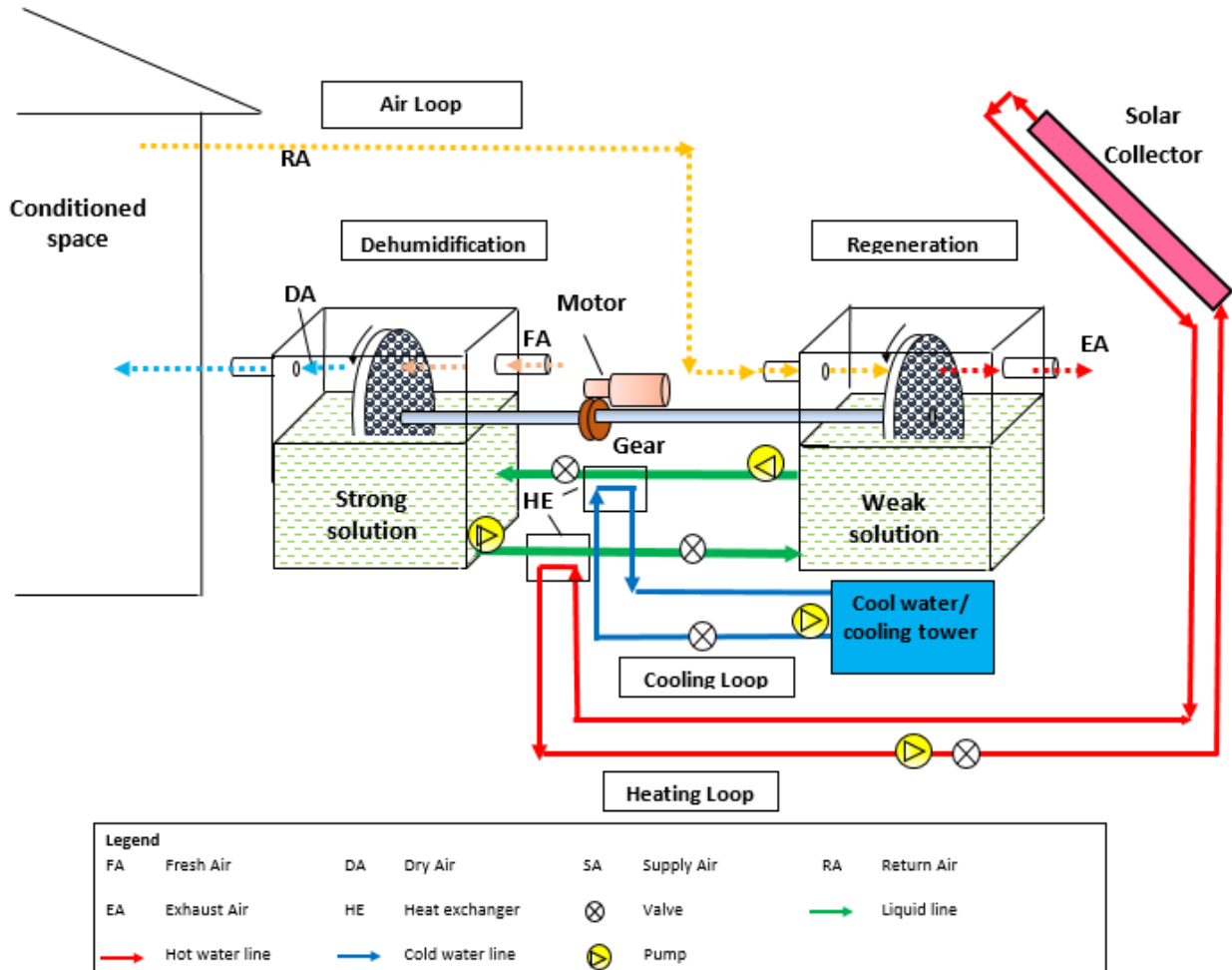


Figure 3.2 Schematic diagram of novel liquid desiccant wheel cooling system in residential building

The heart of the system would be the desiccant wheel as shown in Figure 3.3. It incorporated thousands of small polypropylene (PP) square slots that are wrapped up by aluminium plate to hold the PP board together. The size of the desiccant is 420 mm in diameter and 140 mm in thickness. The weight of the wheels is 2.5 kg each. It acts as the main carrier for the desiccant liquid to dehumidify when in contact with incoming fresh air. It is suitable for air-liquid

dehumidification medium, as it effectively executes heat and mass transfer and its cost is cheaper than other materials.

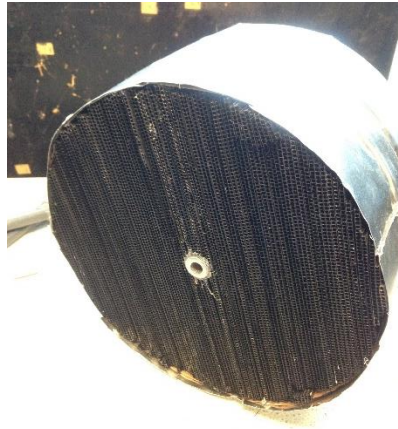


Figure 3.3 The face of the wheel for desiccant wheel cooling system

3.4 Theoretical Background of the Novel Liquid Desiccant Wheel Cooling System

In this section, we will discuss the theoretical background that is essential in evaluating the performance of a novel liquid desiccant wheel cooling system. The desiccant wheel, which is the most crucial part of the system, was treated as a rotor wheel that contains many small square slots that allow air to pass through it. Thus, we identified that heat and mass transfer plays a massive role in modelling the desiccant wheel. Afterwards, using the fundamental theories in thermodynamic cycles, we could model the overall performance of the novel liquid desiccant wheel system.

3.4.1 Mathematical Modelling of Desiccant Wheel

Figure 3.4 shows a schematic of the rotary dehumidifier and refrigerator. The rotor consists of thousands of identical narrow straight, square element uniformly distributed over its rotor cross-section. In this rotor, the three cylindrical coordinates (r, θ, z) , which are radial, angular,

and axial directions, respectively, in the model can rationally be reduced to the two coordinates (θ, z) . By considering the geometry of the bed, each constituent element within the total M identical elements in the θ -direction spanning the entire cross-section of the representative annulus, the remaining space variable is z . Meanwhile, the independent variable in the model is the time t (Ahmed M. Hamed 2005). According to Tanthapaichakoon and Prawarnpit (2002), the dynamic behaviour of the rotor can be simulated by following the transient changes occurring in the z -direction within each of the M slots as the rotor slowly turns. The following simplifying assumptions are stated in driving the heat and mass balance equations in this model:

1. To avoid the problem of varying the amount of airflow rate through each slot, airflow across the dehumidification and the regeneration cross-sections is assumed to be uniformly distributed.
2. The air stream flowing through each slot is assumed to be in piston flow. In the piston flow model, variation does not happen in the radial direction but exists only in the axial or z -direction. The piston-flow model can be considered as a series of a large number of entirely mixed N cells with a very small thickness, as shown in Figure 3.4.
3. In comparison to the convective effects in the same slot, gas-phase heat conduction and mass diffusion in the axial direction are negligibly small. It is because water molecules form strong physical bonds with the desiccant, and therefore the molecular diffusion within the desiccant matrix is little in the axial direction.
4. Each slot is assumed adiabatic, and heat conduction along the slot wall can be neglected.
5. The heat of absorption of the moisture can be approximated by the latent heat of vaporisation of water.
6. Because of the low rotation speed of the rotary dehumidifier and regenerator, the effect of the centrifugal force is neglected.

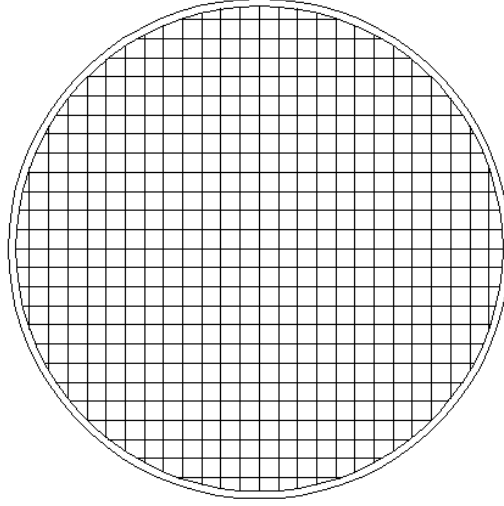


Figure 3.4 Schematic of square slots in rotor wheel

3.4.1.1 Governing Equations

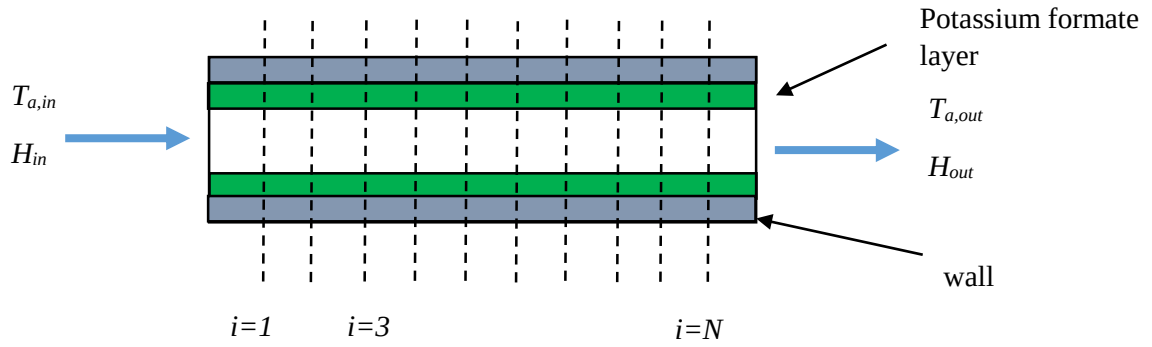


Figure 3.5 Representation of square slot as a series of completely mixed cells

The rotary desiccant wheel can be presented for piston flow assumption in order to create governing equations for heat and moisture transfer, as shown in Figure 3.5. For cell no i ($i=1,2,..,N$), the unsteady mass and energy balances are specified in the next section.

3.4.1.2 Gas-Phase Moisture Balance

The gas-phase moisture balance can be represented as below (Tanthapaichakoon and Prawarnpit, 2002):

$$\frac{dH_i}{dt} = \frac{NG_a V_{Hi}}{L} (H_{i-1} - H_i) - \frac{kA_h N V_{Hi}}{LA_c \varepsilon} (H_i - H_{Si}) + \left(\frac{H_i}{V_{Hi}} \right) \frac{dV_{Hi}}{dt} \quad (1)$$

Where,

H_i = air absolute humidity in cell no. i

H_{Si} = absolute humidity near wall of the desiccant bed in cell no. i

V_{Hi} = the specific volume of humid air (m^3/kg)

G_a = mass flow rate of humid air ($\text{kg}_{\text{dry air}}/\text{m}^2 \text{ s}$)

t = time (s)

L = wheel thickness (m)

N = number of cell

k = mass transfer coefficient between the desiccant and the air per unit area ($\text{kg}/\text{m}^2 \text{ s}$)

ε = porosity of the wheel

A_h = internal surface area of each cell (m^2)

A_c = cross section area of each cell (m^2)

The mass flow rate of the air and the mass transfer coefficient can be calculated from:

$$G_a = \frac{U}{V_{Hi}} \quad (2)$$

$$\quad (3)$$

$$k = 0.704 G_a Re^{-0.51}$$

$$Re = \frac{\rho v D}{\mu} = \frac{U(\frac{4A}{P})}{V_{Hi}\mu} = \frac{G_a(4A_c)}{\beta\mu} \quad (4)$$

Where,

Re = Reynolds number

U = face velocity (m/s)

β = perimeter of square slot

μ = dynamic viscosity of the fluid (Pa·s or N·s/m² or kg/(m·s)).

3.4.1.3 Gas-Phase Energy Balance

The gas-phase energy balance can be shown by this equation (Ahmed M. Hamed 2005):

$$\begin{aligned} & \left(\frac{C_{Pa} + C_{pv}H_i}{V_{Hi}} \right) \frac{dT_{ai}}{dt} \\ &= \frac{NG_a}{L} (C_{Pa} + C_{pv}H_{i-1})(T_{ai} - T_{Si}) - \frac{h_c N A_h}{L A_c \varepsilon} (T_{ai} - T_{Si}) + \frac{k A_h N}{L A_c \varepsilon} (H_i - H_{Si}) \\ & \times (C_{Pw}T_{Si} + C_{pv}(T_{ai} - T_{Si})) + \left(\frac{C_{Pa}}{V_{Hi}^2} \right) \frac{dV_{Hi}}{dt} \end{aligned}$$

$$h_c = 0.671 G_a Re^{-0.51} C_{pa}$$

Where,

C_{Pa} = specific heat of dry air (kJ/kg K)

C_{Pv} = specific heat of water vapour (kJ/kg K)

C_{Pw} = specific heat of liquid water (kJ/kg K)

T_{ai} = temperature of air in cell no i (K)

T_{Si} = temperature of desiccant in cell no i (K)

h_c = heat transfer coefficient between the gas and solid phase (kW/m² K)

3.4.1.4 Solid-Phase Moisture Balance

The rate of change of the moisture content of the desiccant can be shown by the amount of moisture transferred from the air to the desiccant (Ahmed M. Hamed 2005):

$$\frac{dW_i}{dt} = \frac{kNA_h}{LA_c\rho_d}(H_i - H_{Si}) \quad (6)$$

Where,

W_i = desiccant content moisture content in cell no i (kg_{water}/kg_{dry desiccant})

3.4.1.5 Solid-Phase Energy Balance

The solid phase energy balance is shown by this equation:

$$(\rho_f C_{pf} + \rho_d C_{pd} + \rho_d C_{pw} W_i) \frac{dT_{Si}}{dt} = \frac{kA_h N}{LA_c} (H_i - H_{Si}) (Q - C_{pw} T_{Si}) + \frac{hA_h N}{LA_c} (T_{ai} - T_{Si}) \quad (7)$$

Where,

Q = latent heat of vaporization (kJ/kg_{water})

C_{pf} = specific heat of wall material (kJ/kg K)

C_{pd} = specific heat of desiccant (kJ/kg K)

ρ_f = Apparent density of honeycomb, kg/m³

ρ_d = Apparent density of desiccant, kg/m³

The above four equations have six unknowns which are H_i , T_{ai} , W_i , T_{Si} , H_{Si} , and V_{Hi} . Therefore two auxiliary relations are needed. The two relations define the specific volume of air in terms of humidity and temperature, also the humidity is defined in terms of vapour pressure. The specific volume of humid air can be shown in terms of air temperature and humidity as follow:

$$V_{Hi} = (a + bH_i)(T_{ai}) \quad (8)$$

Where,

V_{Hi} = specific volume of humid air (m³/kg)

$a = 0.00283$

$b = 0.00456$

The humidity on desiccant surface can be evaluated in terms of vapor pressure as:

$$H_{Si} = \frac{0.622P_{Vi}}{1.0133 \times 10^5 - P_{Vi}} \quad (9)$$

Where,

P_{Vi} = vapour pressure on the desiccant surface

The absorption equilibrium relation of the liquid desiccant that is used as a working fluid is correlated for calculation purposes. The actual mathematical relationship between the equilibrium thermo-physical properties such as vapour pressure, temperature, and

concentration is rather complex. However, if the concentration (x) is constant, the relation between vapour pressure and solution temperature can be shown as:

$$\ln p_{vi} = A(x_i) - \frac{B(x_i)}{T_{Si} - 273} + \frac{C(x_i)}{(T_{Si} - 273)^2} \quad (10)$$

Where $A(x_i)$ and $B(x_i)$ are concentration dependent parameters and can be expressed as a linear function of (x) as:

$$A(x_i) = a_0 + a_1 x_i \quad (11)$$

$$B(x_i) = b_0 + b_1 x_i \quad (12)$$

$$C(x_i) = c_0 + c_1 x_i \quad (13)$$

The desiccant moisture content of the bed W_i can be shown in terms of concentration of the bed x_i as:

$$W_i = \frac{1 - x_i}{x_i} \quad (14)$$

Where a_0 , b_0 , c_0 , a_1 , b_1 , and c_1 are the regression constants and their values are given in Table 3.1 and depends on range of p_{vi} , T_{Si} , and x_i . Meanwhile, Table 3.2 shows the known values for previous equations.

Table 3.1 Constants for equation (10)-(12) (Ahmed M. Hamed 2005)

Constant	Value	Variable	Range
a_0	9.8043	P_{Vi}	10–200 mmHg
a_1	11.7292	T_{Si}	289–389 K
b_0	–265.906	x_i	0.2–0.5
b_1	2052.21		

c_0	-10743.8		
c_1	5414.7		

Table 3.2 Model running data

Parameter	Value	Parameter	Value
C_{pf}	1.92 kJ/kg K	ρ_f	946 kg/m ³
C_{pd}	2.8 kJ/kg K	ρ_d	1908 kg/m ³
C_{pa}	1.005 kJ/kg K	Q	2,500 kJ/kg
C_{pw}	4.18 kJ/kg K	N	10
C_{pv}	2 kJ/kg K	ε	0.8

3.4.2 Overall System

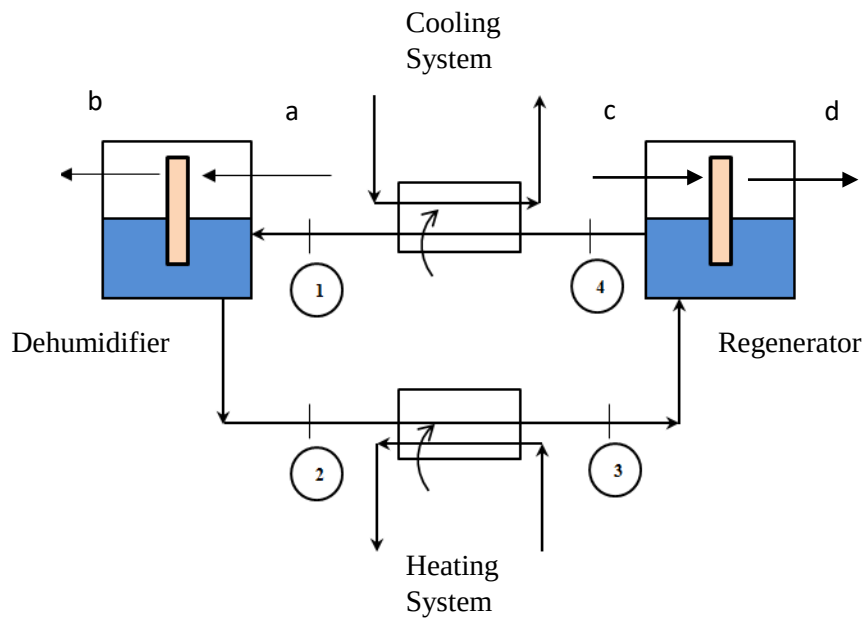


Figure 3.6 2-D schematic diagram of the system

Figure 3.6 shows the schematic diagram use to model the performance of the whole system. Cooling capacity (CC) is the measure of a cooling system's ability to remove heat (Ahmed H. Abdul-Salem 2013). CC can be expressed as the following equation:

$$CC = \dot{m}_{air} \cdot (h_a - h_b) \quad (15)$$

Where,

h_a = enthalpy of fresh air (kJ/kg)

h_b = enthalpy of supply air (kJ/kg)

$\dot{m}_{air}$ = mass flow rate of air (kg/s)

Therefore, we can calculate the COP of the overall system, which can be shown in in the next expression:

$$COP_{Total} = \frac{CC}{E_{Total}} \quad (16)$$

Where E_{Total} is energy consumption (kW) by heating, cooling, and auxiliary component. It can be determined after we calculate all energy consumed by each component individually. We can determine total energy consumption in kW or kJ/kg (E_{Total})

$$E_{Total} = E_{cool} + E_{th} + E_{aux}$$

$$E_{aux} = (E_{fan} + E_{pump} + E_{motor}) \quad (18)$$

Where,

E_{th} = energy consumption from heating loop (kJ/kg)

E_{cool} = energy consumption from cooling loop (cooling tower) (kJ/kg)

E_{aux} = energy from electrical component such as pump, fan and motor

3.4.2.1 Dehumidifier

In dehumidifier section (process 1–2 and a–b), we can determine COP of the dehumidifier (COP_{deh}) if we can investigate the property of air, such as absolute humidity and temperature before and after entering rotor wheel using equation (1) to (4). Using those properties, we can search in the property table of air and determine several enthalpies involved.

Here is the equation for COP_{deh} :

$$COP_{deh} = \frac{\dot{m}_{air} \cdot (h_a - h_b)}{E_{th} + E_{cool} + E_{aux}} \quad (19)$$

Where h_b is enthalpy of dry air and E_{th} , E_{cool} and E_{aux} is energy consumed from heating, cooling, and auxiliary component, respectively.

3.4.2.2 Heating Section

In the heating section (process 2–3), the heat transfer rate, Q_{th} required from the heating loop to the desiccant solution and also thermal energy consumption, E_{th} can be obtained by following expression:

$$Q_{th} = \dot{m}_{sol} \cdot C_{p,sol} (T_3 - T_2) \quad (20)$$

$$E_{th} = \frac{Q_{th}}{COP_{HL}} \quad (21)$$

Where,

$\dot{m}_{sol}$ = mass flow rate of desiccant solution (kg/s)

$C_{P,sol}$ = heat capacity of desiccant solution (kJ/kg K)

T_2 = Temperature entering the heating loop heat exchanger (°C)

T_3 = Temperature leaving the heating loop heat exchanger (°C)

COP_{HL} = Coefficient of Performance of heating loop

For the sake of calculation, the value COP_{HL} is usually set to 5.

3.4.2.3 Regenerator

In the regenerator section (process 3–4 and d–e), COP of the regenerator (COP_{reg}) was represented by the following equation:

$$COP_{reg} = \frac{\dot{m}_{air} \cdot (h_d - h_c)}{E_{th} + E_{cool} + E_{aux}} \quad (22)$$

Where,

h_c = enthalpy of recirculate air (kJ/kg)

h_d = enthalpy of exhaust air (kJ/kg)

3.4.2.4 Cooling Section

Meanwhile, in the cooling section (process 4–1), heat transfer rate from desiccant solution to cooling loop, Q_{cool} , and E_{cool} can be expressed as following:

$$Q_{cool} = \dot{m}_{sol} \cdot C_{P,sol} (T_4 - T_1) \quad (23)$$

(24)

$$E_{cool} = \frac{Q_{cool}}{COP_{cool}}$$

Where,

T_1 = Temperature leaving the cooling loop heat exchanger (°C)

T_4 = Temperature entering the cooling loop heat exchanger (°C)

COP_{cool} = Coefficient of Performance of cooling loop

3.5 Liquid Desiccant Selection

The search for suitable materials is generally the first step in the development of all desiccant dehumidification and cooling systems. Different types of desiccant or adsorbent are commercially available, and the utilisation of the proper desiccant could improve the performance of the system. An adsorbent with high affinity for a polar substance, such as water, are termed ‘hydrophilic’. Non-polar adsorbent, termed ‘hydrophobic’, have more affinity for oils and gasses than for water. These adsorbent include activated carbons, polymers adsorbent, and silicates (Auwal Mukhtar Dodo, 2012). A hydrophilic adsorbent is a more attractive candidate for the proposed ventilation systems.

The selection of desiccant will have a significant effect on the design of the liquid desiccant cooling system. Liquid desiccant is the main carrier of moisture removed from the controlled air in the system. They provide a level of vapour pressure lower than water solution at the same temperature, allowing air that moves through this lower pressure solution to become dehumidified. Depending on the concentration and temperature, liquid desiccants can also achieve equilibrium with air at a fixed humidity level. Liquid desiccants also generally provide

higher capacity than solid alternatives, meaning they can remove higher amounts of moisture per unit of desiccant weight.

Vapour pressure in a liquid desiccant solution is typically proportional to heat levels, but inversely proportional to concentration levels. When the desiccant concentration is decreased, or the temperature is increased, vapour pressure will rise, while reversing the conditions will cause the pressure to drop. The effectiveness of the dehumidification process depends on this balance of factors. Careful considerations need to be given to maintain the required parameters for a successful dehumidification system.

3.5.1 Glycols

Glycols are regularly used in industrial equipment and have important advantages and disadvantages. Glycols, such as propylene glycol and triethylene (Figure 3.7), have low toxicity, and their compatibility with most metals has made them suitable to use in a liquid desiccant cooling system. However, the volatile effect of glycol makes it less practical in liquid desiccant cooling application. A mixture of 96% triethylene glycol and 4% water will dry air to the same dew point as a 42% lithium chloride solution. Nevertheless, in equilibrium, the molar concentration of the glycol in the air will be about 1% of the water vapour. In an HVAC application where a 10,200 m³/h liquid desiccant cooling system operates for 2,000 hours per year, the annual loss of triethylene glycol in the conditioner would be more than 4,500 kg. This fact alone has shown significant shows that glycols can bring harmful environmental effect if it used in the HVAC system (Carlos Eduardo et al., 2014).



Figure 3.7 Triethylene is one example of glycol

3.5.2 Halide Salts

Halide salts are one of the frequent choices in selecting liquid desiccant for the dehumidification system. Halide salts, such as lithium bromide and lithium chloride (Figure 3.8), are very strong desiccants. A saturated solution of lithium bromide can dry air to 6% relative humidity and lithium chloride to 11% (Lowenstein, 2008). However, halide salts are corrosive to most metals. According to Mei and Dai (2006), a lithium-based solution is the most stable desiccant with advantageously low vapour pressure, but its cost is slightly higher compared with others. A large amount of anhydrous salt for lithium bromide and chloride cost USD6.60 and USD9.68 per kilogram, respectively. Although the tank for desiccant storage, such as uninsulated plastic tank, is relatively inexpensive, the cost of the lithium chloride and lithium bromide alone would discourage desiccant storage of more than a fraction of an hour. Calcium chloride is the cheapest and readily available desiccant, but its vapour pressure at a given temperature is relatively high. Due to unstable conditions, its application is limited depending on inlet air conditions and desiccant concentration in solution.



Figure 3.8 Lithium chloride is one example of Halide salts

3.5.3 Salts of Weak Organic Acids

The salts of weak organic acids are less corrosive and non-volatile compare to halide salts. The examples of these salts are potassium formate, sodium formate, and acetate. Formate salts, such as potassium formate, have the advantage of being significantly less viscous than acetate salts at concentrations with equivalent equilibrium relative humidity. Several companies market potassium formate as a low-viscosity heat transfer fluid and biodegradable de-icing solution (Dynalene, 2001). Although potassium formate is a significantly weaker desiccant than lithium bromide or lithium chloride, the ability to dry air below 30% relative humidity could make it an excellent alternative desiccant solution in the liquid desiccant cooling application. Several authors have successfully used potassium formate, as shown in Figure 3.9, in the design of the liquid desiccant cooling system. Jardi and Riffat (2014) have used potassium formate (HCOOK) as a liquid desiccant in biomass-fuelled micro-scale tri-generation system. It has a flow rate of 3.5 l/s and the air intake of 0.12 kg/s, and capable of reducing the relative humidity from dehumidifier inlet to outlet from 87% to 57.1%. Mehta and Rane (2013) have used the solution for liquid desiccant-based solar air conditioning with an evacuated tube collector. The volume flow rate is up to 300 l/h and an accuracy of 0.5% of the

measured value. The system has a liquid concentration change from 18.5% and exit concentration of 70.4% mixture of potassium formate and water.



Figure 3.9 Potassium formate is one example of salts of weak organics acids

3.6 Desiccant Wheel Materials

The desiccant material selection process begun with several studies conducted over a field of different candidate materials. These materials include some commercially available desiccant materials and some composite desiccant materials developed and tested by other researchers in search of a desiccant material with high-adsorption and low-temperature regeneration properties. A brief description of the various candidate desiccant materials is presented below.

3.6.1 Metal Type

Metals are one of the most commonly used classes of engineering material. Metal alloys are common, and they are formed by combining a metal with one or more other metallic and non-metallic materials. The combination usually occurs through a process of melting, mixing, and cooling. The aim of alloying is to improve the properties of the base material in some desirable way. Metal alloy compositions are described in terms of the percentages of the

various elements in the alloy. The percentages of various elements in the alloy are measured by weight. Table 3.3 shows the important properties of metal in identifying a candidate for a heat exchange device. (Web 3.1, Web 3.2, Web 3.3 and Web 3.4).

Table 3.3 Physical properties of potential metal materials.

Metal type	Density - ρ - (10^3 kg/m^3)	Tensile Modulus - E - (GPa)	Tensile Strength - σ - (MPa)	Specific Modulus - E / ρ - (kJ/kg)	Specific Strength - σ / ρ - (kJ/kg)	Maximum Service Temperature ($^{\circ}\text{C}$)
Cast Iron, grade 20	7.15	100	140	13986.0	19.58	300
Steel, AISI 1045	8.03	205	585	25529.3	72.85	650
Aluminum 2045-T4	2.7	73	450	27037.0	166.67	250
Aluminum 6061-T6	2.7	69	270	25555.6	100.00	250
AISI Type 304 Stainless Steel	8	200	215	25000.0	26.88	1455
Titanium Grade 1	4.51	105	240	23281.6	53.22	1670

The traditional metal heat exchanger is mainly made of aluminium, copper, and their alloys, which can be shaped into a surface such as a plate or a tube. Based on P. Rodriguez (1997), to identifying candidate materials for metal, some considerations may affect this process. These considerations are high heat transfer coefficient, good tensile and creep properties, good fatigue, corrosion fatigue and creep-fatigue behaviour. David P. Haack (2001) has developed metal foam components to drive a new application of highly effective and multi-functional heat exchange devices. Metal foam materials have the potential to increase heat transfer rates from solid surfaces by conducting heat to the material struts and inducing a high interaction between the struts and a through-flowing fluid.

3.6.2 Ceramics Type

Ceramic material is an inorganic, non-metallic, often crystalline oxide, nitride or carbide material. Some elements, such as carbon or silicon, are also being considered ceramics. Ceramic materials are brittle, hard, strong in compression, and weak in shearing and tension. They withstand chemical erosion that occurs in other materials subjected to acidic or caustic environments. Ceramics generally can withstand very high temperatures, ranging from 1,000 °C to 1,600 °C. Glass is often not considered a ceramic because of its amorphous (non-crystalline) character. However, glassmaking involves several steps of the ceramic process, and its mechanical properties are similar to ceramic materials. Table 3.4 shows the essential properties of ceramic in identifying a candidate for a heat exchange device. (Perrti Aukerri, 1996, Web 3.5 and Web 3.6).

Table 3.4 Physical properties of potential ceramic materials

Ceramic type	Density - ρ - (10^3kg/m^3)	Tensile Modulus - E - (GPa)	Tensile Strength - σ - (MPa)	Specific Modulus - E / ρ - (kJ/kg)	Specific Strength - σ / ρ - (kJ/kg)	Maximum Service Temperature (°C)
Zirconium Dioxide	6.15	250	711	40650.4	115.61	2522
Magnesium Oxide	3.58	330	166.7	92178.8	46.56	2400
Silica Carbide	4.84	137	1625	28305.8	335.74	970
Silicon Nitride	3.28	287	483	87500.0	147.26	1553
Boron Nitride	2.3	100	83.3	43478.3	36.22	1440
Aluminum Oxide	3.98	413	665	103768.8	167.09	2114

Some research directed toward using materials such as ceramic for heat exchangers in other applications has been reported. Robert J. Kee (2011) conducts the model-based design and

experimental performance evaluation of an all-ceramic compact counter-flow microchannel heat exchanger. The design process is assisted by three-dimensional simulation of the fluid flow and conjugate heat transfer behaviour as well as traditional heat exchanger theory. A manufacturing process called PLIS enables cost-effective fabrication of relatively complex internal manifolds and micro-channels. Ceramic materials enable applications at high temperature and in harsh chemical environments that may be difficult to achieve with metal heat exchangers.

Meanwhile, Charles A. Lewinsohn (2012) describes principles of design, methods of fabrication, and joining techniques for ceramic compact heat exchangers for the integration of such heat exchangers into practical applications. Ceramic compact heat exchangers enable the integration of numerous high-temperature processes with high efficiency. Microchannel heat exchangers can be assembled by joining separate but identical microchannel plates. The principles of design for microchannel plates dictate that channel dimensions and geometries must be selected carefully to optimize performance and reliability.

3.6.3 Polymer Type

Polymers are materials made of long, repeating chains of molecules. The materials have unique properties, depending on the type of molecules being bonded and how they are bonded. Some polymers bend and stretch, like rubber and polyester. Others are hard and tough, like

epoxies and glass. The term polymer is often used to describe plastics, which are synthetic polymers. However, natural polymers also exist; rubber and wood, for example, are natural polymers that consist of a simple hydrocarbon, isoprene. Proteins are natural polymers made up of amino acids, and nucleic acids (DNA and RNA) are polymers of nucleotides-complex molecules composed of nitrogen-containing bases, sugars and phosphoric acid, for example. Table 3.5 shows the important properties of ceramic in identifying a candidate for a heat exchange device (Web 3.7, Web 3.8, Web 3.9 and Web 3.10)

Table 3.5 Physical properties of potential polymer materials.

Polymer type	Density - ρ - (10^3kg/m^3)	Tensile Modulus - E - (GPa)	Tensile Strength - σ - (MPa)	Specific Modulus - E/ρ - (kJ/kg)	Specific Strength - σ/ρ - (kJ/kg)	Maximum Service Temperature (°C)
Nylon 6/6	1.15	3.6	82	3130.4	71.30	100
Polyethylene (HDPE)	0.965	1.09	31	1129.5	32.12	129
Polypropylene	0.91	1.6	40	1758.2	43.96	130
Epoxy	1.25	3.5	69	2800.0	55.20	215
Phenolic	1.35	3.0	6	2222.2	4.44	120
Polyvinyl chloride	1.45	3.4	60	2344.8	41.38	260

The development of polymer heat exchangers has received significant attention in the last decade due to several favourable properties of polymers such as corrosion resistance, low weight, low cost, ease of manufacture and many others. Martinus. A. Arie et al. (2016) said polymer heat exchangers offer good anti-corrosion and antifouling properties. In his work, a cost-effective air-water polymer heat exchanger made of thin polymer sheets was fabricated and experimentally tested. It uses layer-by-layer line welding with a laser through an additive

manufacturing process. Compared to commercially available plain plate-fin surfaces, polymer heat exchanger offers equal or superior heat transfer coefficient performance for the same pressure drop. The results also show that wall thermal resistance is not a limiting factor for low thickness polymer heat exchanger. It only represents up to 3% of the total thermal resistance. Due to their advantage of low weight, low cost, antifouling and anti-corrosion properties as compared to metallic heat exchangers, polymer heat exchangers are promising alternatives to conventional metallic heat exchangers for specific applications.

Meanwhile, Moritz Hinze et al. (2017) has made a completely new approach, thermally conductive polymers as heat exchanger materials for low-temperature adsorption chillers to decrease energy consumption during the adsorption cycle. The authors prepared thermally conductive polymers based on a semi-crystalline polyamide 6 (PA6) with different filler contents. They evaluated their potential for using them as a material for heat exchangers in low-temperature adsorption chillers. By comparing the calculated heat capacity of the aluminium plate specimen with the PA6/expanded graphite-based plate specimens, the thermal energy consumed by the adsorber during the temperature changes within the adsorption and desorption cycles can be reduced by about 30%. As this reduction is based on the material characteristics of the heat exchanger, this could directly enhance the *COP* of a large scale adsorption chiller.

3.6.4 Composite

A composite material is composed of at least two materials, which combine to give properties superior to those of the individual constituents. Fibre-reinforced polymer composites usually come with carbon, glass, aramid, polymer or natural fibres embedded in a polymer matrix. The composite can use other matrix materials and may also contain fillers or nano-materials such as graphene. The variety of component materials and different processes available make composites extraordinarily versatile and efficient. They typically result in lighter, stronger, more durable solutions compared to traditional materials. The properties of potential ceramic material for heat exchanger device is shown in Table 3.6 (M. S. EL-Wazerya, 2017, K.K. Herbert Yeung and K.P. Rao, 2012 and Web 3.11).

Table 3.6 Physical properties of potential composite materials

Composite type	Density - ρ - (10^3kg/m^3)	Tensile Modulus - E - (Gpa)	Tensile Strength - σ - (Mpa)	Specific Modulus - E/ρ - (kJ/kg)	Specific Strength - σ/ρ - (kJ/kg)	Maximum Service Temperature (°C)
Glass-filled epoxy (35%)	1.9	25	300	13157.9	157.89	200
Glass-filled polyester (35%)	2.0	15.7	130	7850.0	65.00	125
Glass-filled nylon (35%)	1.6	14.5	200	9062.5	125.00	110
S-glass epoxy (45%)	1.8	39.5	870	21944.4	483.33	215
Carbon epoxy (61%)	1.6	142	1730	88750.0	1081.25	215
Kevlar epoxy (53%)	1.35	63.6	1100	47111.1	814.81	215

Andres S. Valarezo (2019) has conducted an experimental investigation on the performance of novel composite desiccant Silica Gel and Sodium Acetate Coated Heat Exchanger (SASCHE) for typical summer and winter seasons in the city of Shanghai, China. Then, its performance is compared with Single Silica Gel Coated Heat Exchanger (SCHE), and Potassium Formate and Silica Gel Coated Heat Exchanger (PSCHE). During the summer season, the dehumidification capacity of SASCHE is 10% and 30% higher than the dehumidification capacity of PSCHE and SCHE, respectively. During the winter season, the results proved that SASCHE could humidify and heating the air, supplying acceptable air to satisfy indoor comfort. Low hot water temperature and high inlet air velocity are recommended as optimum operating conditions during the winter season.

Meanwhile, L.M. Hu et al. (2015) analyse the performance of composite desiccant coated heat exchanger (DCHE) by comparing with conventional silica gel DCHE. As a result, the adequate dehumidification time is shorter for silica gel DCHE than composite DCHE. Composite DCHE has higher moisture removal ability than silica gel DCHE. Under the specific condition, the dehumidification capacity of DCHE increases with the raise of inlet air relative humidity, as well as the increase of inlet air temperature. The difference dehumidification capacity between composite DCHE and silica gel DCHE is more significant under low relative humidity.

3.7 Comparison of the Material Types and Results Discussion

Table 3.7 summarises the level of performance of the earlier mentioned materials, in terms of thermal conductivity, specific modulus, specific strength, hardness, durability, and cost.

Table 3.7 Properties summaries of the potential material types

Materials	<i>Thermal conductivity (W/m K)</i>	<i>Specific Modulus - E / ρ – (kJ/kg)</i>	<i>Specific Strength - σ / ρ – (kJ/kg)</i>	<i>Hardness (shaping ability)</i>	<i>Durability</i>	<i>Cost (USD)*</i>	<i>Corrosion</i>
Metal	High	10-27	0.02-0.2	High	Long time	1.8-315	Highly likely
Ceramic	Variable	20-132	0.02-0.3	High	Long time	1.9-150	Unlikely
Polymer	Low	1-3	0.04-0.07	Medium	Medium	1.1-4.9	Unlikely
Composite	Variable	7.8-90	0.1-1.1	Medium	Medium	3.0-110	Variable

* Cost per kg

From Table 3.7, the analyses made are as follows:

- Metal has high thermal conductivity, good specific strength and specific modulus. However, metal tends to be somewhat costlier than non-metallic materials, and they are often vulnerable to corrosion damage as the metals react with their environment to re-form those compounds.
- Ceramic is the perfect materials as the heat/mass transfer membranes with sturdiness and high absorption. However, they are all harder and more expensive than polymer type.
- Polymer exhibit good mechanical properties, moderate durability and hardness. It also has excellent chemical resistance that is important when working in a corrosive environment. It's also one of the cheapest material available in the market

- Carbon is the best selection to produce heat/mass exchange device, which can offer heat/mass transfer and can be easily shaped into different structures. However, the cost of the material is an issue, as it can cost 50 times more than the polymer type.

3.8 Material Selection for Other Fabricated Components

Besides the main component, such as working fluid and rotor wheel materials, the components selection also emphasis on the casing, pumps, and piping system. It was crucial to ensure each component used suitable materials so that problems such as corrosion, leaks, and fire do not occur during the test.

3.8.1 Wheel Enclosure

Wheel enclosure is one of the essential parts of the wheel desiccant cooling system. It is a part that contains the working fluid and wheel while dehumidification and regeneration take place. The enclosure needs to be strong as well as inert to a chemical reaction such as potassium formate. As a result, we chose aluminium as the enclosure material as it is the most widely used metal in the world after iron (Figure 3.10). The properties of aluminium include low density that leads to low weight, high strength, superior malleability, easy machining, excellent corrosion resistance, and good thermal and electrical conductivity.



Figure 3.10 Aluminium plates

The enclosure needs to be as light as possible so that it is easily transported from one testing position to another, even though it is not one of the main priorities. The properties of aluminium are light, with a density of one-third that of steel, 2,700 kg/m³. The low density of aluminium accounts for it being lightweight, but this does not affect its strength. Aluminium commonly has a tensile strength of between 70 MPa and 700 MPa. Unlike most steel grades, aluminium does not become brittle at low temperature, but instead its strength increases. At high temperatures, aluminium's strength decreases. So, it is crucial to ensure that the regeneration temperature does not exceed 100 °C (Web 3.12).

Another reason why aluminium was selected is that machining an aluminium plate is easier compared to other materials. Aluminium is easily worked using most machining methods like milling, drilling, cutting, punching, and bending. In fact, the energy input during machining is low. Furthermore, aluminium also features easy joining with other aluminium to fit the profile design. Fusion welding, friction stir welding, bonding, and taping are also used for joining aluminium materials together.

Besides that, aluminium is also corrosive resistant. Aluminium can react with oxygen in the air to form an extremely thin layer of oxide. Although this layer is extremely thin, it is dense and can provide excellent corrosion protection. The layer is also self-repairing if damaged. Anodising is one technique that increases the thickness of the oxide layer and thus improves the strength of the natural corrosion protection. When aluminium is used outdoors, thicknesses of between 15 µm and 25 µm (depending on wear and risk of corrosion) are common.

3.8.2 Pipes and Connectors

The piping system is one of the most crucial parts of this project. It channels desiccant solution from the dehumidifier to the regenerator and vice versa. These pipes were made by (Acrylonitrile butadiene styrene (ABS), as shown in Figure 3.11. ABS pipes are resistant to most acids, alkalis, and salts. However, they are not resistant to aromatic and chlorinated hydrocarbons. ABS pipe can be used above or below the ground, but ABS is more likely to deform when exposed to the sun. Because of this, some local regulations require ABS pipes to contain pigments to protect it from UV radiation or to be painted with latex paint (Web 3.13).



Figure 3.11 ABS pipes

Compared to PVC, ABS is easier to install, as PVC pipes need a purple primer before each joint is glued together, and the joints must then be held together for 5–10 seconds for the glue to take hold. In term of strength, ABS piping has a higher impact strength than PVC, especially at lower temperatures. However, ABS can deform under sun exposure. ABS is highly durable with high impact strength. PVC is less durable, as it is designed to be flexible and softer than usual plastics. However, both plastics are resistant to chemical and water degradation. In the marketplace, ABS is usually cheaper than PVC.

3.8.3 Magnetic Pump

A pump is one of the essential components of this system because it transports fluid from one place to another. Centrifugal pumps are a sub-class of dynamic axisymmetric work-absorbing turbomachinery, as shown in Figure 3.12. It transport fluids by the conversion of rotational kinetic energy to the hydrodynamic energy of the fluid flow. The rotational energy typically comes from an engine or electric motor. The fluid enters the pump impeller along or near to the rotating axis. It is accelerated by the impeller, flowing radially outward into a diffuser or volute chamber (casing), from where it exists. A centrifugal pump moves the fluid through the use of centrifugal force. The fluid is taken into the centre of the impeller through the inlet connection.

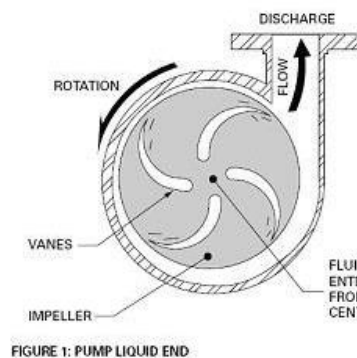


Figure 3.12 Centrifugal Pump

Most centrifugal pumps prefer a positive inlet pressure to prevent cavitation. The vanes of the impeller then catch this fluid as it spins. This rotation of the fluid mechanically by the vanes forces the fluid to the outside of the impeller. Then it goes towards the discharge port of the liquid end of the pump. This mechanical movement of the fluid creates the discharge pressure of the pump. Variables, such as inlet fluid supply pressure, impeller diameter, motor horsepower, and closed face versus open face, all affect the flow and pressure of the pump. Each of these variables can be manipulated to achieve the desired flow or pressure. However,

the centrifugal pump has its drawbacks. Suspended solids can worsen wear on the propeller of these pumps. Corrosion also plays a major issue. Corrosion inside the pump is caused by fluid properties, especially from a chemical fluid. Overheating due to low flow and leakage along the rotating shaft also a drawback of this kind of pump (Web 3.14).

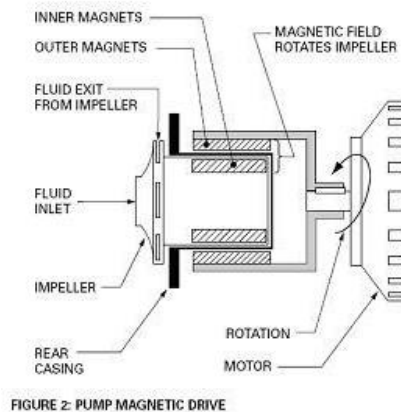


Figure 3.13 Magnetic-drive pump

In order to eliminate this problem, a different kind of pump was used, which is a magnetic drive pump, as shown in Figure 3.13. A magnetic drive pump uses a balanced magnetic field to create the rotation of the fluid impeller. Unlike a traditional centrifugal pump that has a direct drive connection between impeller and motor, a magnetic-drive pump eliminates the direct drive mechanism and replaces it with a magnetic field. An outer magnetic bell housing is mounted on the end of the pump shaft. This outer bell is aligned on the outside of the rear casing. The pump impeller is connected to a smaller magnet assembly and rides on an internal shaft and bushing assembly. The liquid end parts are all isolated within the fluid head of the pump without the need for a mechanical seal. The smaller magnet assembly is mounted within the centre of the magnetic field of the outer bell housing. Although a fluid barrier separates these two magnet assemblies, the magnetic fields are aligned. When the pump motor is started, the outer bell housing begins to rotate. As the outer bell rotates, the rotating magnetic field effects

the inner impeller magnet. As the two magnets begin to turn together, the impeller begins turning and displacing fluid (Wilden, 2004). In the end, this pump was selected for this experiment. It is due to the desired characteristics of the magnetic-driven pump, which is less corrosion when exposed to chemical interaction.

3.9 Summary

In this chapter, the mathematical modelling of the novel liquid desiccant wheel cooling system was carried out to assist us on evaluation purposes. It also helps us to simulate and predict the actual performance of the system that will be developed on the later stage.

Besides that, some of the most important materials also discussed in this chapter. We discussed major liquids used and are available in the market, which are glycol, halide salts, and salts of weak organic acids. After going through extensive research on the best working fluid for this project, we have selected potassium formate, a type of salt of weak organic acids. The main reason for this selection is that potassium formate does not corrode much and is non-volatile compared to halide salts. The price comparison of potassium formate with lithium chloride and lithium bromide is not very different. Also, it can dry air at very low relative humidity, so it is an ideal selection when dealing with air with really low relative humidity.

For the material selection of wheel rotor, four primary materials, namely metal, polymer, ceramic, and composite, were discussed in this chapter. Each type has its favourite configuration when being used as the heat and mass transfer medium. Since conductive heat transfer imposes minimal impact on the sensible heat transfer compared with convective heat transfer, the thermal conductivity of materials is an ignorable parameter in material selection.

Considering the heat and mass transfer capabilities of metal, ceramic, and polymer, they can become the best selection for wheel rotor. However, given their high cost, which is too expensive for this type of experiment, we decided against selecting ceramic and metal. Ceramic has one of the highest weight-to-size ratio, which limits its application in this experiment. A polymer is considered cheap and readily available in the marketplace. Besides, their capability in heat/mass transfer is not too far off compared to carbon and ceramic. Based on this consideration, we finalised our material selection for wheel by choosing polymer such as polypropylene for this project.

For other significant parts, the materials and component such as compartment enclosure, pipes, and pump were considered thoroughly. For compartment enclosure, aluminium was selected because it was one of the strongest, less corroding, and the cheapest available in the market. For piping, ABS pipe was chosen because it is resistant to most chemical liquids, easy to install, durable, and cost-efficient. Lastly, for the operated pump, we selected a magnetic-driven pump as it corrodes less compared to the centrifugal pump when dealing with a liquid such as potassium formate. As we can see, corrosion factor, as well as strength, played a massive factor in materials and components selection.

Chapter 4

Performance Investigation using Simulation Results

4.1 Introduction

In designing an excellent novel liquid desiccant wheel cooling system, the operating conditions and also geometry will play an important factor. The operating process for a novel liquid desiccant wheel cooling system is relatively simple. Still, the mass and heat transfer model in a dehumidifier is considerably complex and need an in-depth understanding and studies. In this chapter, we will discuss simulation steps that have been done using a commercial but relatively simple software called Engineering Equation Solver (EES). In the market, there is a lot of commercial software available for heat and mass transfer analysis. Still, as EES is well known for usage in thermodynamic studies as well as for its user-friendly interface, it was believed that EES is a suitable choice for analysis in this project.

Nevertheless, before running the experiment, one needs to understand the in-depth theoretical background of the novel liquid desiccant wheel cooling system that was discussed extensively in Chapter 3. Most modelling will use heat and mass transfer that happens inside the dehumidifier and regenerator. This mathematical modelling was studied by using many references from books and previous researches completed by several authors. The modelling was started by analysing the occurrence in the desiccant wheel, and this acts as the foundation of this research. Then, we looked at the broader picture by using this to analyse it as a complete

system. Finally, we can utilise the knowledge gained from the model to simulate it into the system that we can predict the overall performance of the system.

4.2 Simulation Using Engineering Equation Solver (EES)

Mathematical modelling discussed in Chapter 3 was simulated using EES, as shown in Figure 4.1. It is a general equation-solving program that can numerically solve thousands of coupled non-linear algebraic and differential equations. The program can also be used to solve differential and integral equations, do optimisation, provide uncertainty analyses, perform linear and non-linear regression, convert units, check unit consistency, and generate publication-quality plots. A major feature of EES is the high-accuracy thermodynamic and transport property database that is provided for hundreds of substances in a manner that allows it to be used with the equation solving capability (Web 4.1).

The equation that was developed in the theoretical background will be modelled in the EES's equation windows, as shown in Figure 4.2. Then the data table was created for the selected source code and generated the graph for analysis. In these equations, most of it was solved by Euler's method and 4th degree of Runge Kutta method. In this section, we simulated the results of dehumidifier extensively so we can evaluate the performance and size of that section.

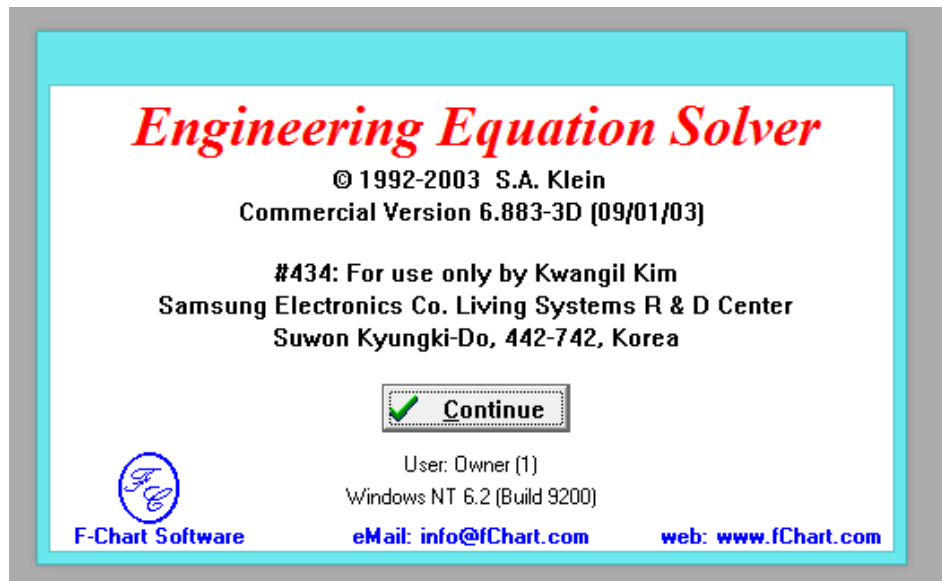


Figure 4.1 Engineering Equation Solver (EES) software

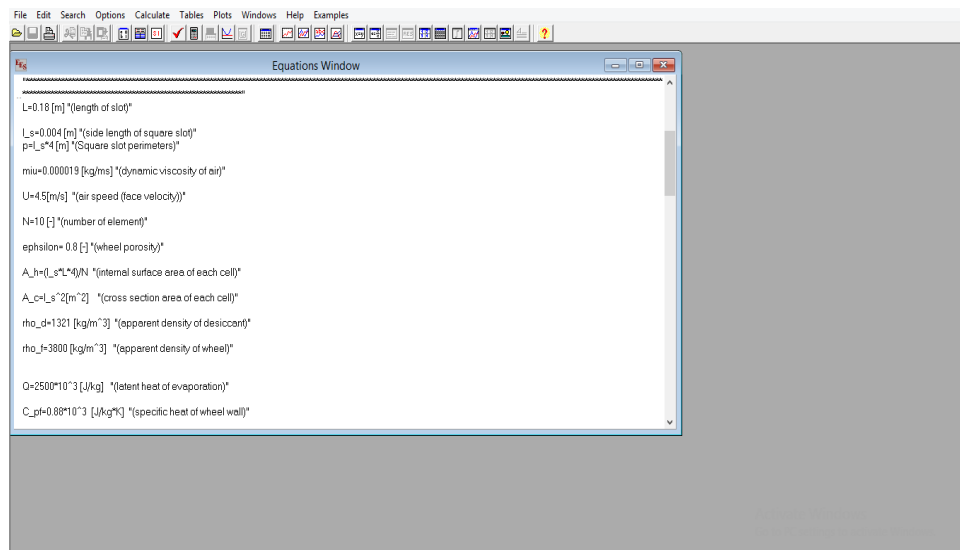


Figure 4.2 Modelling source code in the EES's equation windows

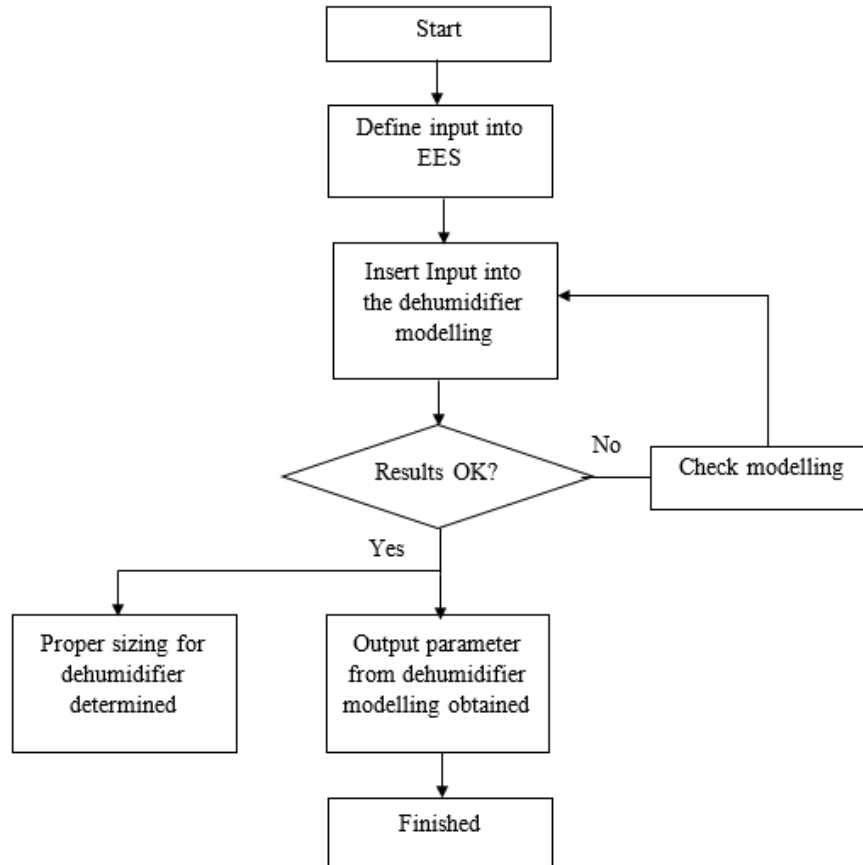


Figure 4.3 Process flow for simulation

Figure 4.3 shows the process flow for simulation of novel liquid desiccant wheel system. We start the simulation by the input parameters that were set before, which is relative humidity and temperature into the dehumidifier section. Then using psychrometric modelling in EES, we use these inputs and turn it into humidity ratios and enthalpies. Then we use this data and simulate it into the dehumidifier modelling that we developed. Using this modelling, we can determine the output parameter, such as relative humidity and temperature. Besides that, proper sizing for the dehumidifier section can also be calculated. If it found that the results are unsatisfying, the modelling will be recheck if there is an error occurred. If the results are satisfied, it means that we have obtained an appropriate sizing and parameter output for the system. A comparison with experimental results also been made so that the accuracy of the modelling can be determined.

4.2.1 Determination of Enthalpy Using EES Software

The sample calculation of air enthalpy at the given temperature, relative humidity, and constant atmospheric pressure can be done by using EES, as shown in Figure 4.4. By knowing all those data, we can key in the data and acquire the enthalpy at ease. It is vital as we can use the data from the dehumidifier and use it to calculate enthalpy for the overall performance. Figure 4.4 also shows the psychrometric chart for the required enthalpy. The simulation codes for this section can be referred to Appendix A.

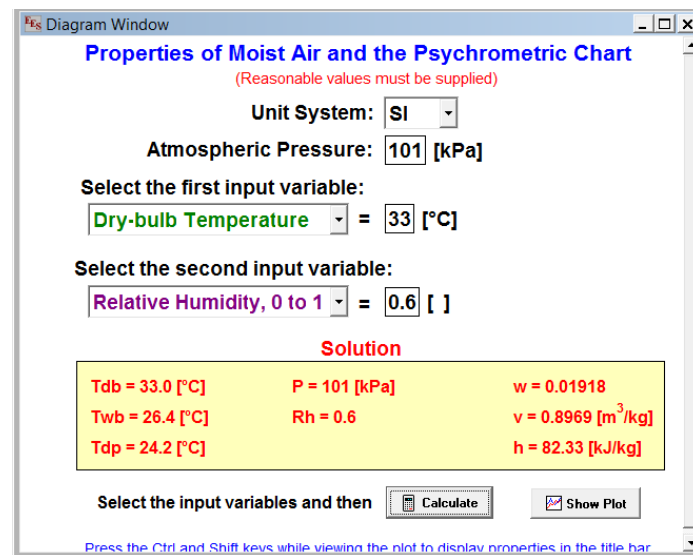


Figure 4.4 Sample calculation of air enthalpy using EES

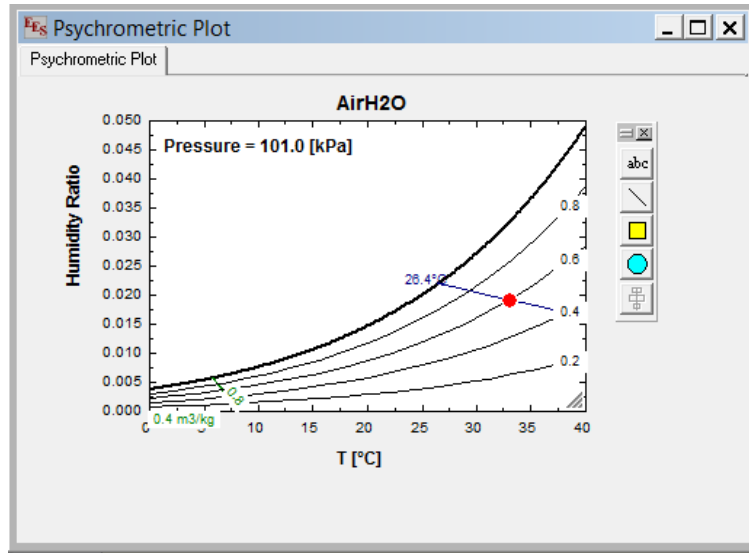


Figure 4.5 Psychrometric chart of air properties at selected air temperature and humidity

4.2.2 Simulation in the Dehumidifier Section

First, we wanted to make sure that we had selected the right liquid desiccant for this simulation. Figure 4.6 shows the influence of mass flow rate of solution ($\dot{m}_{sol}$) to the overall Coefficient of Performance of the system (COP_{total}) for potassium formate, lithium chloride and ethyl alcohol. As we can see, COP_{total} will decrease with increasing value of $\dot{m}_{sol}$. This is because a higher flow rate in heating and cooling heat exchanger will decrease the effectiveness of heat transfer needed to or from the liquid. As we can see from all three graphs, lithium chloride has shown the best performances, followed by potassium formate and ethyl alcohol. However, based on literature review, salts of weak organic acids, such as potassium, have been considered as less corrosive alternatives than any other liquids. Formate salts have the advantage of being considerably less viscous than acetate salts at concentrations with equivalent equilibrium relative humidity (Lowenstein 2008).

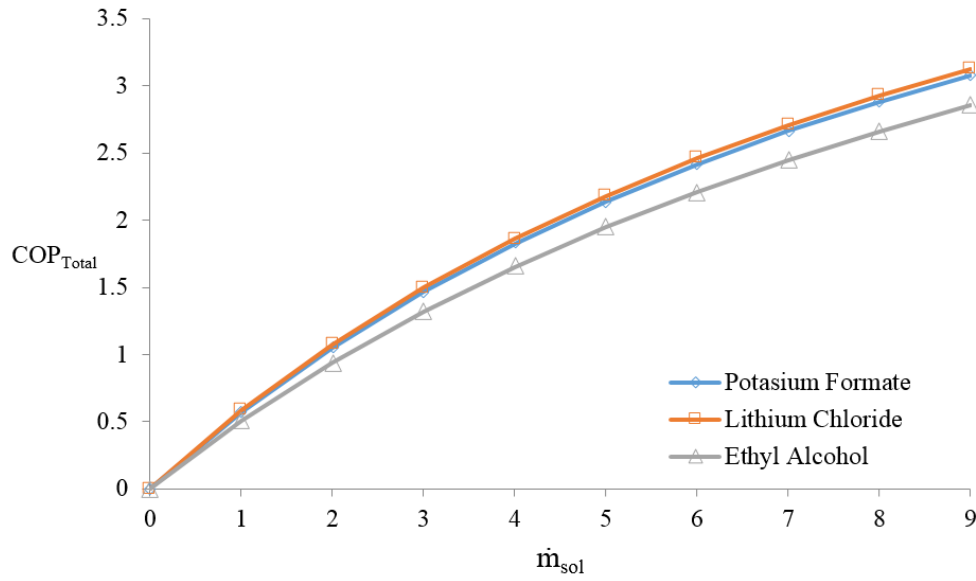


Figure 4.6 Overall COP versus mass flow rate of solution

After determined the suitable liquid desiccant solution for this study, we simulated the performance output of the dehumidification section. In this study, we run the simulation by altering the parameters in the dehumidification section and seeing its effect on the absorption performance in the system. The parameters to be changed are the speed of humid air, the length of the wheel slot, and the size of the wheel slot. Firstly, we varied the parameters of the length of the wheel slot. The wheel slot is the small holes in the desiccant wheel. During the dehumidification process, these small holes will carry the desiccant material before it interacts with the air stream. In this simulation, five different lengths of wheel slots have been selected, which are 2 cm, 6 cm, 10 cm, 14 cm, and 18 cm. From the simulation results, the graph of humidity ratio ($kg_{water}/kg_{dry\ air}$) against time with five different values of the length of the wheel slot has been plotted, as shown in Figure 4.7.

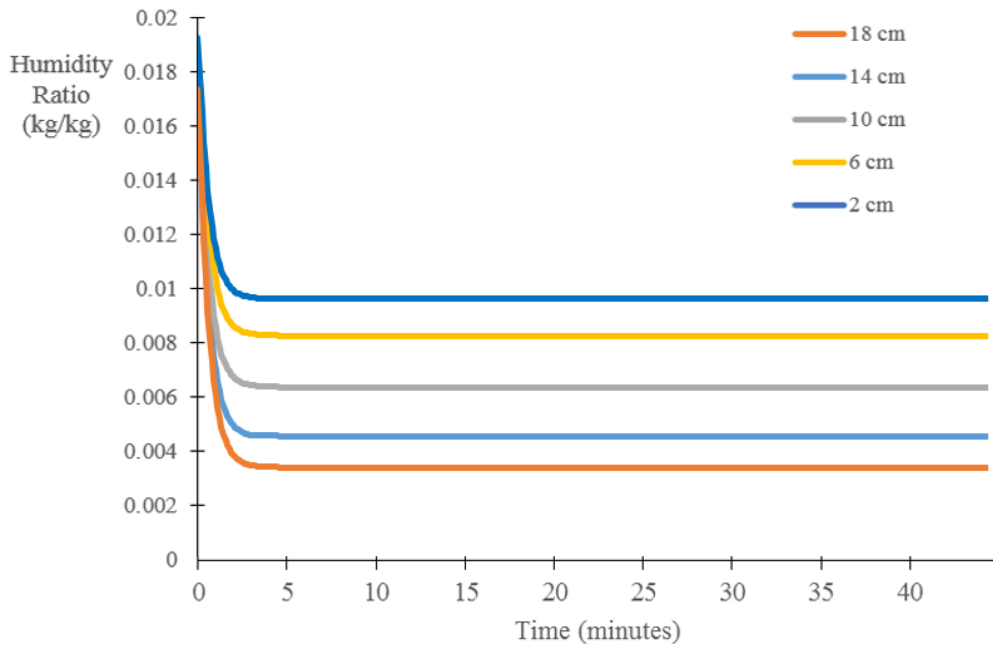


Figure 4.7 Graph of humidity ratio against time with varied length of wheel slot

Table 4.1 Percentage differences of humidity ratio between each length of wheel slot

Gap (cm)	Upper limit (kg _{water} /kg _{dry air})	Lower limit (kg _{water} /kg _{dry air})	Percentage (%)
2-6	0.009733	0.008218	15.6
6-10	0.008218	0.006291	23.5
10-14	0.006291	0.004414	29.8
14-18	0.004414	0.003512	20.4

For the length of the 2 cm slot wheel, the ratio of humidity ratio is at around 0.009733 kg_{water}/kg_{dry air}. We were then followed by a 6 cm length slot, which shows humidity ratio around 0.008218 kg_{water}/kg_{dry air}. After that, the length slot of 10 cm reduces the humidity ratio to 0.006291 kg_{water}/kg_{dry air}. Then it is followed by the 14 cm length slot, which recorded humidity ratio equals to 0.004414 kg_{water}/kg_{dry air}. Finally, for the wheel length of 18 cm, the humidity ratio is at 0.003512 kg_{water}/kg_{dry air}, and it is also the lowest humidity ratio that can be achieved at this point. It shows that the length of the wheel slot influenced the performance of the system, whereby the longer wheel slot will increase the moisture absorption rate in the system. Then in Table 4.1, we can observe how much difference that length of the slot will make on the humidity ratio. As we can see that range between 10 cm to 14 cm will bring about

29.8 %, which is the most significant difference among other lengths. Then the smallest differences that we can record here is between 2 cm to 6 cm which is 15.6 % only.

Next, we look into the variation side length of the slot to the humidity ratio. In this simulation, five different sizes of wheel slot used; they are 2 mm, 4 mm, 6 mm, 8 mm, and 10 mm. From the simulation results, the humidity ratio against time with five different sizes of wheel slot has been plotted, as shown in Figure 4.8.

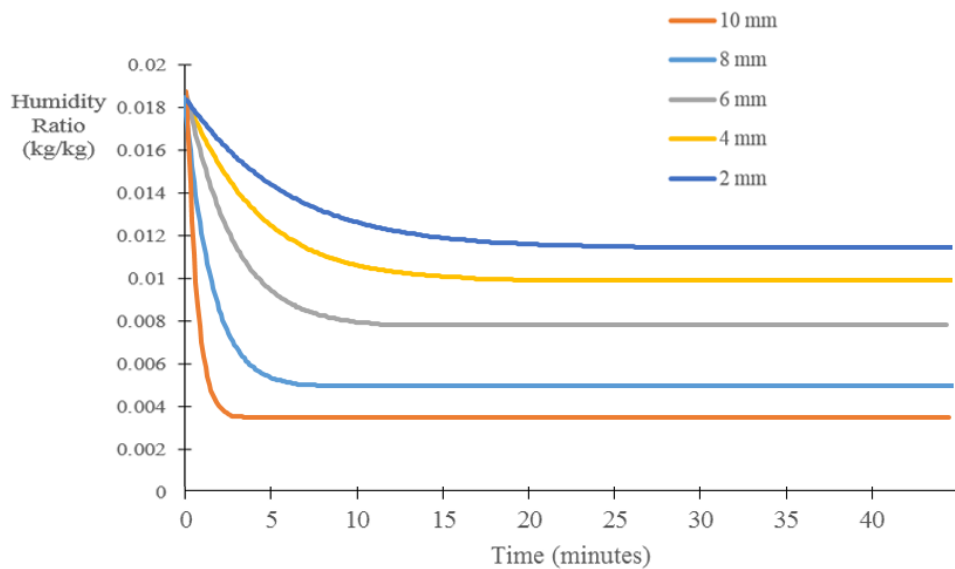


Figure 4.8 Graph of humidity ratio against time with varied side length of wheel slot.

Table 4.2 Percentage differences of humidity ratio between each side length of wheel slot

Gap (mm)	Upper limit (kg _{water} /kg _{dry air})	Lower limit (kg _{water} /kg _{dry air})	Percentage (%)
2-4	0.01193	0.01020	14.5
4-6	0.01020	0.007932	22.2
6-8	0.007932	0.005122	35.4
8-10	0.005122	0.003735	31.0

Initially, for the 2 mm wheel slot, the humidity ratio is the highest at 0.01193 kg_{water}/kg_{dry air}, followed by 4 mm side length that recorded humidity ratio at around 0.01020 kg_{water}/kg_{dry air}.

Then the humidity ratio for 6 mm side length of wheel slot is at 0.007932 kg_{water}/kg_{dry air}. Next, for a side length of 8 mm, the humidity ratio is 0.005122 kg_{water}/kg_{dry air}. Lastly, for the 10 mm side length of wheel slot, the humidity ratio value is the lowest at 0.003735 kg_{water}/kg_{dry air}. This simulation proves that the size of the wheel slot has a significant effect on the performance of the system; bigger-sized wheel slots cross-section increases the rate of moisture absorption in the system. From Table 4.2, we noted how much difference that side length of the slot would make on the humidity ratio. As we can see that range between 6 mm to 8 mm will brings about 35.4 %, which is the most significant difference among other lengths. Then the smallest differences that we can see here is between 2 mm to 4 mm which only managed around 14.5 %

Besides sizing, another important thing in simulation is to analyse the parameter input to the performance of the dehumidifier. In this simulation, five different humid air velocity been used; they are $v_1 = 2$ m/s, $v_2 = 4$ m/s, $v_3 = 6$ m/s, $v_4 = 8$ m/s, and $v_5 = 10$ m/s. From the simulation results, the ratio of humidity against time with five-velocity values of the humid air has been plotted, as shown in Figures 4.9.

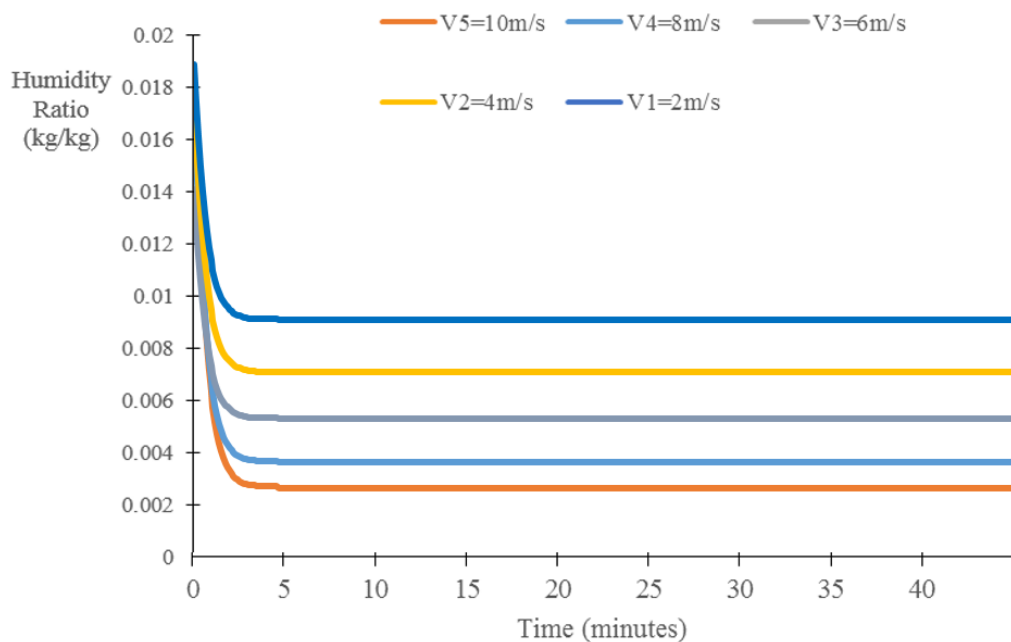


Figure 4.9 Graph of humidity ratio against time with varied air velocity

Table 4.3 Percentage differences of humidity ratio between each air velocity

Gap (m/s)	Upper limit (kg/kg)	Lower limit (kg/kg)	Percentage (%)
v_1-v_2	0.009399	0.007397	21.3
v_2-v_3	0.007397	0.005599	24.3
v_3-v_4	0.005599	0.003861	31.0
v_4-v_5	0.003861	0.002463	36.2

Based on the simulation, for humid air at the velocity of $v_1=2$ m/s, the humidity ratio was at 0.009399 kg_{water}/kg_{dry air}. Next, when air velocity increased to $v_2=4$ m/s, the humidity ratio decreased to 0.007397 kg_{water}/kg_{dry air}. When air velocity increased to $v_3=6$ m/s, the humidity ratio was around 0.005599 kg_{water}/kg_{dry air}. When the air velocity raised further to $v_4=8$ m/s, the humidity ratio reduced to 0.003861 kg_{water}/kg_{dry air}. At the highest velocity, which is $v_5=10$ m/s, the humidity ratio shows the highest drop, which was around 0.002463 kg_{water}/kg_{dry air}. It proves that air velocity affects the performance of the system. If the velocity of the air entering the system is increased, the rate of moisture absorption has become higher. Next, Table 4.3 shows that how much difference humidity ratio can be affected with different air velocity. As we can see that range between v_4 to v_5 obtained around 36.2 %, which is the most significant difference among other velocities. Then the smallest differences that we can see here is between v_1 and v_2 , which only achieved around 12.3 %.

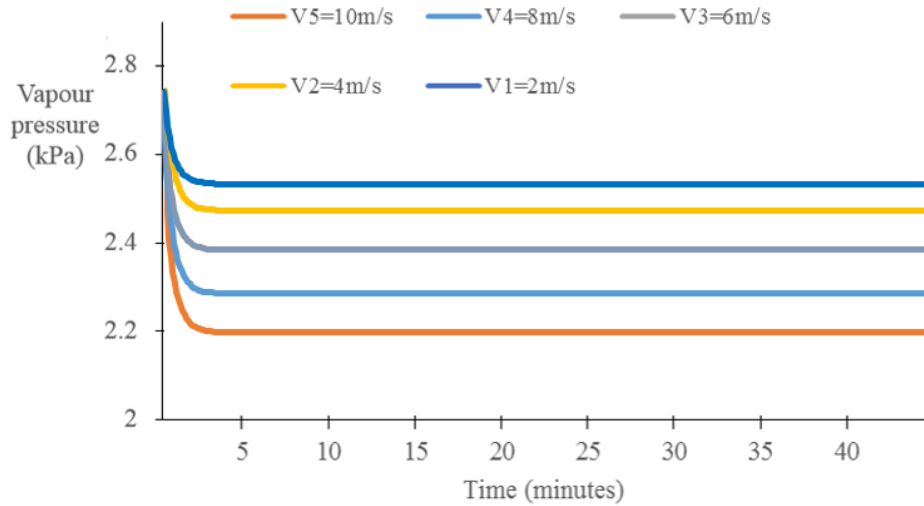


Figure 4.10 Graph of vapour pressure against time with varied air velocity

Table 4.4 Percentage differences of vapour pressure between each air velocity

Gap (m/s)	Upper limit (kPa)	Lower limit (kPa)	Percentage (%)
v_1-v_2	2.582	2.506	3.0
v_2-v_3	2.506	2.401	4.4
v_3-v_4	2.401	2.318	3.6
v_4-v_5	2.318	2.189	5.9

Next, we simulate the vapour pressure against time in the dehumidifier, as displayed in Figure 4.10. The vapour pressure is an indication of a liquid's evaporation rate. Based on the simulation, for the velocity of humid air at v_1 , the vapour pressure is the highest at 2.582 kPa. When air velocity decreased to v_2 , the vapour pressure increased to 2.506 kPa. The vapour pressure was around 2.401 kPa after air velocity was set to v_3 . When air velocity decreased again at v_4 , the vapour pressure increased to 2.318 kPa. Lastly, when the air velocity is at the highest, which is v_5 , the vapour pressure was the highest at 2.189 kPa. It shows that air velocity will affect vapour pressure in the wheel slots, which means that the higher the air velocity, the higher the vapour pressure drop that can be seen. A pressure drop was exhibit when air velocity at 10 m/s shows that higher tendency for particles to escape from the liquid desiccant state. Then, the percentage difference of vapour pressure can be affected by different air velocity is

shown in Table 4.4. As we can see that range between v_4 to v_5 obtained around 5.9 %, which is the biggest difference among different air velocities. Then the smallest differences that we can see here is between v_1 and v_2 , which only achieved around 3.0 % differences.

Based on these results, we can tell that air velocity plays a significant role in the dehumidification process. In term of sizing, the bigger the size of the wheel slot, the better the dehumidification will be. However, it is desirable that the size of the wheel must be as compact as possible. Therefore, the length of the slot and wheel around 18 cm was chosen so it can perform the best result without needing a bulky size. Thus, 18 cm is the right length for this wheel design. Then the side length of 8 mm was selected because we used the wheel matrix that is readily available in the market, and this is size also shows a good performance.

In terms of performance-wise, the dehumidifier section also performed well. It will produce the lowest humidity ratio at the highest air velocity, which is $0.002463 \text{ kg}_{\text{water}}/\text{kg}_{\text{dry air}}$. Even at the lowest velocity, it still can dehumidify air at quit a higher rate. In terms of vapour pressure, we can also say there is a significant drop in vapour pressure when we use the highest air velocity. The lower the vapour pressure, the better dehumidification of the system can be. In this simulation, it was found that the lowest vapour pressure that was produced is around 2.189 kPa.

4.3 Summary

A simulation of the novel liquid desiccant wheel cooling system that studies the performance in the dehumidification process has been developed. Detailed modelling studies of airflow in the wheel section and the overall performance also have been discussed. The simulation was

carried out under different inlet air velocity conditions. This simulation was restricted to specific running data to get a similar picture with the experimental result that will be obtained later. The simulation results show that the wheel performs adequate dehumidification under certain operating conditions. The higher the airflow coming into the wheel rotor, the better the overall performance of the dehumidifier. The sizing also can be obtained based on the simulation result that was conducted. However, we also need to take into account the availability of the material size so that it can be easily obtainable and also cost-effective.

This chapter also shows that the modelling used in the simulation can be used to predict the real output data of the wheel desiccant cooling system. The modelling is relatively simple but very efficient in simulating our desired output. Further analysis and modelling can be developed if the system was integrated with a more complex arrangement such as multi-stage or deep dehumidification process.

Chapter 5

Prototype Design and Experimental Set-Up

5.1 Introduction

As it was known, experimental data from the laboratory can provide the main sources of information about the system's behaviour. In this project, the experiment was carried out on wheel desiccant cooling rig in the Marmonth Laboratory in the School of Architecture and Built Environment, University of Nottingham. This chapter is presenting an outline of the design and construction of the experimental rig used to study the heat and mass transfer occurring in the novel liquid desiccant wheel cooling system. This chapter presents a detailed methodology and procedures followed in acquiring the experimental data. It also provides an overview of the experimental test rig facility and further information on important test rig components, such as wheel assembly, airflow, and liquid flow arrangement. Details of data acquisition, instrumentations, and calibration have also been discussed in this chapter.

5.2 Experimental Data Acquisition and Logging

Data acquisition and logging involve gathering information using a combination of computer-based measurement hardware and software to provide a flexible, user-defined measurement system. The software transforms the personal computer and the data-logging hardware into a complete data acquisition, analysis, and presentation tool.

As to be described in the following section, four humidity, temperature transducers, four thermocouples, and two air velocity transducers were needed to be powered simultaneously while also recording temperature, humidity, and air velocity from a various transducer. A large amount of data was to be collected in a short period, making it essential that these data were collected by and stored in a computer.

EK-H4 data logger software was installed on the computer. A general-purpose data taker, DT80, was utilised for data acquisition and conversion from the output voltage to physical value (temperature, relative humidity, and air velocity) based on calibration results obtained for each measurement probe. Voltage was recorded through a computer interface using the data logger software. Because data logger has the flexibility of a programming tool combined with additional built-in tools designed specifically for test, measurement, and control, an application was created, and measurement data were recorded at every one-second interval.

A schematic diagram of the arrangement used for the data collection is shown in Figure 5.1, and the data acquisition system was carried out using a program block assembled in the data logger software as shown in Figure 5.2. We also used data logger EKH4 for recording humidity and temperature of the air using SHT71 sensors and data logger DT80 for recording liquid temperature using K-type thermocouple, as shown in Figure 5.3. For SHT71 sensors in Figure 5.3 a), it has a temperature measurement range of $-40\text{ }^{\circ}\text{C}$ to $80\text{ }^{\circ}\text{C}$, with an accuracy of $\pm 0.2\text{ K}$. Meanwhile for its relative humidity, it has a measurement range of 0 to 100% with an accuracy of $\pm 1.7\%$. For this experiment, The K-Type thermocouple in Figure 5.3 b) has a temperature range of -40°C to 375°C with an accuracy of $\pm 0.5^{\circ}\text{C}$. Airflow rate across the input and output airstream was measured using air The OMEGATM HHF-SD1, as shown in Figure 5.3 c). It has combination hot wire, and standard thermistor anemometer with SD card

data logger has multiple features that make it suitable to use in such applications as environmental testing. Then we also have a concentration sensor in Figure 5.3 d). The concentration sensor will collect the data in density range, and further conversion needs to be done to gain the real concentration of the potassium formate. The data logger setup can be referred to in Appendix B.

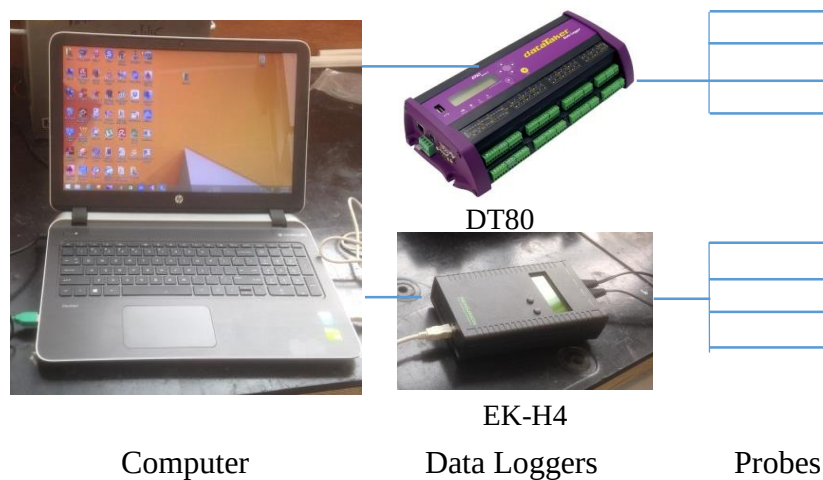
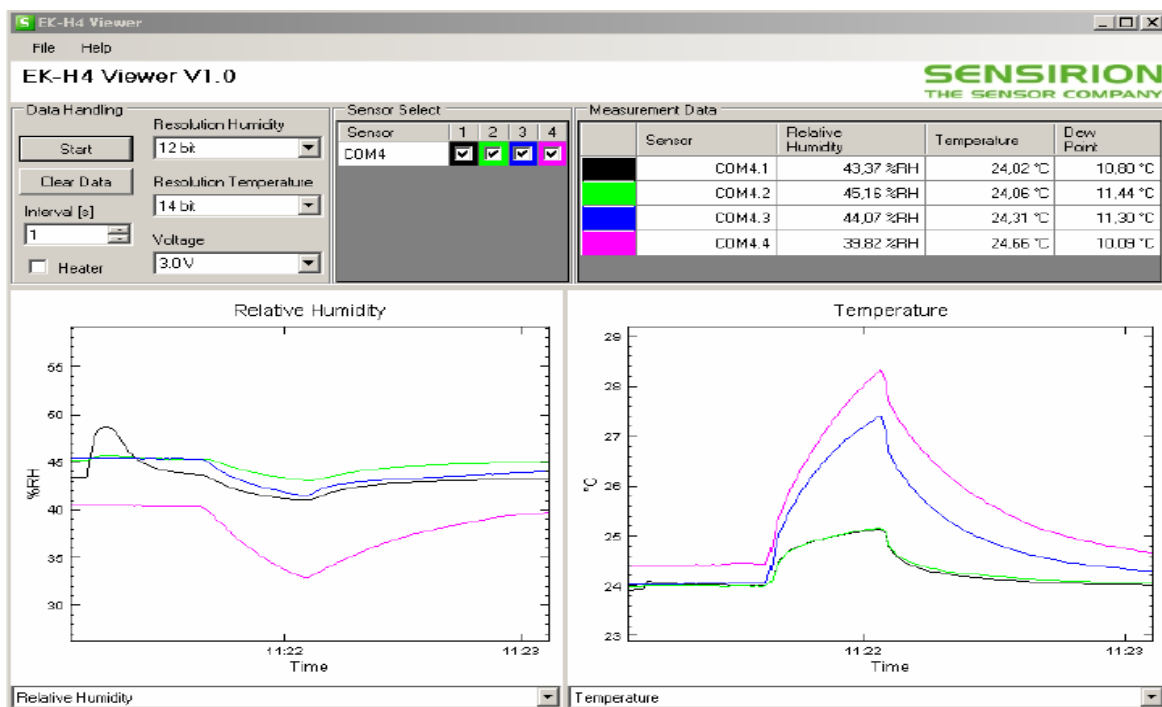
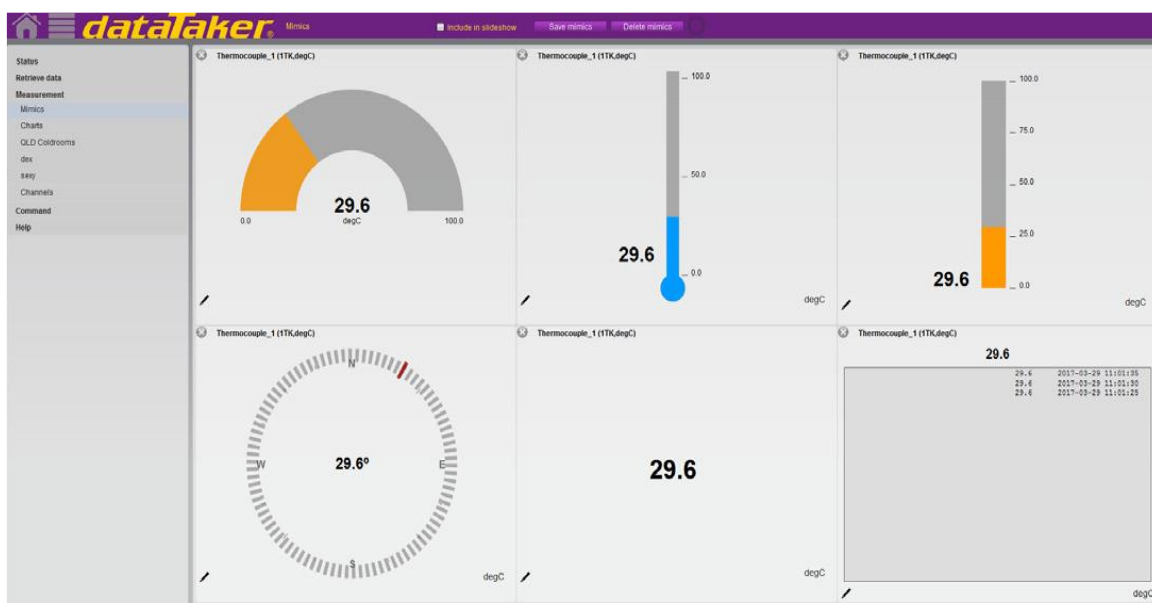


Figure 5.1 Sketch diagram of data logging arrangement

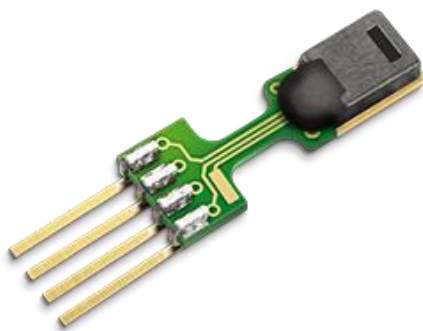


(a)

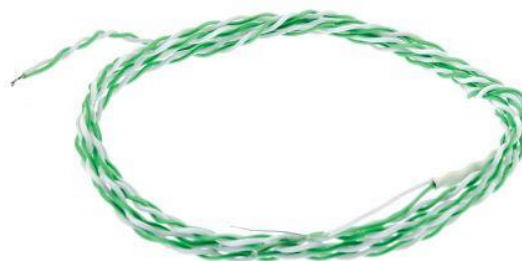


(b)

Figure 5.2 Data display for (a) EKH4 and (b) DT80 data acquisition system



(a)



(b)



(c)



(d)

Figure 5.3 Probes used for experiment: a) SHT71 sensor for EK-H4 b) K-Type thermocouple for DT80 (c) anemometer (d) concentration sensor

5.3 Testing with Complete System

Prior to commencing the experimental work, all fabrication work such as the wheels and the test rig was done in the workshop for about three months. This prototype needs to have specific criteria so that it can meet the expectation that has been set. All the details for the complete system are discussed in the following section.

5.3.1 Introduction

The novel desiccant wheel cooling system experiment will be the main focus of this project. This test was conducted in Marmonth Laboratory. The rotor wheel was placed in the built test rig and tested for heat and mass transfer performances. It was tested using an environmental chamber in the laboratory. This test was conducted to fulfil the objective of the project, which is to combine the rotor wheel technology that usually seen in solid desiccant with the liquid

desiccant mechanism. This experiment was conducted to study whether this system has a high *COP* compared to other air cooling technology in the market.

Based on the results from simulation, we can design the rotor wheel appropriately before installing it to the test rig. After the installation, the Area Safety Officer will check and scrutinise the test rig to make sure it is safe for the experiment. The most danger that they would like to avoid the most is chemical, electricity, or movement hazard. Then the rig was tested with the water as a working fluid to see whether any leakages were present prior to the real experiment.

5.3.2 Design

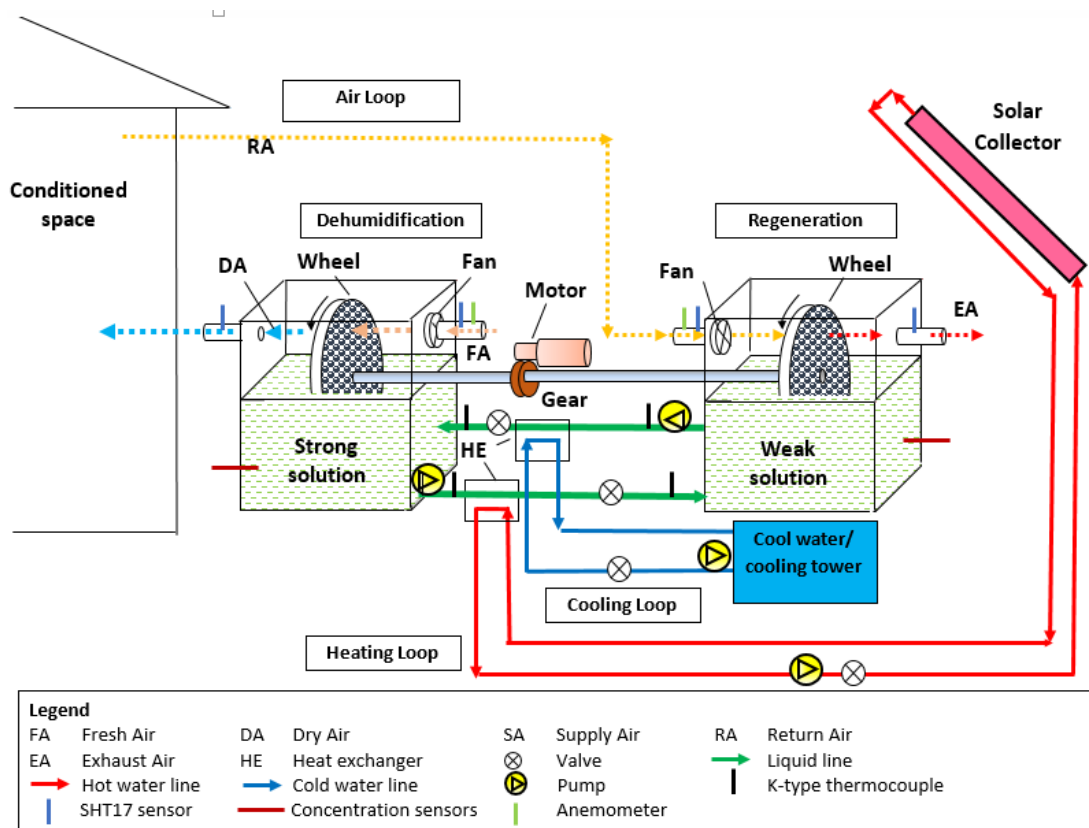


Figure 5.4 Schematic diagram of novel liquid desiccant wheel cooling system

The schematic diagram of the system is shown in Figure 5.4. All of the relevant components have been placed in this drawing. Even the locations of the relevant sensor also been shown in this figure. This drawing design is crucial that it needs to be detailed and understandable so that it can be used for the fabrication stage afterwards. Following is the explanation and cycle flow of this system.

i) Process 1–2

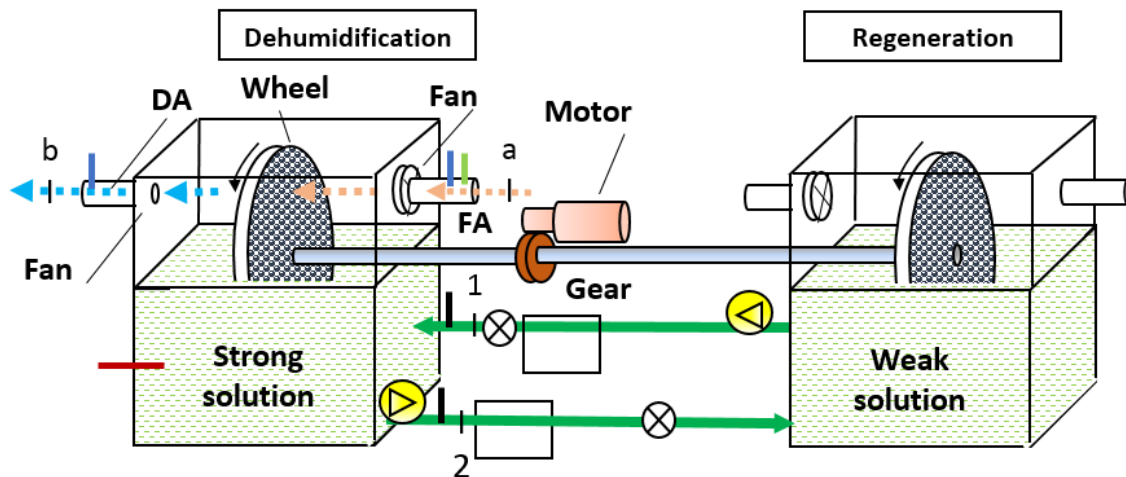


Figure 5.5 Dehumidification process 1–2 (liquid) and a–b (air)

During process ‘1–2’, dehumidification happens around the wheel, as shown in Figure 5.5. At 1 and 2, we put K-type thermocouples so that we can measure the initial and final temperature (short black line) at the dehumidifier. Here, the green line represents the flow path of the liquid desiccant. The pink arrow at ‘a’ represent the fresh air (FA), and the light blue arrow at ‘b’ represent the dry air (DA). The humidity and temperature sensors, SHT71 (short blue line) also been placed at the former and latter. These sensors were used to measure the temperature and humidity of air when entering and leaving the humidifier. A hand-held anemometer (short light green line) was used to measure the input air velocity. There is no need to put another anemometer at the output because it only gives the same air velocity as the input. Then, there was also a concentration sensor (short maroon line) that was placed in the tank that was used to measure the concentration level of liquid in the dehumidifier. At ‘1’, the strong liquid desiccant entered the dehumidification tank. Then, some of the strong liquid solutions were absorbed by the submersed wheel and was brought upward during its rotation. Then the FA,

which was extracted by the fan, passed through the liquid solution inside the wheel and the liquid absorbed the air's moisture. As a result, the air stream left the wheel as DA at 'b'. This wheel rotation was driven by the motor and gears that located outside of the dehumidifier. The wheel kept rotating until the solution in the tank becomes weaker and left at '2' by using the pump.

ii) *Process 2-3*

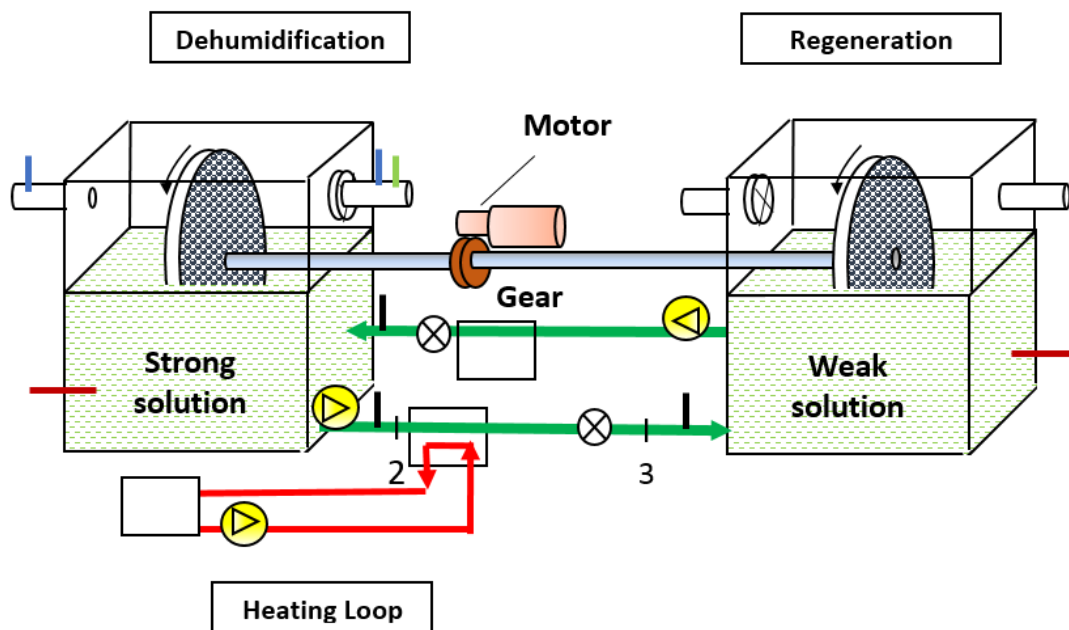


Figure 5.6 Heating Process 2–3 (liquid)

After that, the weak solution needs to be regenerated to reuse it again. Therefore, heating the solution will be done in process '2–3' as shown in Figure 5.6. Here, the K-type thermocouples have also been placed at '2' and '3', respectively. The weak solution at '2' entered and heated by hot water in the heating loop (red line) where heat interaction between two fluids took place

in the heat exchanger. Hot water can be obtained by using a flat-plate solar collector, which is widely used in the market, as it has good heat efficiency. Liquid desiccant required a low-grade heat source such as solar energy because potassium formate only needs to be regenerated at low temperatures between 50 °C and 60 °C. However, for the convenience, an electric water heater has been utilised so that a consistent heating temperature can be obtained. When the fluid is hot enough, a valve was open so that solution left at '3' before entering the regeneration tank.

iv) *Process 3-4*

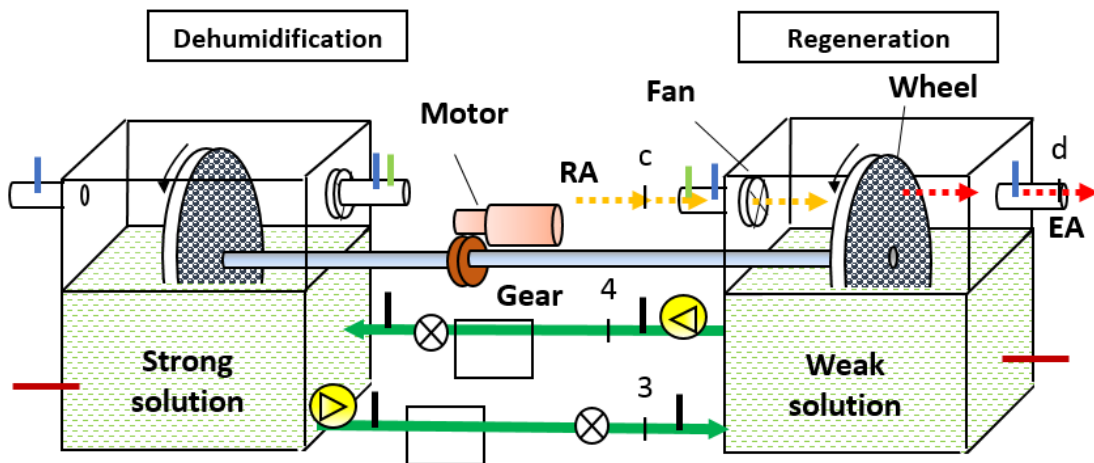


Figure 5.7 Regeneration process 3–4 (liquid) and c–d (air)

Figure 5.7 shows process '3–4' which enables the solution to remove excess moisture in the regeneration process. Just like at dehumidifier, several sensors have also been installed at the regenerator. The SHT17 sensors have been installed at 'c' and 'd', respectively. An anemometer has also been installed at 'c' so that we can measure the incoming air velocity. Besides that, a K-type thermocouple also located at '4' and just now at '3' as explained in the earlier process. The hot solution from '3' entered the regeneration tank, which has a similar

configuration to the dehumidification section. In the regeneration tank, most of the solution absorbed by the wheel. The solution encountered the airstream at the rotation speed that was similar to the wheel in the dehumidification section. In air stream, recirculated air (RA) at 'c' was the air that has been used in the building. The fan that was located in the regenerator input drove the RA (orange arrow). This air consists of less humid air and tends to absorb moisture when in contact with a weak and hot solution. As a result, the air became exhaust air (EA), which means the air is hot and more humid because of the moisture and heat transfer in the rotor wheel. The EA (red arrow) then were rejected to the surrounding at 'd'. Eventually, the weak solution then became strong again after the interaction with the air stream, and this process will continue until all solution became strong again. The concentration sensor has also been placed to measure the concentration level. After that, the solution can be channelled out from the tank to '4' using the pump.

iv) *Process 4-1*

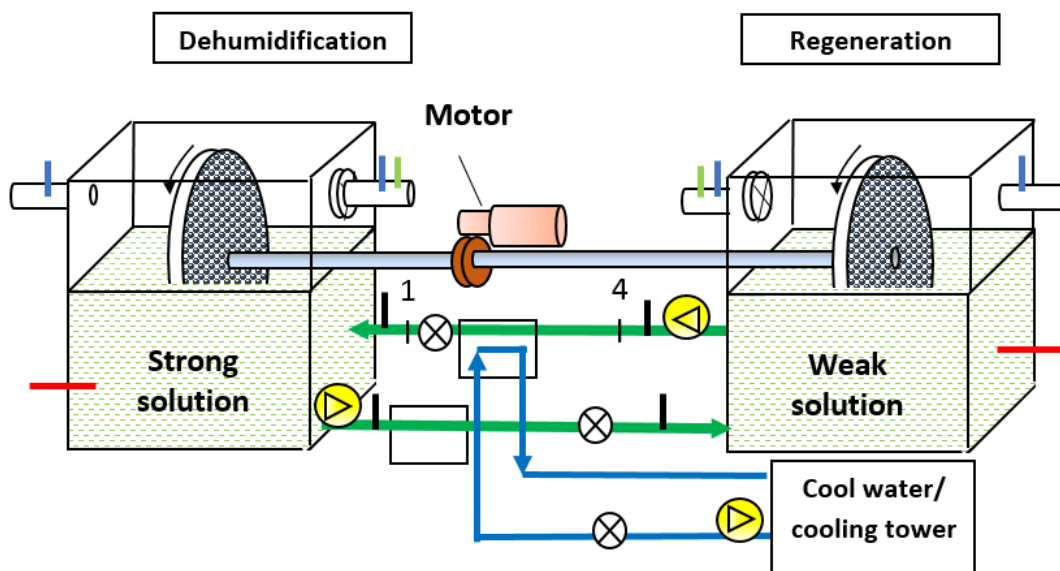


Figure 5.8 Liquid desiccant storage cooling process 4-1 (liquid)

Figure 5.8 shows process '4-1', which is required prior to the dehumidification process. In the heat exchanger, the solution needs to be cooled before entering the dehumidification section so that it will give the desired air cooling required by the system. The liquid solution interacts with the cold air that was represented by the blue line in the heat exchanger. During this time, the warm liquid from 4 transferred the heat to the cold water and left the heat exchanger and '1' as a cold liquid desiccant. A valve was used to control the flow rate of the solution into the dehumidification section. It will complete the cycle, and this process is continuous and ready to use all over again.

A process flow chart of the novel liquid desiccant wheel cooling system was illustrated in Figure 5.9. This chart can show the overall process of this experiment most straightforwardly and more efficiently. Firstly, the prototype start-off by starting the motor so that the wheels can rotate and the fan blows the outside air into the dehumidifier. The dehumidifier ran continuously until all the liquid's concentration become less than 30 wt.%. At the same time, the air came out from the dehumidifier was measured in terms of humidity and temperature. Then the magnetic pump and water heater were turned on so that the liquid can be heated in the heating loop. Next, the heated liquid was channelled to a regenerator. The liquid was brought into contact with recirculated air via the second wheel. In the regenerator, the liquid was concentrated again until it reaches around 30 wt.% of concentration. During this point, the air came out from regenerator as exhaust air and was measured for humidity and air temperature. Then another magnetic pump and water cooling loop was turned on to cool down the liquid. Finally, the liquid was channelled back to the dehumidifier, where we measured the

results again. The scrutiny process will be made if necessary on the system so that we can get the desired output which is dry and low-temperature air.

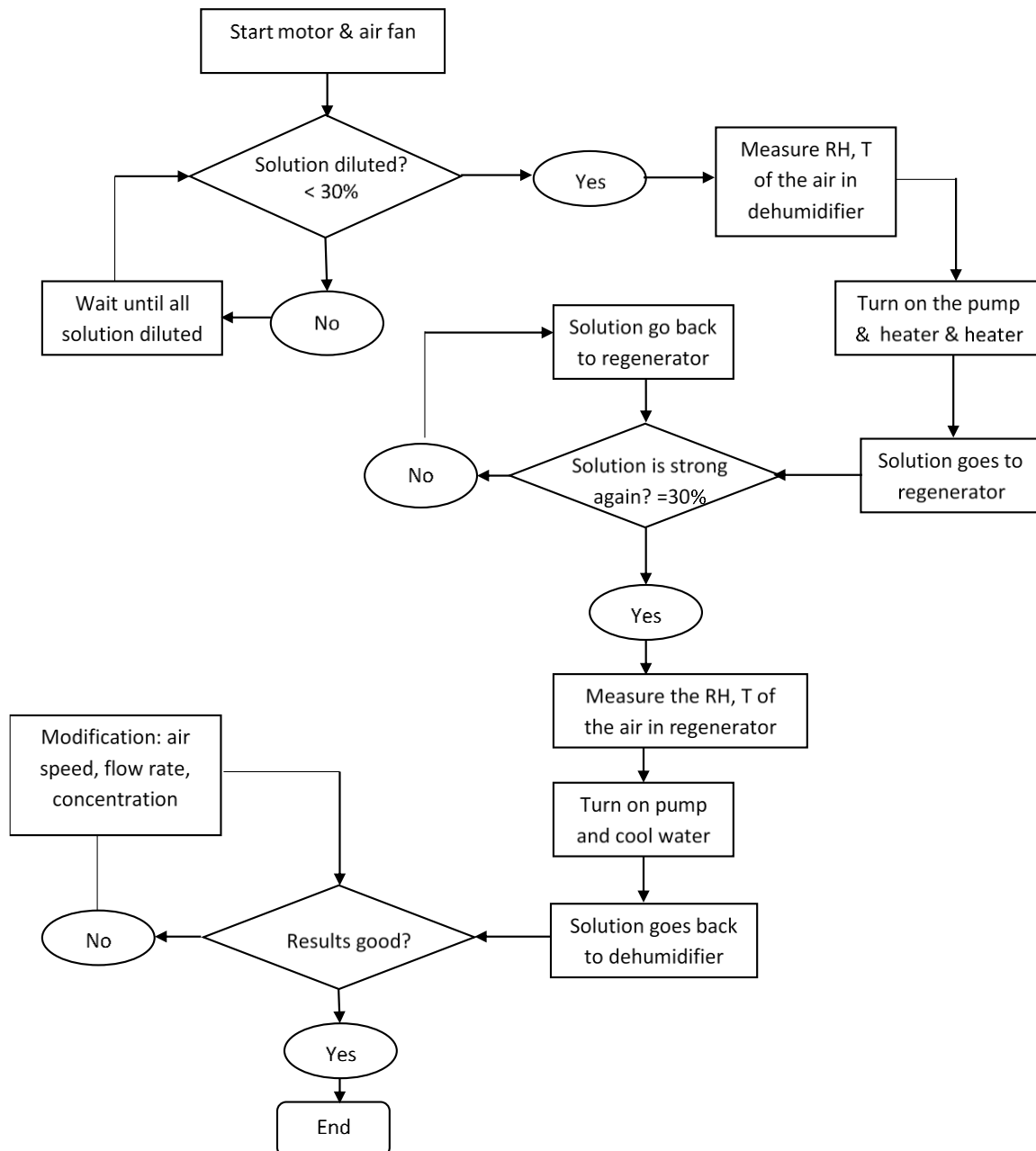


Figure 5.9 Process flow chart of novel desiccant wheel cooling system

5.4 Methodology

After all the fabrication and curtailed standards of procedures has been determined for novel desiccant wheel cooling system, the experiment methodology were commenced in Marmonth Laboratory. One of the facilities that were utilised in the laboratory is the Environmental Chamber. It is capable to provide a controlled ambient air for a specific duration of time. For this methodology, it will be categorised by two phase which is velocity variation and performance analysis. For the first phase, most of the time were spent on collecting data from the test rig that was connected with the environmental chamber. Most of the test were conducting the velocity variation inside the dehumidifier and regenerator. For the second phase, the data need to be analysed into a programme that was developed in EES. This method is very convenient where it help to determine the *COP* and the overall performance of the system.

5.4.1 Velocity Variation

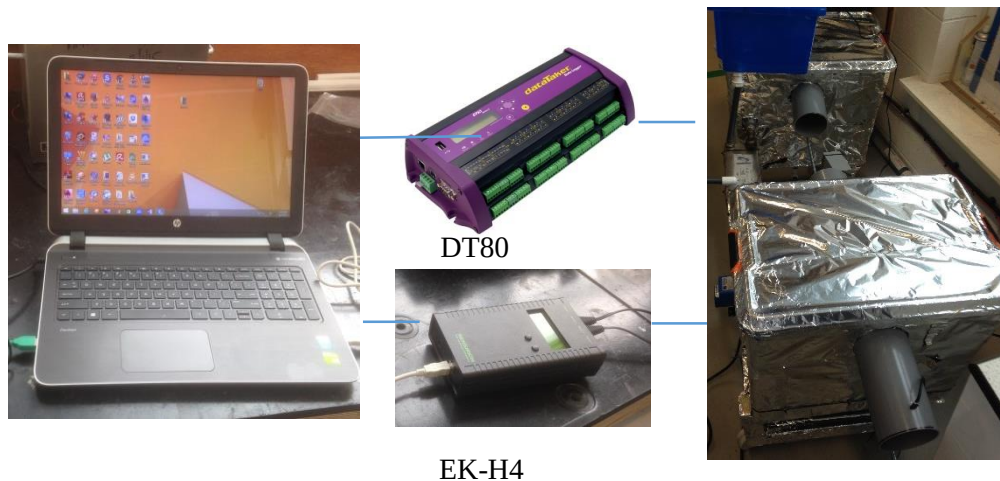


Figure 5.10 The test rig organisation of novel liquid desiccant wheel cooling system

The experimental setup for this experiment has been shown in Figure 5.10. A computer was connected with two data acquisition system, which is DT80 and EKH4. The former was connected with thermocouple K-type and concentration sensors while the latter was connected to SHT17 sensors. All the purposes and locations of all these sensors have been elaborated in section 5.2 and 5.3.

The test rig used ambient air as an air inlet and potassium formate solution (30 wt.%) as the working fluid. For this experiment, the air velocity variation via fan was mainly focused and studied. Five-velocity types were used, which are $v_1=2$ m/s, $v_2=4$ m/s, $v_3=6$ m/s, $v_4=8$ m/s and $v_5=10$ m/s. The environmental chamber in Marmonth Laboratory controlled the ambient air of this test. For the initial input, the temperature was set at 30 °C and 60% relative humidity. Each test took about 45 minutes for the each velocity. For this test, we fixed the rotor wheel speed at 1 RPM. The data such as temperature and relative humidity of the air will be stored into EKH4 data acquisition system. Then it was stored as a Microsoft Excel Comma Separated Value file. Meanwhile, temperature and concentration measurements for potassium formate were collected by DataTaker DT80 and were stored as a notepad file.

5.4.2 Programming Method to Determine COP and CC

In this stage, the data that were collected earlier will be used to determine the actual performance of the system. The data were key-in to the programme that was developed in EES so that it gave the desired output, which is in this case, *COP* of the novel liquid desiccant wheel cooling system. The theoretical background for the programme can be referred to in section 3.4.2. The programme for this purpose was shown in Figure 5.11.

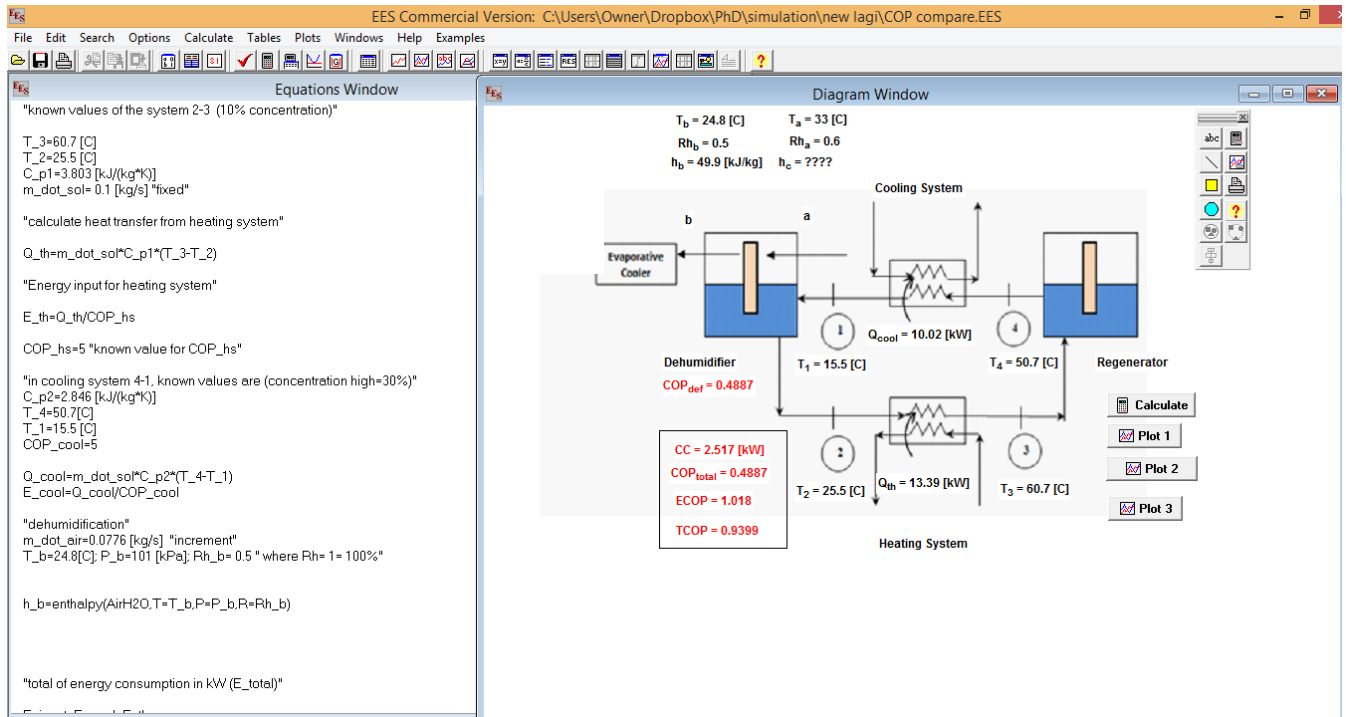


Figure 5.11 Determination of COP using EES

After the program displayed the *COP* output, these data were then gathered into each set of air velocity. Besides that, this programme also can show the cooling capacity (*CC*) of the system. After that, the graph of *COP* and *CC* can be plotted against the air velocity. This tool has been a great addition to this research since it helps a lot with calculations and saving a lot of time in determine *COP* and *CC*.

5.5 Hazard Analysis and Risk Assessment

During the operation of experimental work, it is compulsory to exercise the possibilities of hazard and risk that might occur. Some troubles were identified, and necessary modifications were implemented to prevent any unexpected events. Some modifications are described below:

- i) A drainage channel of the liquid must be provided in order to ensure there is a safe passage of liquid exit if anything goes wrong. The valve will control it, and the liquid will be channelled to a particular container that keeps the liquid from spilling onto the floor. The liquid can be kept in a safe place or sent to the disposal team to be discarded safely.
- ii) All outer components that possibly deal with heat will be insulated with thermal insulation to minimise heat loss or safety reasons. Among the parts that were protected are the dehumidifier tank, regenerator tank, liquid pipe, and air duct.
- iii) Also, the emergency shutdown button was installed on the rig where it can be easily reached to shut down the rig. In case of an emergency, the button is pressed to shut down the power supply.

5.6 Summary

This chapter presented an overview of the experimental facility and gave detailed information about the test rig design and assembly. It also contained detailed information about instrumentation used for this experiment and the data acquisition systems employed during the investigation. The chapter also detailed the research methodology, and the procedures followed in acquiring the experimental data. It also explained various sensor calibration details and further discussed the hazard analysis carried out in the experiment facility.

Chapter 6

Experimental Investigation of the Performance of Novel Desiccant Wheel Cooling System

6.1 Introduction

Following the design, construction, and development of desiccant wheel cooling system experimental rig explained in Chapter 5, a series of test was conducted to evaluate the performance of the system under a range of operating parameters. These parameters include relative humidity, air temperature, air velocity and many more. The desiccant wheel used in the experimental rig has been fabricated with polypropylene board immersed with potassium formate liquid solution. The results of this experimental test will be presented and discussed in this chapter. The summary of the conclusion reached for the system's performance under a variety of operating conditions will be provided at the end of this chapter.

System performance evaluation through theoretical modelling and simulation alone cannot replace a real experimental test. Thus, these experimental laboratory investigations remain a crucial part in assessing the overall performance of the desiccant cooling system. Such experimental study will then be used to validate the simulation results that were obtained earlier.

6.2 Brief Description of the System

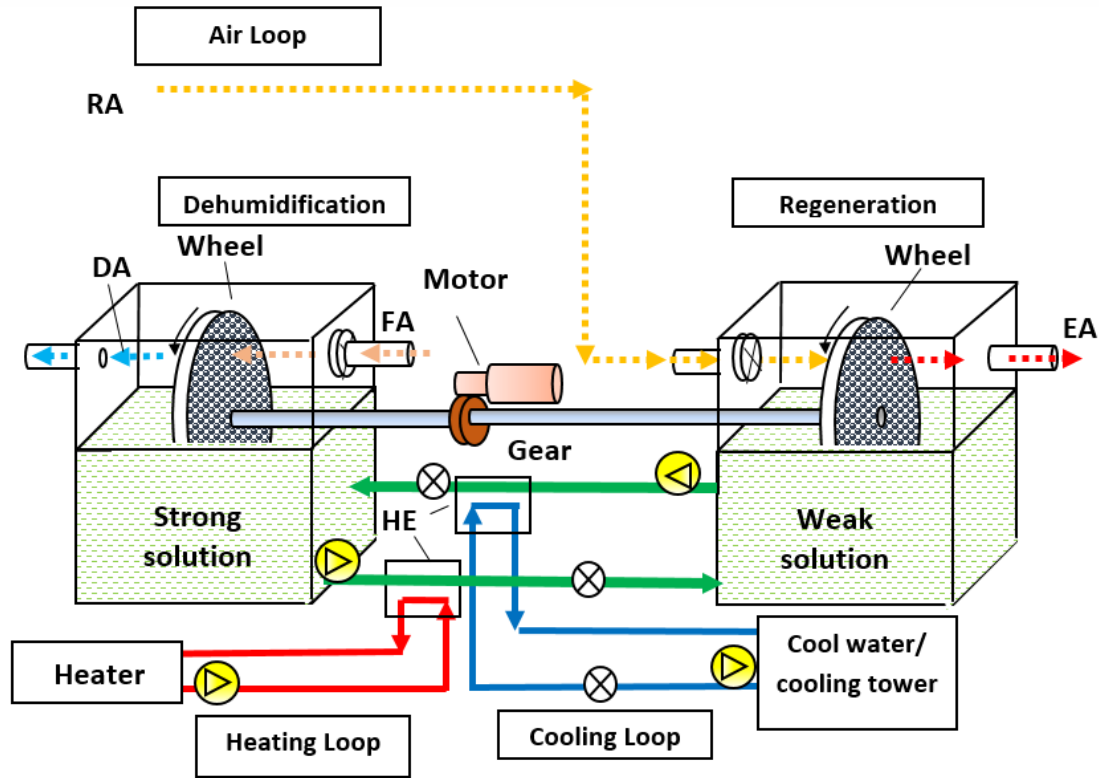


Figure 6.1 Schematic diagram of the system

Figure 6.1 shows the diagram of the novel liquid desiccant wheel cooling system that were investigated experimentally in this chapter. The environmental chamber supplied the fresh air (FA) to the dehumidification section. Then the potassium formate liquid around the wheel dehumidified the air. The liquid was able to be kept inside the wheel slots since it was submerged with that liquid earlier on. At this stage, the liquid was weak because it contains a lot of water. Then, the liquid will be channelled by the magnetic pump to a regenerator where it will be heated and humidified again. In the regenerator, the water content was removed from the liquid via to recirculated air (RA) so that the liquid desiccant will become strong again. Then, it was cooled and channelled back to the dehumidifier where the cycle can be completed, and the cycle will be continued until the desired results are obtained. All

temperature and humidity readings have been measured accordingly at designated locations in the test rig. The results of the preliminary and complete system will be discussed in the following section.

6.3 Experimental Procedure

The desiccant wheel cooling system was constructed and prepared for performance evaluation. Experiments for the performance evaluation was conducted in Marmonth Laboratory in School of Architecture & Built Environment, Faculty of Engineering, University of Nottingham. The test to assess the performance of the system under varying operating conditions were conducted in a quiet laboratory environment, and additional precautions were taken to minimise extensive heat exchange with the surrounding laboratory environment. The experimental test was conducted at ambient temperature and relative humidity conditions, and these conditions were later varied using heating and humidification source via climatic chamber.

Experimental tests were conducted under different conditions and settings to identify the best combinations of wheel speed rotation, air velocity, inlet temperature, and relative humidity that can provide the system to operate at its best-operating conditions. As described in Chapter 5, Section 5.2 of this thesis, the whole system was well-instrumented with temperature, relative humidity, air velocity, and concentration sensors. Experimental data acquisition was carried out using multichannel data taker DT80 and EKH4 data logger. Air temperature and humidity were monitored for both inlet, and outlets pipe of the process using SHT71 sensors and desiccant solution temperature were measured using K-type thermocouple.

The instantaneous values of the above mentioned experimental monitoring sensors were recorded at a given time interval by the data logger and saved into a computer. The experimental data was gathered at a second-time interval for all initial and final experiment observation for this test rig. All recorded data were uploaded to the computer from the data logger after every test for further analysis. All the data taken were presented in figures and been discussed in the following section.

6.4 Psychrometric Chart

The surrounding atmospheric air is a mixture of dry air, and water vapour often referred to as moist air. Psychrometric is a term used to describe the study of physical and thermodynamic properties of air-water vapour mixture.

A psychrometric chart is a graph which represents the thermodynamic parameters of moist air at a constant pressure. It graphically shows how these various air parameters relate to each other and is thus a graphical equation of state. Figure 6.2 presents the various thermophysical properties found on a psychrometric chart. The properties are dry-bulb temperature, wet-bulb temperature, dew point temperature, relative humidity, humidity ratio, specific enthalpy, and specific volume. Only two independent properties at constant pressure are needed to characterise air because the point of intersection of any property line determines the state point of air on a psychrometric chart. Other air properties can be obtained directly once the state point is located on the chart.

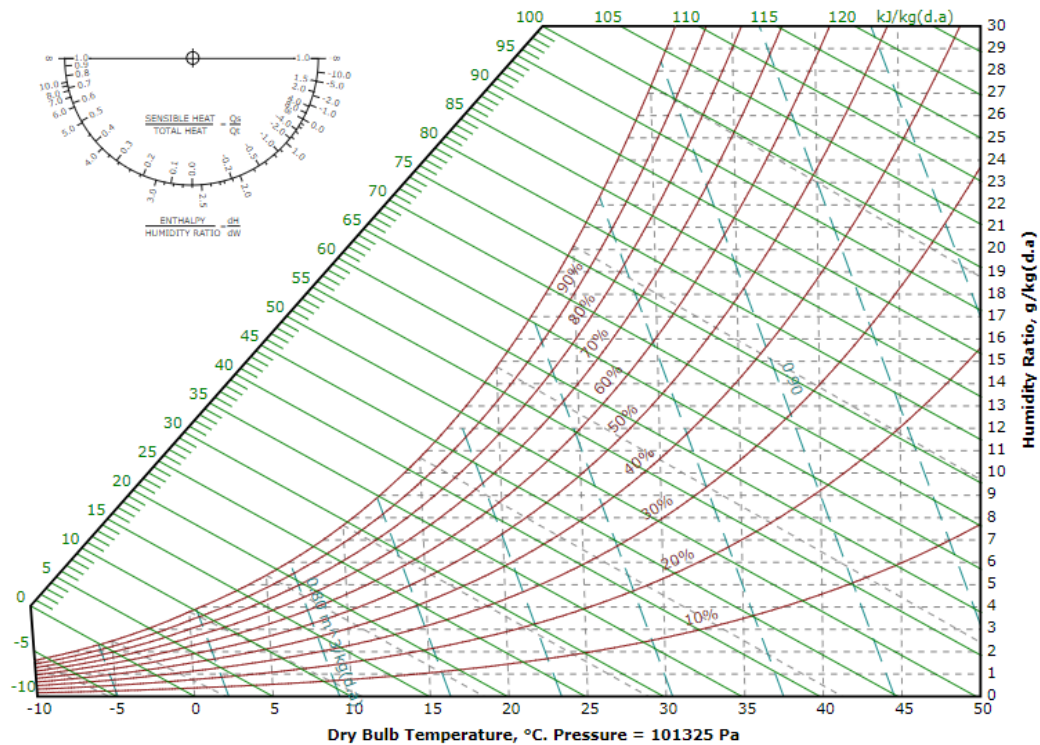


Figure 6.2 Psychrometric chart with variety of air properties (Web 6.1)

6.5 Factors Affecting the Desiccant Wheel Cooling System

The performance of the desiccant wheel cooling system is mainly affected by humidity level, temperature, air movement, and concentration of the liquid. These parameters usually interact with each other and as a result, influence the dehumidification, regeneration, and cooling rate of the cooling system. These parameters have been highlighted below:

6.5.1 Temperature

The regeneration is a process which requires heat and could come from a variety of sources. The rate of regeneration increase with high temperature and low pressure. It is because at a higher temperature, the molecules of water vapour have higher average speed, and more particles are able to break free from the surface of the liquid.

6.5.2 Humidity Level

The amount of water vapour present in the air is called air humidity. The humidity of air is measured in terms of absolute humidity, specific humidity, or relative humidity. A measurement of the actual moisture content in grams in a given cubic meter volume of air is referred to as absolute humidity. Specific humidity is expressed as the amount of water vapour in grams present in one kg of dry air, and it is sometimes referred to as humidity ratio. The most common measurement of air is relative humidity and is defined as the amount of water vapour in the air as a percentage of the maximum quantity of water vapour that the air capable of holding at a specific temperature. The use of psychrometric charts can estimate the various types of humidity described above. When the relative humidity of air reaches 100%, it is said to have reached saturation state and usually referred to as the dew point.

6.5.3 Surface Area

The humidification increases with an increase in the surface area of the desiccant wheel matrix. The larger the desiccant wheel, the more moisture will be allowed to be absorbed by the desiccant material over many cycles.

6.5.4 Air Pressure

Air pressure determines how fast water molecules will diffuse away from the water surface. Higher air pressure will slow down the diffusion process, while low air pressure will speed it up. Air pressure below the vapour pressure of water vapour at the same temperature will make the water to boil faster. As the atmospheric pressure compress the water molecules near the surface, more energy is required under high pressure to speed up the water molecules

to escape. For instance, water molecules need more energy at sea level in order to evaporate than at higher attitudes under the same relative humidity and temperature.

6.5.5 Air Flow

The movement of air which could come naturally or forcedly (using an electric fan) is an essential factor affecting both dehumidification and regeneration process of a desiccant wheel cooling system. Humid air outside the system continuously flows through the process air section of the desiccant wheel where moisture is adsorbed and the warm air from air-conditioned space flow to regeneration section and to the desiccant wheel, which helping regeneration. Warm air flow in the regeneration air section supply drier air to the desiccant wheel surface and remove moisture around the matrix of the wheel.

6.6 Factors Considered in Analysis

Experimental performance evaluation test of the desiccant wheel cooling system was conducted to analyse its performance under different operating conditions that are important in evaluating the performance of the dehumidification and refrigeration processes. An initial test was conducted to demonstrate the desiccant solution's effect across the wheel, and thermocouple sensors were used to measure temperature change on the surface of the liquid solution in regenerator and dehumidifier. A further test was later conducted on the prototype under varying operating conditions to evaluate overall performance. The varied operating conditions include the effect of a varying inlet air velocity into the system.

During all the tests were carried out under the above mentioned various operating conditions, test results at a different value of inlet temperature, humidity, and corresponding outlet

conditions were recorded. Values of airflow velocity at both dehumidification and regeneration processes were also recorded under steady-state conditions, where all measured values displayed negligible change over a long period.

6.7 Complete Wheel Liquid Desiccant Cooling Test

After a complete simulation of the performance of the wheel matrix, as shown in Chapter 4 earlier, the testing of a full scale of the novel liquid desiccant wheel cooling system was carried out in this section. The overall system's experimental setup has been discussed extensively in Chapter 5.

6.7.1 Air Velocity Variation versus Time

In this experiment we used five different velocity settings, which are $v_1 = 2$ m/s, $v_2 = 4$ m/s, $v_3 = 6$ m/s, $v_4 = 8$ m/s, and $v_5 = 10$ m/s. We tested the prototype for about two months, and then we stored the data in the data acquisition system and analysed using the computer. As the velocity of air is varied, we fixed other variables such as wheel rotation at 1 RPM. Then another parameter input was fixed, which is relative humidity inlet (RH_{in}) at 60%, and air temperature inlet (T_{in}) 33°C in the dehumidifier. Using the psychrometric chart generated in EES, the humidity ratio taken was 0.01918 kg_{water}/kg_{dry air}

For the experiment, we varied the air velocity at both dehumidifier and regenerator for about 45 minutes for each test. As shown in Figure 6.3, for air velocity $v_1 = 2$ m/s the humidity ratio dropped exponentially from 0.01918 kg_{water}/kg_{dry air} to 0.009821 kg_{water}/kg_{dry air} in the dehumidifier at the end of the experiment. As we can see, the humidity ratio was dropping very

steeply for about three minutes before it steadied itself afterwards. Then at $v_2 = 4$ m/s, the humidity ratio fell from initial which is $0.01918 \text{ kg}_{\text{water}}/\text{kg}_{\text{dry air}}$ to $0.007795 \text{ kg}_{\text{water}}/\text{kg}_{\text{dry air}}$ in the dehumidifier. After the air velocity was set at $v_3 = 6$ m/s, it was observed that the humidity ratio in dehumidifier dropped significantly from initial to $0.006027 \text{ kg}_{\text{water}}/\text{kg}_{\text{dry air}}$. Next, the humidity ratio in the dehumidifier dropped drastically $0.004108 \text{ kg}_{\text{water}}/\text{kg}_{\text{dry air}}$ when we set the air velocity $v_4 = 8$ m/s. For the last air velocity, which is $v_5 = 10$ m/s, it can be observed that the humidity ratio in the dehumidifier fell rapidly to $0.002511 \text{ kg}_{\text{water}}/\text{kg}_{\text{dry air}}$ only. As we can see from this pattern, the higher air velocity that we put in, the more profound drop of humidity ratio can be achieved.

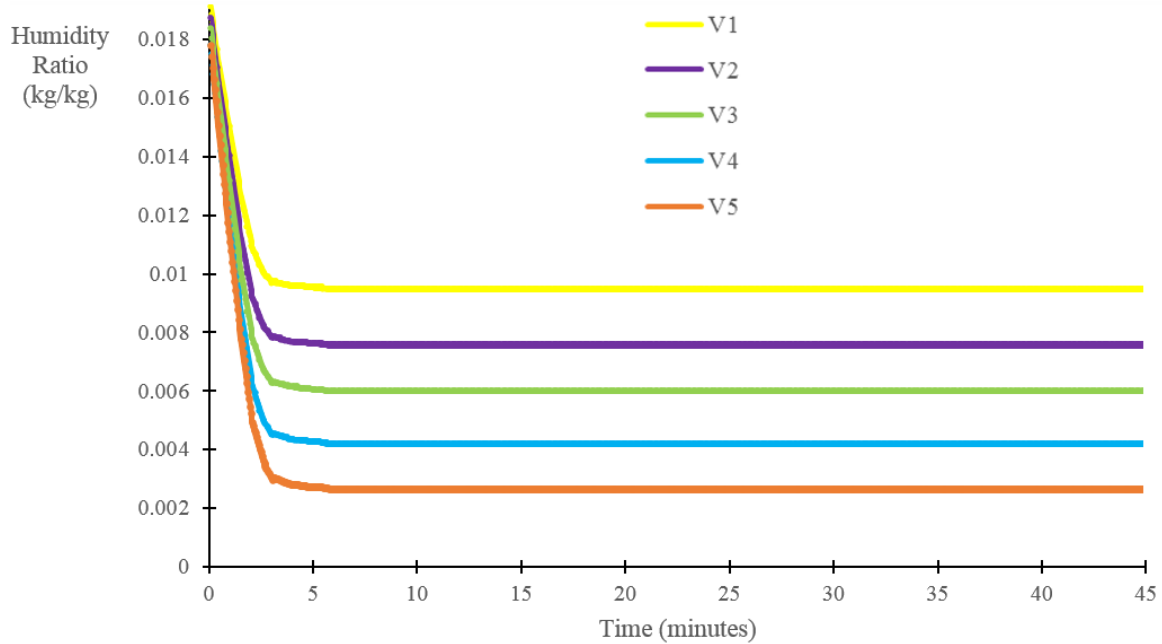


Figure 6.3 Humidity ratio variation according to different air velocity versus time in dehumidifier.

Figure 6.4 shows the trend for air inlet temperature in the dehumidifier. From this figure, at v_1 the air temperature was dropping from 33°C to 24.8°C also after 3 minutes, and then it became

steady at around 24.8 °C. Then at v_2 , we observed that air temperature was dropping from 33°C to 21°C.

Next at v_3 , the air temperature also fell significantly from initial to 18.1 °C. After we increased the air velocity to v_4 , the temperature was also reduced to 16.2 °C. Finally, at highest air velocity v_5 , the air temperature dropped and stable around 15 °C. From these results in the dehumidifier, it was observed that the higher air velocity at the inlet, the lower air temperature would be at the outlet.

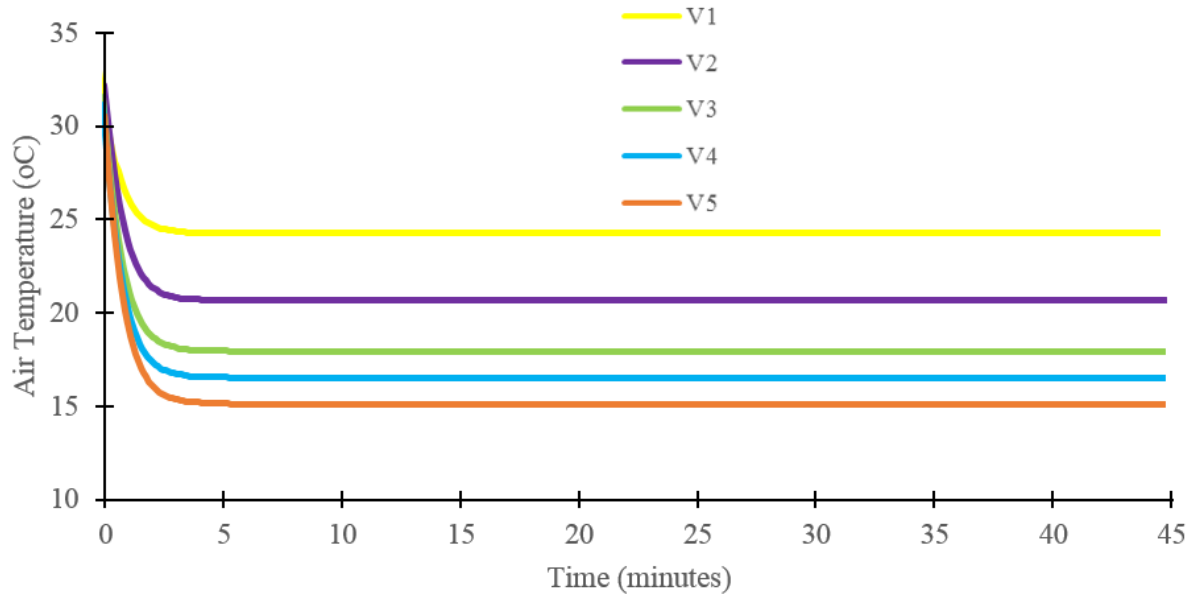
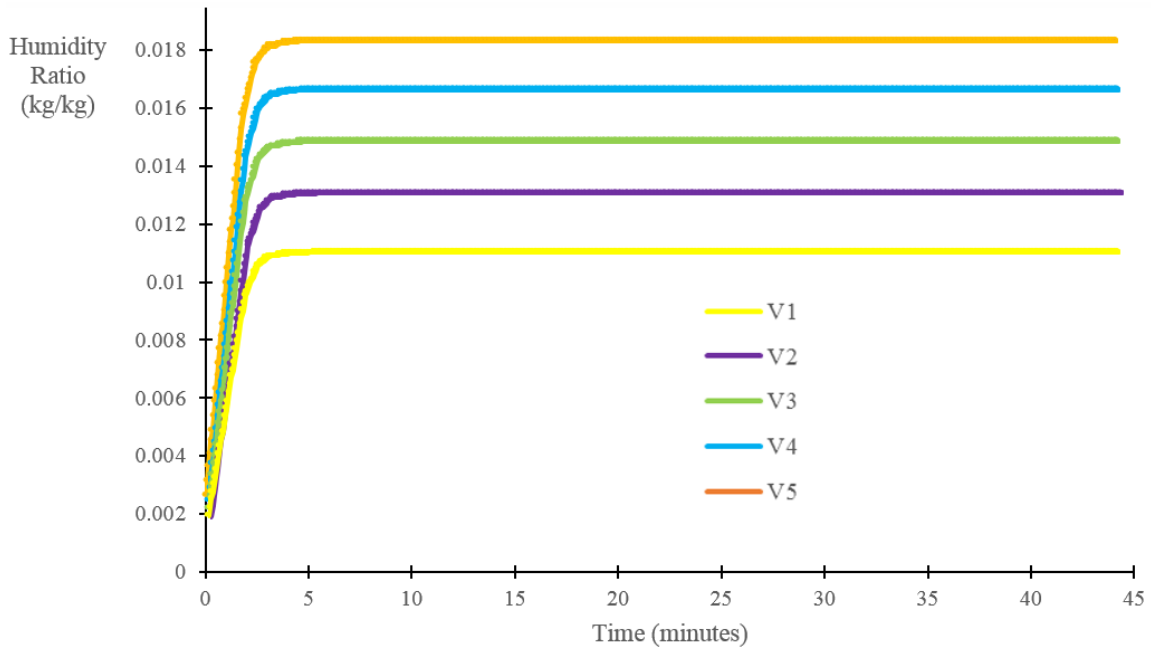


Figure 6.4 Air temperature variation according to different air velocity versus time in dehumidifier.

After that, we looked for the same parameters, but this time we tested it out at the regenerator. As shown in Figure 6.5, the humidity ratio of around 0.002138 kg_{water}/kg_{dry air} was from the conditioned space and used as humidity input into the regenerator. In the regenerator, using the initial air velocity v_1 , the humidity ratio raised from 0.002138 kg_{water}/kg_{dry air} air to 0.01256 in three minutes before it had reached a plateau. After we increased the air velocity to v_2 , the

humidity ratio increased from $0.002138 \text{ kg}_{\text{water}}/\text{kg}_{\text{dry air}}$ to $0.01253 \text{ kg}_{\text{water}}/\text{kg}_{\text{dry air}}$. Next, the humidity ratio increased again from $0.002138 \text{ kg}_{\text{water}}/\text{kg}_{\text{dry air}}$ to $0.01432 \text{ kg}_{\text{water}}/\text{kg}_{\text{dry air}}$ when we set the air velocity to v_3 . We then increased the air velocity to v_4 and found that the humidity ratio also increased from initial to $0.01624 \text{ kg}_{\text{water}}/\text{kg}_{\text{dry air}}$. For the highest air velocity, which is v_5 , the humidity ratio is also the highest, which is $0.01813 \text{ kg}_{\text{water}}/\text{kg}_{\text{dry air}}$. Based on these results in the regenerator, the higher the air velocity, the higher the humidity ratio increased when leaving the regeneration section.



(b)

Figure 6.5 Humidity ratio variation according to different air velocity versus time in regenerator.

After that, we are going to look at air temperature behaviour in the regenerator, as shown in Figure 6.6. At first air velocity v_1 , the initial air temperature at 15.2°C , is increased within three minutes before it finally steadied at around 22.5°C . Then at v_2 , the air temperature also

increased from 15.2 °C to 24.1 °C. Next, we increased the air velocity to v_3 where the air temperature increased from initial temperature to 26.2 °C. Afterward, we set the air velocity to v_4 the air temperature also raised from 15.2 °C to 28.3 °C. For the final air velocity v_5 , the air temperature at this point is the highest where is stable at around 30.5 °C. It can be seen from the graphs that the air temperature output increased as a result of more air velocity flowing into the regeneration section.

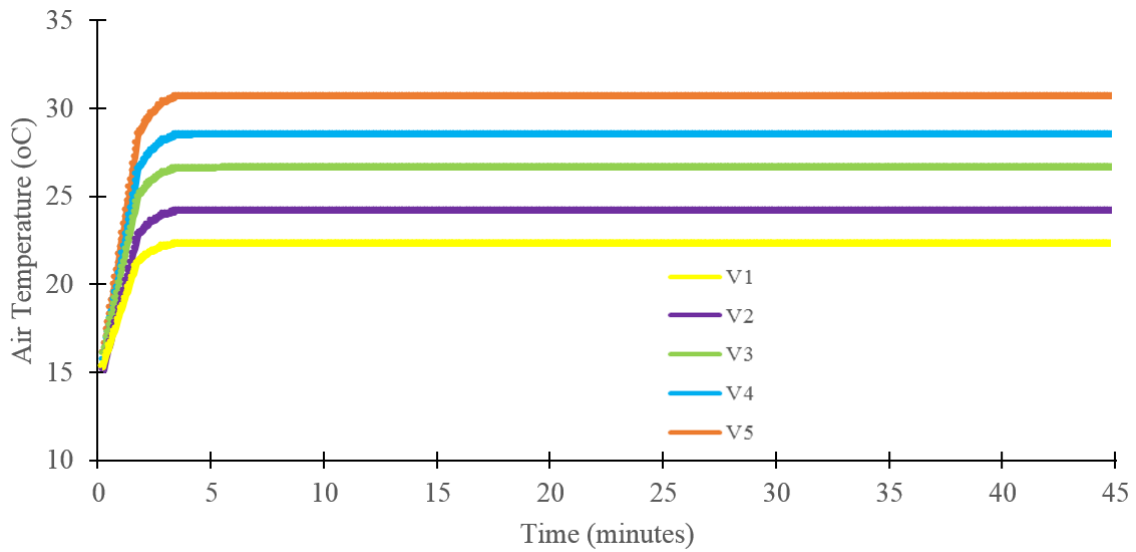


Figure 6.6 Air temperature variation according to different air velocity versus time in regenerator.

After observing the air's humidity ratio and temperature, we observed the liquid desiccant, potassium formate. Here, we measured the liquid desiccant's temperature at the inlet and outlet pipes of the dehumidifier. As shown in Figure 6.7, the liquid desiccant temperature was about 14.5 °C when it entered the dehumidifier. When it left the dehumidifier at v_1 , the temperature had risen to 25.5 °C. Then at v_2 , the temperature of the liquid increased to 27.8 °C. After the air velocity increased to v_3 , the temperature of the liquid also increased to 30.2 °C. Next, the liquid

temperature rises to 33.6 °C when the air velocity was set to v_4 . At the final air velocity v_5 , it stated the highest liquid temperature that is 36.1 °C. Based on these results, the higher the air velocity, the greater the liquid temperature in the dehumidifier.

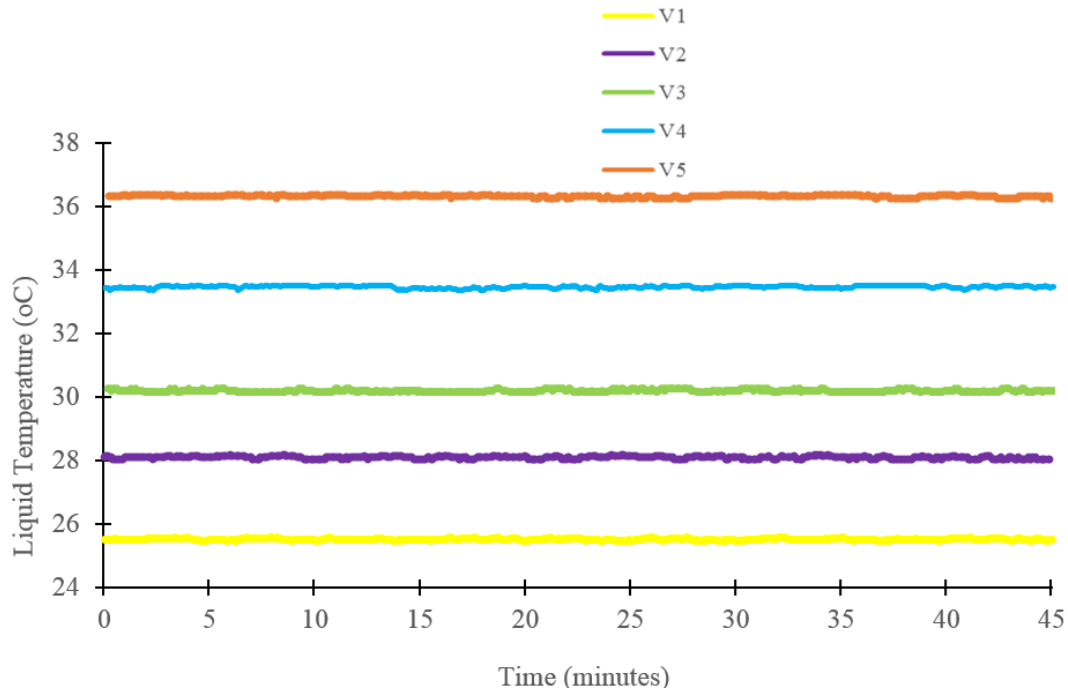


Figure 6.7 Liquid temperature at the outlet of dehumidifier section versus time.

Besides dehumidifier, we also focused on the temperature distribution in liquid desiccant that occurred in the regenerator, as shown in Figure 6.8. Initially, the temperature of liquid desiccant was 60.7 °C before it entered the regenerator. When the liquid desiccant left the regenerator, the temperature was about 50.7 °C when we used air velocity v_1 . After that, we set the air velocity to v_2 and found the liquid temperature reduced at 46.8 °C. When we test it at air velocity v_3 , the liquid temperature was also reduced to 42 °C. Afterward, the liquid temperature was reduced again to 37.2 °C when we use air velocity v_4 . Finally, when air velocity was set at v_5 , the liquid temperature leaving the regenerator recorded around 32.8 °C. As shown by the

figure, the potassium formate liquid temperature was significantly reduced when we increased the air velocity into the regenerator.

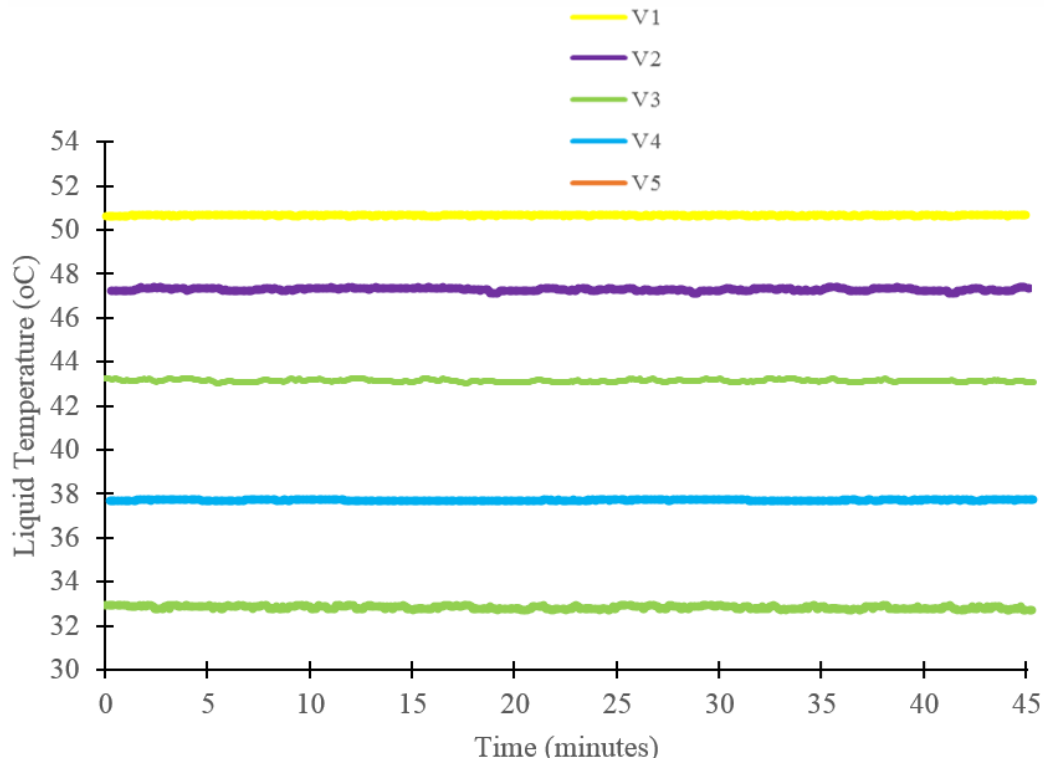


Figure 6.8 Liquid temperature at the outlet regenerator section versus time.

Another important aspect of this analysis is the concentration of liquid desiccant in the dehumidifier, as shown in Figure 6.9. In this figure, when we set the air velocity to v_1 , the potassium formate concentration drops from 30 wt.% to 21.1 wt.% at the end of the experiment. Then when the air temperature was set at v_2 , the concentration of the liquid also subsequently drops from 30 wt.% to 19.2 wt.%. Afterward, the liquid concentration was reduced further to 15.5 wt.% when air velocity is at v_3 . Next, we step up the air velocity to v_4 ; the potassium formate exhibit a decrease in its concentration, which lies around 13 wt.%. Lastly, for air velocity at v_5 , the liquid concentration fell drastically from 30 wt.% to 9.9 wt.%. As for the

findings of the liquid's concentration, the higher the air velocity will be in significantly reducing liquid concentration.

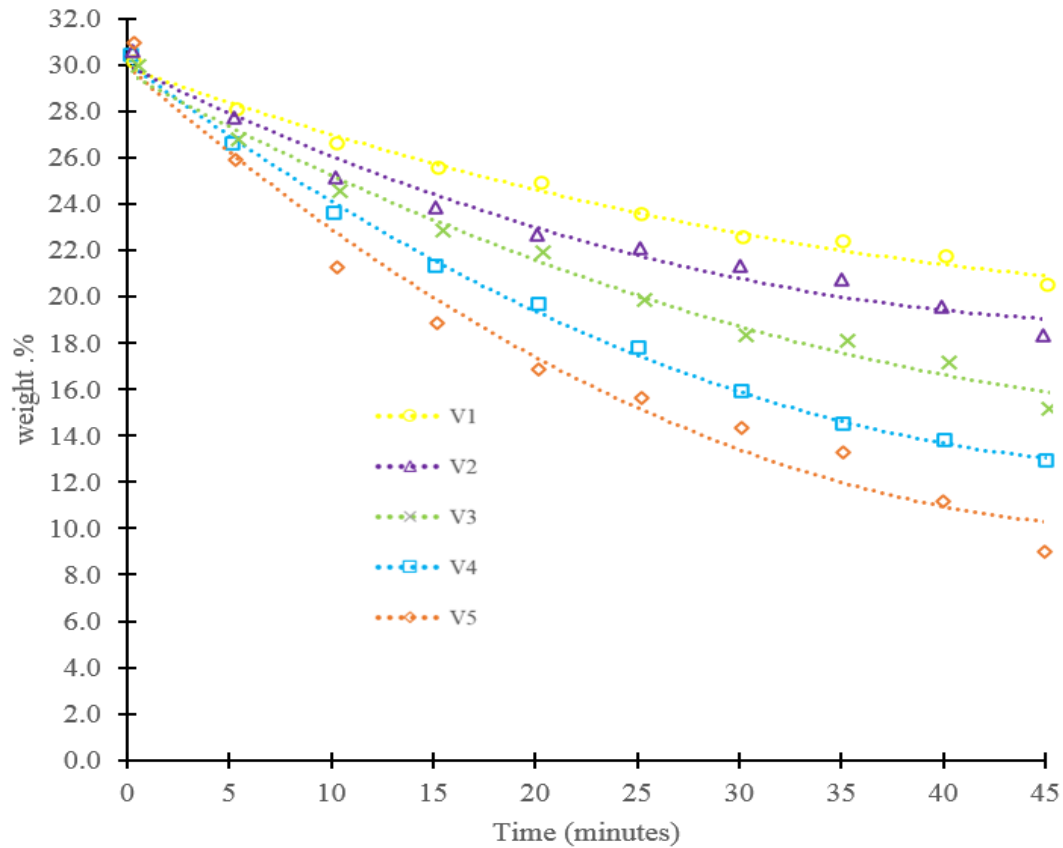


Figure 6.9 Concentration of liquid versus time at various air velocity

6.7.2 Analysis and Discussion of Air Velocity Variation in the System

Figures in Section 6.7.1 show that increasing the velocity will enhance the novel liquid desiccant wheel cooling system. In the experiment, we varied the velocity into five velocities, which are $v_1=2$ m/s, $v_2=4$ m/s, $v_3=6$ m/s, $v_4=8$ m/s, and $v_5=10$ m/s. From here, we can evaluate the performance of the system in terms of humidity ratio, air temperature, liquid desiccant temperature, and desiccant concentration in the dehumidifier and regenerator, respectively. Those results were tabulated in tables shown in this section.

Table 6.1 Performance evaluation of average air parameter output according to air velocity variation in dehumidifier

Velocity	Dehumidifier	
	HR_{out} (kg _{water} /kg _{dry air})	$T_{air, out}$ (°C)
v_1	0.009821	24.8
v_2	0.007795	21.0
v_3	0.006027	18.1
v_4	0.004108	16.2
v_5	0.002511	15.0

The performance evaluation of the dehumidifier can be further elaborated, as shown in Table 6.1. Initially, the humidity ratio and air temperature inputs were 0.01918 kg_{water}/kg_{dry air} and 33 °C, respectively. From a humidity ratio point of view, the wheel cooling system reduced the humidity ratio with increasing air velocity. We could also see the difference in terms of air temperature where the air temperature was subsequently reduced when the air passed through the wheel matrix. From these data, we could say that the air's relative humidity can be significantly reduced if more air is put into the wheel matrix. The more air coming into contact with the liquid desiccant layer, the more moisture content could be removed from the air. It also increased the water content that was absorbed by potassium formate, becoming a weak desiccant solution.

Meanwhile, air temperature output decreased along with the increasing amount of air velocity. It was because of the cooling sensation that was brought by the liquid desiccant from the liquid pipe inlet to the dehumidifier. When it was brought into contact with air by the wheel matrix, the heat exchange also occurred where heat from the air portion was absorbed or transferred to the liquid layer, which lowered the air temperature. For instance, at $v_1=2$ m/s, the air inlet temperature was around 33 °C, and the liquid desiccant inlet was 15°C. When both fluids left

the dehumidifier, the air temperature dropped to 24.8°C, and the temperature of liquid desiccant increased to 25.5 °C, respectively. It happened in controlled volume cases, and both included fluid, which are liquid and gas. Convection happens when particles with a lot of heat energy in the liquid move and replace particles with less energy in the gas state. Heat is transferred from a hot place to a cooler place by convection between liquid and gas. As the liquid in hot areas is less dense than the gas in the cold area, it travels to cold areas. The denser cold gas will fall into the warm areas that are available in the liquid.

Table 6.2 Performance evaluation of average air parameter output according to air velocity variation in regenerator

Velocity	Regenerator	
	HR_{out} (kg _{water} /kg _{dry air})	$T_{air, out}$ (°C)
v_1	0.01256	22.5
v_2	0.01253	24.1
v_3	0.01432	26.2
v_4	0.01624	28.3
v_5	0.01813	30.5

The regenerator displayed the same concepts and principles as the dehumidifier, except that it rejected heat and moisture through the air exhaust outlet. As we can see from Table 6.2, the output of air relative humidity is around 0.012 kg_{water}/kg_{dry air} to 0.18 kg_{water}/kg_{dry air} air. Nevertheless, we need to consider the air relative humidity input to the regenerator, which is the air that was circulated in the room. Initially, the humidity ratio and air temperature inputs were 0.002138 kg_{water}/kg_{dry air} air and 15.2 °C, respectively. This circulated air is used to remove moisture and heat from the regenerator, as it is mostly still dry and cold. As we can see from the table, the air input was lower in terms of humidity ratio but can produce a higher humidity ratio at the regenerator's outlet. This higher relative humidity was intended to be rejected to

outside of a building or to the surrounding. As we can see, the humidity ratio was getting higher along with increasing air velocity. It means that the higher the air velocity, the higher the moisture content can be removed from the desiccant solution to moving dry air in the wheel matrix. It will help the desiccant solution to become strong again and be channelled back to the dehumidifier. The same goes for air temperature, where input air temperature decreases as air velocity increases. If more air was pumped to the wheel matrix, heat from the desiccant solution can be transferred to moving air, which would make the desiccant solution much cooler.

Table 6.3 Performance evaluation of liquid desiccant output according to air velocity variation in dehumidifier

Velocity	Dehumidifier
	T_{out}
v_1	25.5
v_2	27.8
v_3	30.2
v_4	33.6
v_5	36.1

Besides air, we also looked into the parametric behaviour of liquid desiccant in the dehumidifier. As we can see in Table 6.3, the temperature of desiccant liquid was fixed at about 14.5 °C for each test when it left the cooling section. We noticed that when liquid desiccant leaves the dehumidifier, the temperature of the liquid desiccant increased considerably. The output temperature of the liquid is also increased when we increased the air velocity for each test. The more air we pumped into the wheel matrix, the more heat can be gained to the wheel matrix's liquid desiccant layer. Heated air, which is denser, will transit to the colder liquid, which is less dense. This heat exchange also helps the air become much cooler when supplied

out of the dehumidifier. Simultaneously, the liquid desiccant also becomes weaker because of the high moisture content being absorbed from the air in the dehumidifier.

Table 6.4 Performance evaluation of liquid desiccant output according to air velocity variation in regenerator

Velocity	Regenerator
	T_{out}
v_1	50.7
v_2	46.8
v_3	42.0
v_4	37.2
v_5	32.8

We also observed the regenerator's performance in terms of the heat exchange capability of the liquid desiccant. As we can see in Table 6.4, we managed to produce a lower temperature of liquid desiccant after executing mass and heat transfer with circulated air in the wheel matrix. After each test, on average, all desiccant solution showed a reduction of output temperature compared to the input temperature. It also showed that the temperature reduction became more extensive for the desiccant solution when the air velocity was raised. It shows that the liquid desiccant with a considerable amount of energy in particles will move to the spaces occupied by air particles with less energy. The hot area will leave the desiccant solution to the colder area in moving air, leaving the desiccant solution much cooler, and this will reduce the energy needed for further cooling in the cooling section. This solution is then ready to be reused for the dehumidifier to ensure that the cycle is continuous.

Table 6.5 liquid desiccant concentration according to air velocity variation in dehumidifier

Velocity	Concentration (wt.%)
v_1	21.1
v_2	19.2
v_3	15.5
v_4	13.0
v_5	8.9

As we can see in Table 6.5, the liquid desiccant concentration also reduced along with increasing air velocity. The more considerable amount of air pumped into the wheel matrix, the larger the surface area, and the bigger the moisture content that can be taken out and transferred to the liquid desiccant layer. This high-moisture liquid desiccant can be restored to a stronger concentration when channelled into the regenerator. It is where the liquid was heated, and moisture was being removed via outside air. It was seen that the liquid could operate at numerous times without any significant reduction of its volume and also quality. It is imperative as it is the primary working fluid that will continuously work in the long run.

6.7.3 Results and Discussion of *CC* and *COP*

This section is the most critical part of this project since it will measure the novel liquid desiccant wheel cooling system's overall performance. Using the input parameters that we set earlier, we calculate and analyse the Coefficient of Performance (COP_{Total}) and measure it according to increased velocity and time. However, it is also interesting to see the value of Cooling Capacity (*CC*) and how it performs in this system. The analysis of *CC* and *COP* according to a different set of velocity was shown in this subsequent analysis.

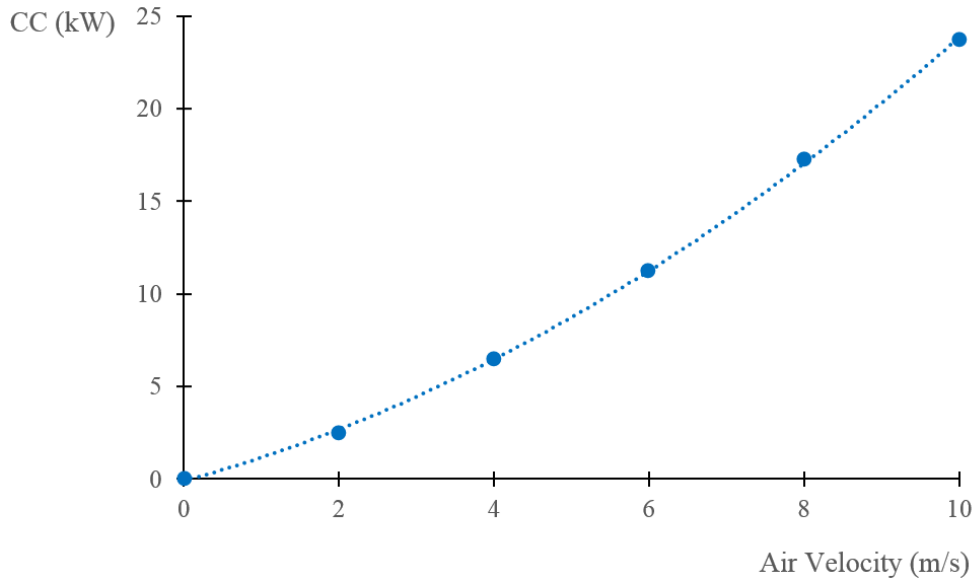


Figure 6.10 CC against different settings of air velocity

As we can see in Figure 6.10, the CC value increased with larger air velocity in the system. First, at v_1 , the CC value is 2.517 kW. Then, we increase the air velocity to v_2 , and the CC value becomes 6.433 kW. Next, the CC value subsequently increased to 11.24 kW when we raised the air velocity to v_3 . After that, we increased the air velocity again to v_4 so that the CC value become 17.24 kW. Finally, at v_5 , the maximum the CC value obtained from this experiment is 23.71 kW. The CC value is important as we can determine the value needed to cool a particular room in a building and determine our desired value of COP_{Total} .

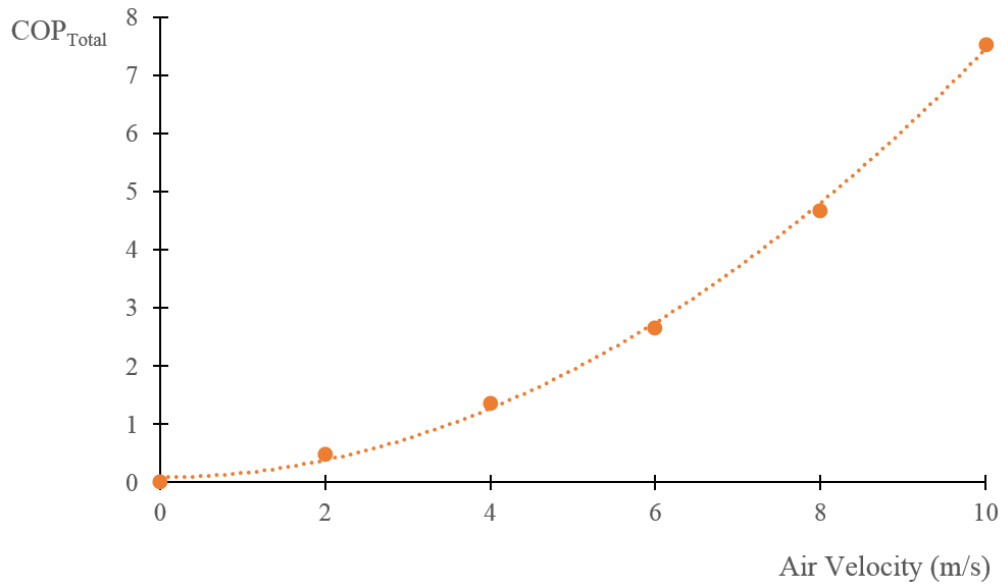


Figure 6.11 COP against different settings of air velocity

In Figure 6.11, we can see that the value of COP_{Total} is increasing with the additional value of air velocity. Firstly, we can see that when air velocity $v_1 = 2$ m/s, the COP_{Total} value will be 0.489. Then we increase the air velocity to $v_2 = 4$ m/s; the COP_{Total} would become 1.365. Next, the COP_{Total} increased again to 2.669 when we raised the air velocity $v_3 = 6$ m/s. Afterwards, when the air velocity was set to $v_4 = 8$ m/s, the COP_{Total} value become 4.676. Lastly, when the maximum air velocity $v_5 = 10$ m/s was utilised, the COP_{Total} value equals to 7.547. This result is significant because this will tell the novel liquid desiccant wheel cooling system's whole performance. The higher value of COP_{Total} , the better performance as it can produce high energy output using a low energy input.

6.8 Simulation Validation Using Experimental Data

To validate the simulation used in EES is suitable for the novel liquid desiccant wheel cooling system, we took the results from the simulation and compared it with the results from the real experiment. We must validate the simulation results as we can use it to predict the experimental results of the novel liquid desiccant wheel cooling system. In order to validate this, the humidity ratio against time was chosen as the comparison study for the system. We compare the result by varying the air velocity parameters which is $v_1 = 0.5$ m/s, $v_2 = 1.5$ m/s, $v_3 = 2.5$ m/s, $v_4 = 3.5$ m/s and $v_5 = 4.5$ m/s.

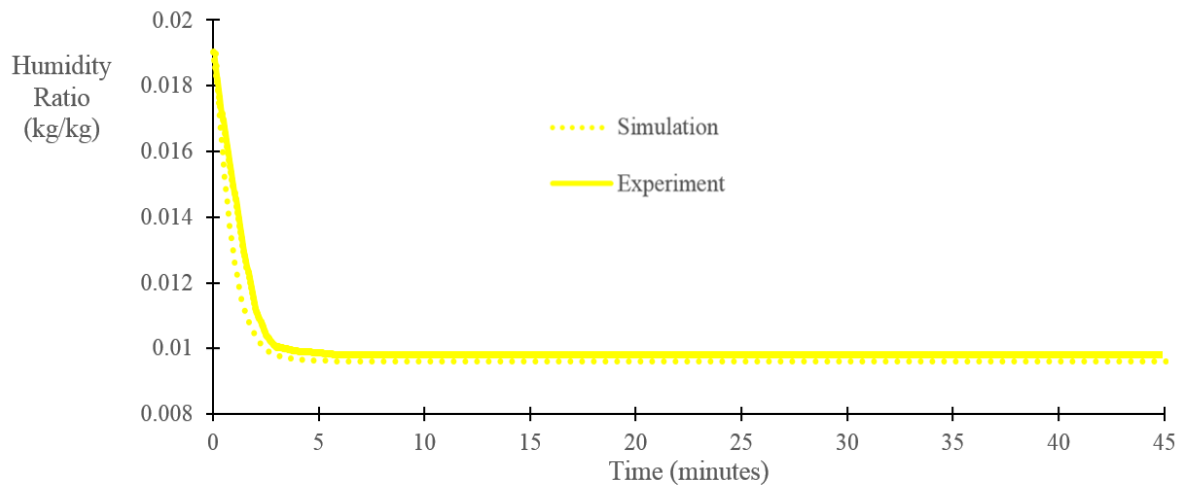


Figure 6.12 Comparison of simulation and experimental results of humidity ratio versus time at velocity $v_1 = 2$ m/s

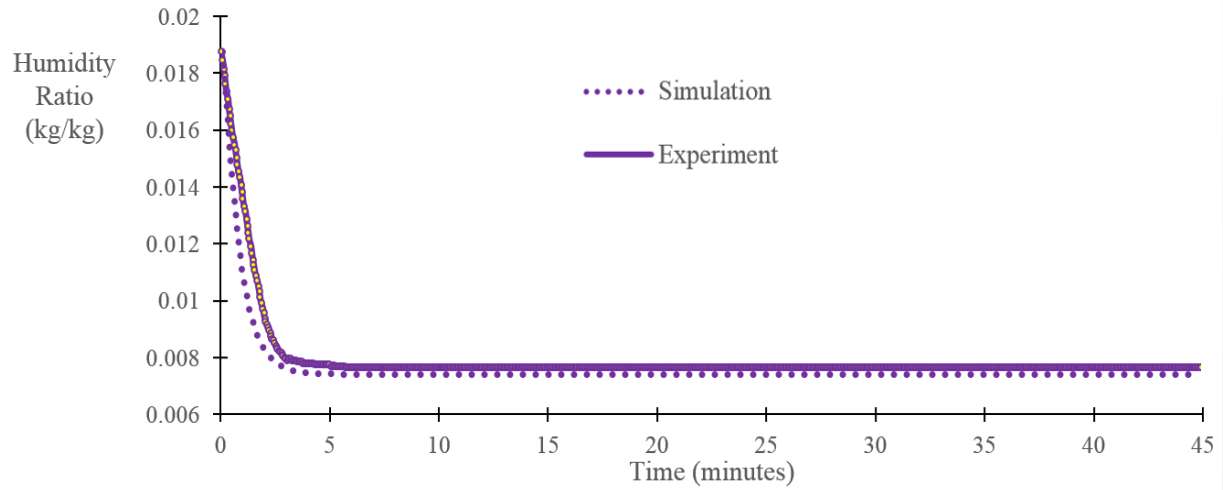


Figure 6.13 Comparison of simulation and experimental results of humidity ratio versus time at velocity $v_2=4\text{m/s}$

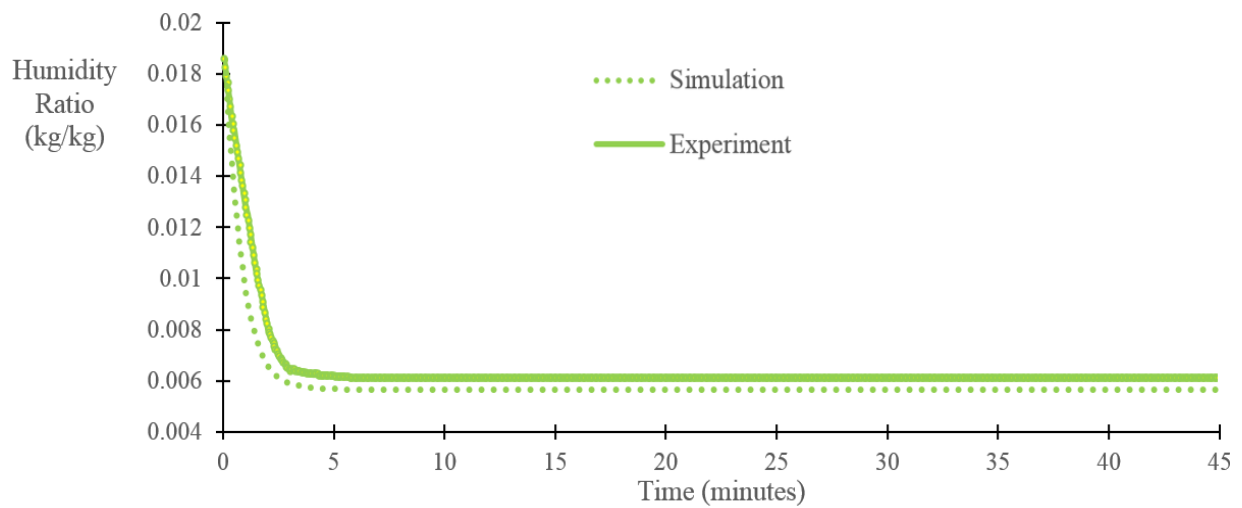


Figure 6.14 Comparison of simulation and experimental results of humidity ratio versus time at velocity $v_3=6\text{ m/s}$

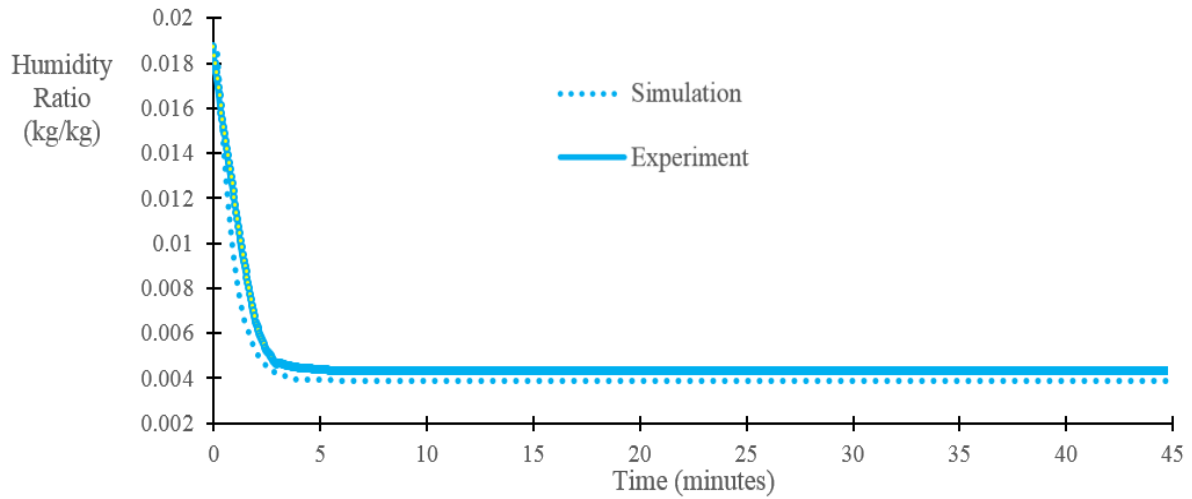


Figure 6.15 Comparison of simulation and experimental results of humidity ratio versus time at velocity $v_5=8$ m/s

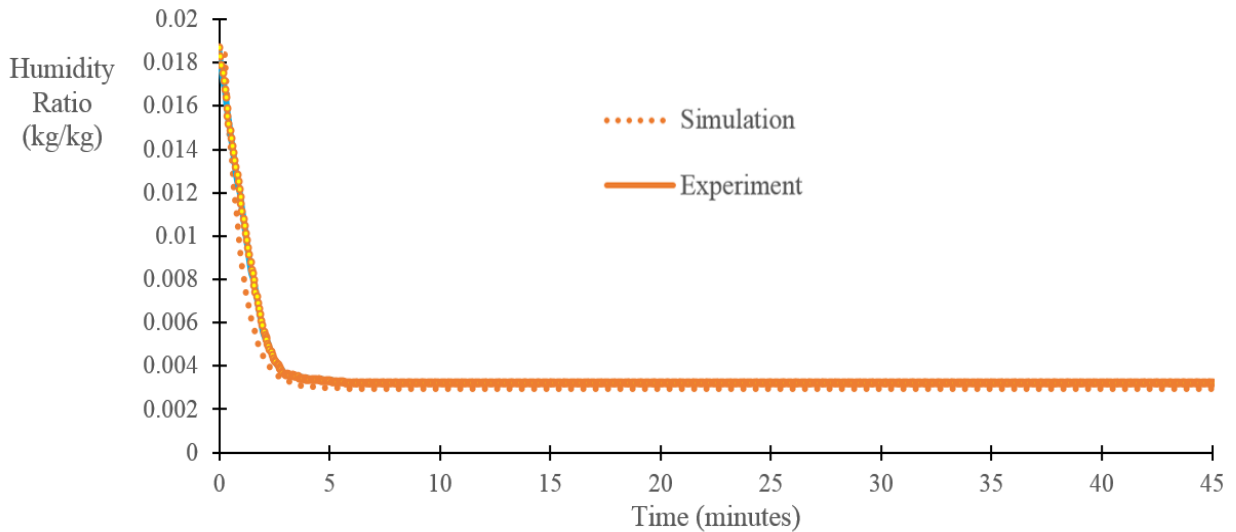


Figure 6.16 Comparison of simulation and experimental results of humidity ratio versus time at velocity $v_5=10$ m/s

An extensive comparison between experimental and simulation results was made, and it shows it differs from one to another. As shown in Figure 6.12, when the air velocity around $v_1=2$ m/s, the humidity ratio was around $0.009621 \text{ kg}_{\text{water}}/\text{kg}_{\text{dry air}}$. Whereas, the simulation results show that it reached about $0.009399 \text{ kg}_{\text{water}}/\text{kg}_{\text{dry air}}$ for humidity ratio. In Figure 6.13, we increased the velocity to $v_2=4$ m/s; the humidity ratio was $0.007795 \text{ kg}_{\text{water}}/\text{kg}_{\text{dry air}}$ for the experiment and

0.007397 $\text{kg}_{\text{water}}/\text{kg}_{\text{dry air}}$ for simulation results. Next, when the velocity was $v_3=6$ m/s, as shown in Figure 6.14, the experimental results show that the humidity ratio was 0.006027 $\text{kg}_{\text{water}}/\text{kg}_{\text{dry air}}$, and the simulation otherwise shows the humidity ratio 0.005599 $\text{kg}_{\text{water}}/\text{kg}_{\text{dry air}}$. After that, in Figure 6.15, we again increased the air velocity to $v_4=8$ m/s. The experimental data exhibited a humidity ratio of 0.004108 $\text{kg}_{\text{water}}/\text{kg}_{\text{dry air}}$, and simulation results show a humidity ratio of 0.003861 $\text{kg}_{\text{water}}/\text{kg}_{\text{dry air}}$. Finally, at $v_5=10$ m/s, as shown in Figure 6.16, the experimental data shows the humidity ratio of 0.002511 $\text{kg}_{\text{water}}/\text{kg}_{\text{dry air}}$, and for simulation data displayed, the humidity ratio was about 0.002463 $\text{kg}_{\text{water}}/\text{kg}_{\text{dry air}}$.

Table 6.6 Percentage differences between simulation and experiment

Items	Air velocity (m/s)				
	2	4	6	8	10
Experiment ($\text{kg}_{\text{water}}/\text{kg}_{\text{dry air}}$)	0.009821	0.007795	0.006027	0.004108	0.002511
Simulation ($\text{kg}_{\text{water}}/\text{kg}_{\text{dry air}}$)	0.009399	0.007397	0.005599	0.003861	0.002463
Differences (%)	4.3	5.1	7.1	6.0	1.9

Based on these results, we made further analysis and gathered them into Table 6.6. As we can see, there is a good agreement between the simulation and experiment where the highest error is 7.1% at velocity 6 m/s, and the lowest is 1.9% at velocity 10 m/s. We can say that the error or differences between experimental and simulation is relatively small, which is acceptable for this research. For research project purposes, the error tolerance between experimental and simulation results should be below 10%. However, it is still not changing the fact that there is still an error in our study. It occurs because some small leakage and instability during the experiment affect the flow of air and liquid in the novel liquid desiccant wheel cooling system.

Nevertheless, this is an early prototype, and this kind of problem is common. However, a subsequent and improved version of this prototype can minimise this kind of error in the future. Comparing these humidity ratio results shows that this simulation can have successfully predicted the output of the liquid desiccant cooling system. It is a massive step in ultimately predicting the liquid desiccant cooling system's performance with a more complex arrangement and process.

6.9 Summary

Poor thermal comfort in air-conditioned space is due to an uneven temperature and relative humidity, affecting the occupants' level of comfort and productivity. The conventional vapour compression cycle consumes a large amount of energy for their operations. Most conventional systems use fossil fuels and non-environmentally friendly sources of energy. This reason drove our effort to explore more efficient and cleaner alternative energy.

The novel liquid desiccant wheel cooling system that was developed and investigated here is considered environmentally friendly and does not include the use of dangerous substances that are harmful to nature. It shows the innovation and creativity on how to utilise rotor wheel with the liquid desiccant technology for dehumidification and cooling purposes, thus contributing to new findings and knowledge. The developed system can be used in many applications, such as commercial buildings, residential houses, supermarkets, and any other buildings that require a high dehumidification rate.

An extensive and comprehensive set of experiments has been done to study the effect of various parameters on the novel liquid desiccant wheel cooling system's performance. A preliminary and complete test rig has been developed and equipped with good instrumentation to ensure the effectiveness of data collected from the system. The temperature and humidity at various inlet and outlet in the dehumidifier and regenerator channel were measured with temperature and humidity probes. The air velocity at both air inlet and outlet was measured by using an air velocity transducer. Experimental data were recorded by using a data logger, which was connected to the computer. The data gathered in this chapter has shown positive results on dehumidification, heating, and regeneration and cooling sections that genuinely reflect this system's potential. In terms of overall performance, this prototype shows a very energy efficient cooling system with the highest COP_{Total} recorded was around 7.547. However, bear in mind that these results are only part of the findings thus far, but it is sufficient to encourage further research on optimisation and improvement in such a system.

This chapter also shows that the simulation used in the simulation can predict the real output data of the novel liquid desiccant wheel cooling system. The simulation error is from 1.7% to 7.1% and is good tolerance for any cooling system. The modelling is relatively simple but very efficient in simulating our desired output. Further analysis and modelling can be developed if the system was integrated with more complex arrangements such as multi-stage or deep dehumidification.

Chapter 7

Environmental & Economic Analysis

7.1 Introduction

Based on the Chapters 3,4,5 and 6, the wheel desiccant cooling system was investigated extensively. The analyses were carried out, and the results proved that the heating and cooling energy heavily influence the *COP*. So in this system, utilising renewable energy such as solar energy is the key to reduce fossil-fuel energy's dependency and eliminate CO₂ emissions. Therefore, this system's economics and environmental perspective is highly recommended to prove that this system can contribute to a new design of an air conditioning system.

7.2 Environmental Analysis Tools

This analysis aims to estimate the absolute values of the system in accordance with environmental aspects. In this section, it is possible to make reasonably rough calculations, and some simplifications are acceptable for the sake of analysis. In this section, the life cycle plays a vital role starting from production stages until the end of its life span, which is at the disposal stage. The study included the environmental impact at the production phase because several production methods have different environmental burdens. Thus, the environmental impact of energy production for residential cooling is studied using the model that we used here to describe the environmental analysis for our prototype.

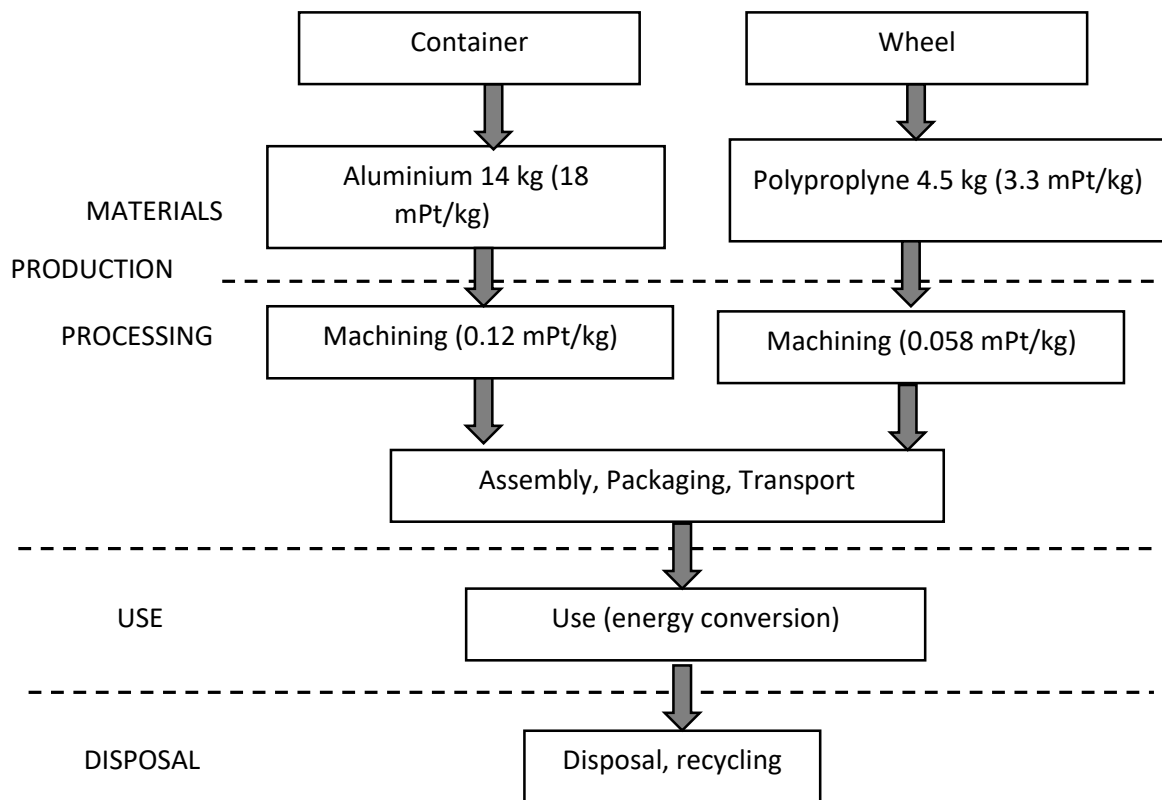


Figure 7.1. Process tree of wheel desiccant cooling with material data and assumptions.

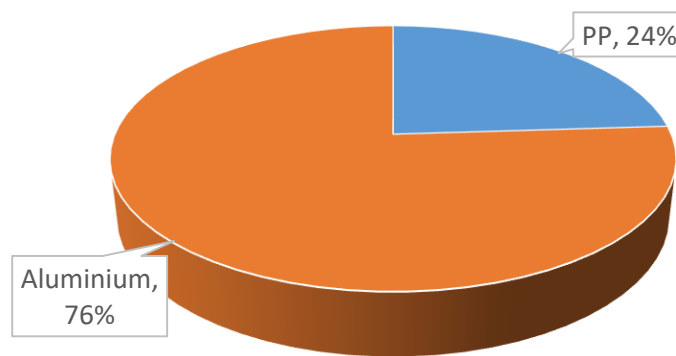


Figure 7.2. Eco-indicator values for wheel desiccant cooling, Aluminum is 0.253 Pt, PP is 0.015 Pt, total 0.268 Pt

The environmental analysis that was made in this study is made by using the Eco-indicator 95 method (Matjaz Prek, 2004). The guideline values for the materials and processes are derived from the design specifications, using the table values from Mark Goedkoop (1995). A

simplified life cycle of an air conditioning system with point values is given in Figure 7.1. The formula to calculate the points are as follows:

$$\text{Points} = \text{weight} \times (\text{materials (mPt/kg)} + \text{process (mPt/kg)}) \quad (7.1)$$

The material and process selection covers the whole range of elements, which are necessary for the specific cooling system. Such a process tree provides useful insight into further analysis, as all elements of the system are analysed with the same procedure.

In the case of the novel liquid desiccant wheel cooling system, two primary materials were analysed. We chose aluminum and polypropylene (PP) because these materials covered the vast majority of the material for this system. Both materials were fabricated in-house, whereas other materials were bought and installed directly to the system, such as the pump. In Figure 7.1, we measured the weight of two major materials for production, which are aluminium (14 kg) and PP (4.5 kg). Both materials then underwent the necessary machining process to fit into the wheel liquid desiccant cooling system. Once the component is fabricated, it will be assembled, packaged, and transported to the consumer. After several usages by the consumers, we assume it will be disposed of, and some parts can be recycled.

Figure 7.2 presented the result of analysis for wheel desiccant cooling system, gathered via the Eco-indicator method. The results reveal the great importance of aluminium, which represented about 76% of the overall impact, while PP stood at 24%. Then, the Eco-indicator shows that aluminium and PP would score about 0.253 Pt (point) and 0.015 Pt, respectively. For the calculation example, we can acquire the point value using Formula 7.1. The calculation example points for aluminium:

$$\begin{aligned}\text{points} &= 14\text{kg} \times (18 \times 10^{-3} \text{ Pt/kg} + 0.12 \times 10^{-3} \text{ Pt/kg}) \\ &= 0.253 \text{ Pt.}\end{aligned}$$

while for PP,

$$\begin{aligned}\text{points} &= 4.5 \text{ kg} \times (3.3 \times 10^{-3} \text{ Pt/kg} + 0.058 \times 10^{-3} \text{ Pt/kg}) \\ &= 0.015 \text{ Pt.}\end{aligned}$$

From those values, the total Eco-indicator would be 0.268 Pt. This value would be much higher if other materials were chosen, such as copper, since it has high points, thus will increase the overall impact. So having this value could be reflected as good as it produces less harm to the environment when taking account of the system's life cycle.

7.3 Equipment Capital and Running Cost

In order to explain the economic aspect of the wheel desiccant cooling system, it is best to compare it with other traditional HVAC systems available in the market today. When we talk about economic analysis, the most significant factor we look into is the payback period. The formula for the payback period is:

$$\text{Payback Period} = \frac{\text{Initial Investment}}{\text{Cash Inflow}} \quad (7.2)$$

where,

Initial Investment = early cost differences between the new invention and the conventional system

Cash Inflow = Net annual cost of system maintenance (such as energy to run the system, materials replacement and etc.)

Table 7.1 shows how we calculate the equipment capital and running cost of the novel liquid desiccant wheel cooling system. Firstly, we compare in terms of capital cost for both systems where the traditional HVAC has only one capital cost while the new system has separated components that have several costs, resulting in higher capital cost. Then we look at the maintenance cost where the traditional system is a bit higher than the new system. The dehumidifier and regenerator wheel will deteriorate for long-term running in the novel liquid desiccant wheel cooling system and need to be replaced each year. This replacement will cost just around £96.98 each year.

Table 7.1 Equipment Capital and Running Cost

Items	Traditional HVAC System	Wheel Liquid Desiccant Cooling System	
Capital Cost	2500	Dehumidifier + Regenerator	£1100
		Solar Collector (Web 7.1)	£475.20
		Cooling Tower (Web 7.2)	£1299
		Working Solution (5kg) (Web 7.3)	£310
		Total Cost	£3,184.20
Maintenance Cost	250	Dehumidifier +Regenerator material replacement (Web 7.4)	£96.98
		Cooling tower (Web 7.5)	£50
		Total Cost	£146.98
COP	3	7.547	
Total Input Energy (electricity)	1.37kW	1.017 kW	
Total Input Energy (gas)	3.74 kW		
Energy Price (electricity)(Web 7.6)	14.01 p/kWh	14.01 p/kWh	
Energy Price (gas) (Web 7.6)	4.12 p/kWh		
Annual Running Hours	8 hours per day for 3 months (720 hours)	8 hours per day for 3 months (720 hours)	
Annual Running Cost	£251.19	£102.51	
Payback year	2.72		

The standard *COP* value of the conventional vapour compression air conditioning system is 3.0 (S.B. Riffat and Guoquan Qiu, 2004). The *COP* of the novel liquid desiccant wheel cooling system that we obtained is around 7.547. Then, we identified the total input energy needed from both systems. It was stated that the traditional system required 1.37 kW and 3.74 kW for electricity and gas energy input, respectively. Whereas the new system only requires 1.017 kW of electrical energy input. The installation of a solar collector will help to reduce the dependency of external heating for the system. We also listed the current energy price for electricity and gas to calculate the actual annual running cost for both systems. It is assumed that the air conditioning system running 8 hours per day to cool the office or houses by air conditioning each year. The system will have fully utilised solar energy during summer from May to July in the UK. That is why we considered around 720 hours per year of solar energy usage to save energy needed for the wheel desiccant cooling system. By using the information from Table 7.1, we can calculate the payback period for the system using Formula 7.2, as shown:

$$\begin{aligned} \text{Payback period} &= \frac{(\text{£}3184.20 - \text{£}2500)}{(\text{£}250 + \text{£}251.19)/\text{year} - (\text{£}146 + \text{£}102.51)/\text{year}} \\ &= 2.72 \text{ years} \end{aligned}$$

This calculation has considered the initial cost, maintenance, and annual running cost of both traditional HVAC and the novel liquid desiccant cooling wheel system. It means that it will take around 2.72 years for the system to pay off its initial investment. Based on Table 1, we found out that this system can save 2,946.96 kWh of energy per year compared with the traditional HVAC system. This savings is due to the implementation of renewable energy, such as solar energy, into the system. For the fossil-fuels energy, 1 kWh power would release about

0.6083 kg CO₂ into ambient air (Liu, 2008). Thus, this system can prevent at least 1792.63 kg of CO₂ emission into the atmosphere per year.

On the other hand, it was realised that specific components' costs might rise with a subsequent year. That is why some scenario was added to predict the payback period if the capital cost increased annually (E. W. Larson & C. F. Gray, 2011). For example, if the capital cost increased around 10%, 20%, 30%, 40%, and 50% of the initial cost, how much it will affect the payback period in the future. These percentage ranges were chosen to represent the cost that might occur for any product under development. As shown in Figure 7.3, the increase in capital cost has shown an exceptionally high payback period. It has been expected since the increase in capital costs, such as 50%, would give a payback period of 9.04 years, which is quite long. Nevertheless, this analysis only considers the worst-case scenario because it is not common for any capital cost to increase drastically in such a short period (J.H. Buys & E.H. Mathews, 2005). Thus, this is a fair assessment for the system if there is any drastic capital cost increment and how it will affect investors interested in investing in such a project.

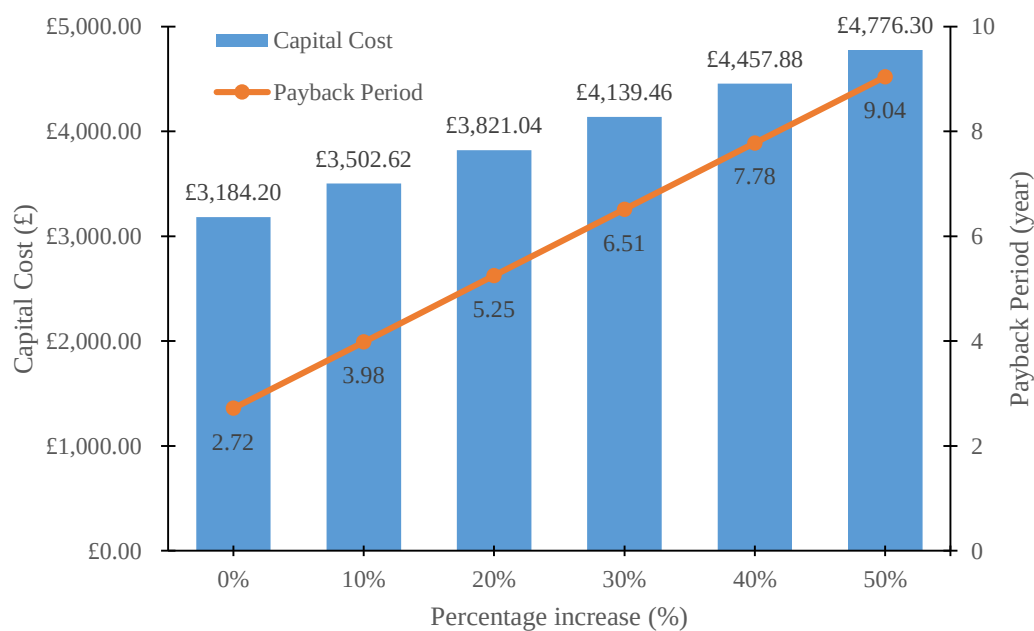


Figure 7.3 Total capital cost and payback period versus percentage increase of capital cost

Besides that, an analysis of the effect of energy prices has also been studied, as shown in Figure 7.4. In this study, the energy price averagely increased by about 5.3% each year in the UK [Web 7.7]. It is appropriate to analyse energy prices because of the lack of information on the material price history from suppliers and the labour cost fluctuation in the last five years. Thus, the forecast has been made only for energy prices for the next five years. The capital cost was fixed for this analysis. It was interesting to see that the payback period was improved with increasing energy prices each year. From this figure, 0 % is the current year, 5.3% is after one year, 10.6% is after two years, 15.9% is after three years, 21.2% is after four years, and 26.5% is after five years. For instance, currently, the payback period is shorter at year 5 (26.5%), which 2.34 years compared to the current energy price, which is 2.72 years. The main reason was when the payback period's formula was referred, and it was found that the cash inflow increased each year that contribute to the lower value of the payback period. The cash inflow increased because when energy price increased, the conventional system's energy cost increased dramatically compared to the new system that required less energy. This system will save more money annually if the energy price becomes high, and it can be a valuable solution in the future.

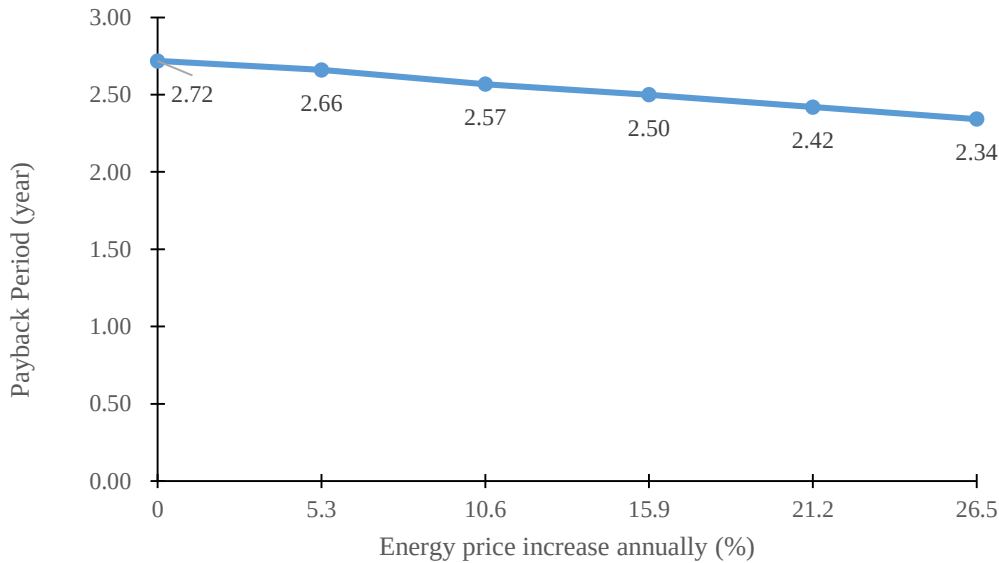


Figure 7.4 Payback period versus year of energy price increased

7.4 Summary

The Eco-indicator 95 method has been used for the environmental analysis and optimisation of air conditioning systems. This method enables environmentally aware design and is an open-working method with a platform on which both industry and science can integrate the environmental aspects into the design development. The result allows the consumer to see how much environmental impact these design alternatives will have. The designer can quickly and effectively evaluate the consequences of an idea and establish clear selection standards for an idea. Environmental impacts from the production processes of products and materials are taken into consideration. This research showed that the novel liquid desiccant wheel cooling system with certain construction materials would influence the Eco-indicator value. From the results, the Eco-indicator 95 shows the value of 0.269 points, which is relatively low compared to other air conditioning and heating systems developed, which is 1.359 points. (Matjaz Prek, 2004)

Economic analysis of the conventional HVAC system and the novel liquid desiccant wheel cooling system were also investigated. It is known that it can save at least 4.093 kWh of energy

than the traditional HVAC system. The novel liquid desiccant wheel cooling system's base capital is £684 higher than a conventional HVAC system. The wheel materials need to be replaced annually, where it cost about 96.98 pounds per year. It is for the sake of maintaining the dehumidifier and regenerator performance in the system. The payback period is not that long as it only takes 2.72 years to return the initial investment in the early stage. It is also shown that it is an effective way to save fossil-fuels energy and reduce CO₂ emission compared to using the conventional HVAC system.

Chapter 8

Conclusion & Recommendation

8.1 Conclusions

In recent years we have seen many research and development projects on dehumidification and cooling technologies to seek improvement in this area. These technologies are considered environmentally friendly and can reduce hydrochlorofluorocarbons (HCFC) and chlorofluorocarbons (CFC) based on conventional air conditioning systems. Much attention has been given to new methods and materials associated with desiccant dehumidification cooling so that this technology becomes more efficient and cost-effective. For this purpose, the novel liquid desiccant wheel system has been developed to contribute a significant knowledge on the desiccant cooling technology.

In this thesis, an extensive literature review of previous work on desiccant dehumidification cooling was conducted. The review in Chapter 2 was initially focused on the desiccant dehumidification system that caters to both solid and liquid desiccant as the working mechanism. Besides dehumidification, we also covered the regeneration system and its methods. All types and arrangements of dehumidification, such as multistage dehumidification cooling and hybrid systems, were also covered widely.

This research also investigated the selection of desiccant absorbent materials that can be used in the novel liquid desiccant wheel cooling system. Various desiccant candidates were investigated based on the literature review, and the material with the best properties was selected for the system. As a result, PP for wheel material and potassium formate as the working fluid was chosen for this research.

To answer all the objectives highlighted in Chapter 1 of this thesis, let's take a look at what kind of achievement has to be acquired to fulfil those objectives. Firstly, an extensive literature review and ways to integrate the rotor wheel's best features and liquid desiccant into a hybrid system have been carried out. All the necessary literature reviews have been presented in Chapter 2 that mainly cater to the solid, liquid, and hybrid desiccant cooling technology. The rotor wheel's features and description, liquid desiccant, and the overall system have been highlighted in Chapter 3. Secondly, the optimising of wheel sizing and predicting the system's final output by using the developed mathematical model also successfully done. It was shown that the model simulated using EES software manages to give a proper sizing of the rotor wheel and successfully predicts the output can be obtained under various operating conditions, as shown in Chapter 4. Thirdly, conducting the experimental work to study the system's physical and chemical behaviours by using full-scale laboratory testing has also been completed. An extensive procedure of the experiment also has been explained in Chapter 5. Besides that, some full-scale laboratory testing has been highlighted in the earlier part of Chapter 6. Fourthly, analysing the specific experimental results and comparing them with the simulation results has been successfully done. This achievement is significant because the result came out from the experiment will determine the effectiveness in fulfilling the objective that has been set. In the end, good COP results have been achieved, and it manages to reflect the same results as simulation work, as shown in Chapter 6. Finally, but not least, the economic and environmental

analysis also has been successfully carried out. It is essential that the feasibility and practicality of the system were conducted properly. The analysis using Eco-indicator 95 shows tremendous results where the minimal impact on the environment can be seen following the prototype's fabrication.

The economic analysis also shows that the system can give a short payback period compare to the conventional cooling system, as shown in Chapter 7. Following is the summary of the prototypes that have been developed.

8.2 The Evaluation of the System using Simulation

Besides experimental-based testing, we also conducted a simulation to evaluate and validate the performance of the system. In the simulation, we varied the air velocity and other environmental properties. Based on the results, we noticed that air velocity could significantly influence the novel liquid desiccant wheel cooling system's performance. It has become our fundamental knowledge prior to the real experiment at the later stage. Besides that, the experiment results show almost a similar pattern with the one obtained in the simulation. Besides that, the proper sizing for the system is also determined and useful before fabrication. This shows that this experiment can be well predicted and simulated using valuable tools such as the Engineering Equation Solver (EES) software.

8.3 The Performance of the Whole System Under Different Air Velocity

As discussed in the previous section, all the tests involving the complete system will be summarized. A various test was carried out at different inlet air velocity and other different

conditions. Based on test results, it was revealed that moisture was removed from incoming process air that came from inlet air in the system. Moisture removal kept increasing when we increased the inlet air velocity. Besides that, we observed that the temperature supply would decrease, and as a result, colder air will flow into the intended space. The higher the incoming air velocity we set into the system, the more air will be dehumidified and cooled by this system. This system relatively used less energy as it only needs to powered electric motor and pump that run the system, which is less significant when considering the system's cooling effect.

8.4 Contribution and Innovation to Knowledge

Based on the results that have been gathered, the performance of the novel liquid desiccant wheel cooling system has performed well compared to the conventional system available in the market. Based on both system's coefficient of performance (*COP*), the conventional system shows a *COP* of 3.0, whereas our prototype indicates a *COP* of 7.547. This prototype also shows that it is possible to develop a high-performance liquid desiccant cooling system that uses inspiration from a wheel mechanism that usually available in solid desiccant system. Due to this fact, it contributed to a higher *COP* than any conventional cooling system. Moreover, the system shows a low payback period where it only takes 2.72 years to earn the investment return. In terms of environmental analysis, by using the Eco-indicator 95 method, it only indicates 0.269 points where it imposes a low effect on the environment.

8.5 Further Work and Recommendation

Based on the results that we obtained throughout this project, it was believed that substantial improvement could be made in a variety of aspects. For instance, the analysis could have been better on the simulation side if we had considered other simulation tools, such as Matlab.

Nevertheless, EES is thermodynamically friendly to use as all the property table is available in the software, whereas in Matlab, we need to start from scratch to build the simulation tools. However, Matlab is known for its good computation accuracy and may perform slightly different results than EES.

This research work also could reflect better performance if it could have identified better wheel materials for the heart of dehumidification. The polymer fibre was suitable for dehumidification but probably lacks durability when dealing with long-term usage and hazardous conditions. Wheel rotor made of composite or hybrid materials could do the trick by providing the toughness and rigidity needed and provide the dehumidification required by the system. However, it was to make sure the system can commence admirably for the time being, and the PP material is a viable solution for the project. For the next stage, using a hybrid material could be the next step for this kind of prototype.

Another area that can be improved is to find a better solution for the working fluid that runs through the vein of this prototype. Potassium formate shows good characteristics, but there is always another desiccant solution that can perform better dehumidification capabilities. One of the suggested solutions is to use the ionic liquid for future work. It is one of the most promising desiccant solutions, as many researchers have started it in their projects.

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Appendix

Appendix A

Programming of the System Using EES

EES coding of Psychrometric chart of air properties

"This problem is an exercise in determining the properties of atmospheric air given the atmospheric pressure and any other two compatible intensive properties from the following list: relative humidity, dry-bulb temperature, wet-bulb temperature, specific humidity (humidity ratio), and dew point temperature."

"After specifying the atmospheric pressure on the diagram window, we select two independent intensive properties, and EES determines the other properties."

`$checkUnits off`

`$local off`

Procedure Find(Prop1\$,Prop2\$,Value1,Value2,P:Tdb,Twb,Tdp,h,v,Rh,w,pl)

"Due to the very general nature of this solution, a large number of 'if-then-else' statements are necessary."

`k=StringPos('/',Prop1$);`

`if (k>0) then Prop1$:=copy$(Prop1$,1,k-1)`

`k=StringPos('/',Prop2$)`

`if (k>0) then Prop2$:=copy$(Prop2$,1,k-1)`

`If Prop1$='Dry-bulb Temperature' Then`

`Tdb=Value1`

`pl=1`

`If Prop2$='Dry-bulb Temperature' then Call Error('Both properties cannot be Dry-bulb Temperature, Tdb=xxx°F',Tdb)`

`if Prop2$='Relative Humidity, 0 to 1' then`

`Rh=Value2`

`pl=2`

`h=enthalpy(AirH2O,T=Tdb,P=P,R=Rh)`

`v=volume(AirH2O,T=Tdb,P=P,R=Rh)`

`Twb=wetbulb(AirH2O,T=Tdb,P=P,R=Rh)`

`Tdp=dewpoint(AirH2O,T=Tdb,P=P,R=Rh)`

`w=humrat(AirH2O,T=Tdb,P=P,R=Rh)`

`endif`

`if Prop2$='Wet-bulb Temperature' then`

`Twb=value2`

`pl=3`

`if Twb>Tdb then Call Error('These values of Dry-bulb Temperature and Wet-bulb Temperature are incompatible, Tdb=xxx°F',Tdb)`

`h=enthalpy(AirH2O,T=Tdb,P=P,B=Twb)`

`v=volume(AirH2O,T=Tdb,P=P,B=Twb)`

`Tdp=dewpoint(AirH2O,T=Tdb,P=P,B=Twb)`

`w=humrat(AirH2O,T=Tdb,P=P,B=Twb)`

`Rh=relhum(AirH2O,T=Tdb,P=P,B=Twb)`

`endif`

`if Prop2$='Dew Point Temperature' then`

`Tdp=value2`

```

        pl=4
        if Tdp>Tdb then Call Error('These values of Dry-bulb Temperature and Dew Point
Temperature are incompatible, Tdb=xxx°F2',Tdb)

        h=enthalpy(AirH2O,T=Tdb,P=P,D=Tdp)
        v=volume(AirH2O,T=Tdb,P=P,D=Tdp)
        Twb=wetbulb(AirH2O,T=Tdb,P=P,D=Tdp)
        w=humrat(AirH2O,T=Tdb,P=P,D=Tdp)
        Rh=relhum(AirH2O,T=Tdb,P=P,D=Tdp)
    endif

    if Prop2$='Enthalpy' then
        h=value2
        pl=5
        Twb=wetbulb(AirH2O,T=Tdb,P=P,H=h)
        w=humrat(AirH2O,T=Tdb,P=P,H=h)
        Rh=relhum(AirH2O,T=Tdb,P=P,H=h)
        Tdp=dewpoint(AirH2O,T=Tdb,P=P,w=w)
        v=volume(AirH2O,T=Tdb,P=P,R=Rh)
    endif

    if Prop2$='Humidity Ratio' then
        w=value2
        pl=6
        h=enthalpy(AirH2O,T=Tdb,P=P,W=w)
        v=volume(AirH2O,T=Tdb,P=P,W=w)
        Twb=wetbulb(AirH2O,T=Tdb,P=P,W=w)
        Tdp=dewpoint(AirH2O,T=Tdb,P=P,w=w)
        Rh=relhum(AirH2O,T=Tdb,P=P,w=w)
    endif
Endif

If Prop1$='Dew Point Temperature' Then
    Tdp=Value1
    pl=7
    If Prop2$='Dew Point Temperature' then Call Error('Both properties cannot be Dew Point
Temperature, Tdp=xxx°F2',Tdp)
    if Prop2$='Relative Humidity, 0 to 1' then
        Rh=Value2
        pl=8
        h=enthalpy(AirH2O,D=Tdp,P=P,R=Rh)
        Tdb=temperature(AirH2O,h=h,P=P,R=RH)
        v=volume(AirH2O,T=Tdb,P=P,R=Rh)
        Twb=wetbulb(AirH2O,T=Tdb,P=P,R=Rh)
        w=humrat(AirH2O,B=Twb,P=P,R=Rh)
    endif

    if Prop2$='Wet-bulb Temperature' then
        Twb=value2
        pl=9
        if Tdp>Twb then Call Error('These values of Dew Point Temperature and Wet-bulb
Temperature are incompatible, Tdp=xxx°F3',Tdp)
        Pw=pressure(steam,T=Twb,x=1)
        Pv=pressure(steam,T=Tdp,x=1)
        "Pv=Pw-(P-Pw)*(Tdb-Twb)*1.8/(2800-1.3*(1.8*Twb+32)) Carrier Equation"
        Tdb=Twb+(Pw-Pv)/(P-Pw)*(2800-1.3*(1.8*Twb+32))/1.8
        h=enthalpy(AirH2O,T=Tdb,P=P,D=Tdp)
        Rh=relhum(AirH2O,T=Tdb,P=P,D=Tdp)
        v=volume(AirH2O,T=Tdb,P=P,D=Tdp)
        w=humrat(AirH2O,T=Tdb,P=P,D=Tdp)
    endif

```

```

endif

    if Prop2$='Enthalpy' then
    h=value2
    pl=10
    Tdptest=temperature(AirH2O,h=h,P=P,R=1)
    if Tdp>Tdptest then Call Error('These values of Dew Point Temperature and Enthalpy
are incompatible, Tdp=xxx°F3',Tdp)
    Pv = pressure(steam, T=Tdp, x=1)
    w=molarmass(steam)/molarmass(air)*Pv/(P-Pv)
    Tdb=temperature(airH2O,h=h,P=P,w=w)
    Twb=wetbulb(AirH2O,T=Tdb,P=P,D=Tdp)
    Rh=relhum(AirH2O,T=Tdb,P=P,D=Tdp)
    v=volume(AirH2O,T=Tdb,P=P,R=Rh)

endif

    if Prop2$='Humidity Ratio' then
    w=Value2
    pl=11
    Call Error('The properties cannot be Dew Point Temperature and Humidity Ratio,
Tdp=xxx°F3',Tdp)
    endif
Endif

If Prop1$='Wet-bulb Temperature' Then
    Twb=Value1
    pl=12
    If Prop2$='Wet-bulb Temperature' then Call Error('Both properties cannot be Wet-bulb
Temperature, Twb=xxx°F2',Twb)
    if Prop2$='Relative Humidity, 0 to 1' then
        Rh=Value2
        pl=13
        Tdb=temperature(AirH2O,B=Twb,P=P,R=RH)
        h=enthalpy(AirH2O,T=Tdb,P=P,R=Rh)
        v=volume(AirH2O,T=Tdb,P=P,R=Rh)
        Tdp=dewpoint(AirH2O,T=Tdb,P=P,R=Rh)
        w=humrat(AirH2O,B=Twb,P=P,R=Rh)

    endif

    if Prop2$='Dew Point Temperature' then
        Tdp=value2
        pl=14
        if Tdp>Twb then Call Error('These values of Wet-bulb Temperature and Dew Point
Temperature are incompatible, Twb=xxx°F3',Twb)
        Pw=pressure(steam,T=Twb ,x=1)
        Pv=pressure(steam,T=Tdp ,x=1)
        "Pv=Pw-(P-Pw)*(Tdb-Twb)*1.8/(2800 -1.3*(1.8*Twb+32)) Carrier Equation"
        Tdb=Twb+(Pw-Pv)/(P-Pw)*(2800-1.3*(1.8*Twb+32))/1.8
        h=enthalpy(AirH2O,T=Tdb,P=P,D=Tdp)
        Rh=relhum(AirH2O,T=Tdb,P=P,D=Tdp)
        v=volume(AirH2O,T=Tdb,P=P,D=Tdp)
        w=humrat(AirH2O,T=Tdb,P=P,D=Tdp)

    endif

    if Prop2$='Enthalpy' then
    pl=15
    Call Error('The properties cannot be Wet-bulb Temperature and Enthalpy, Twb=xxx°F3',Twb)
    endif

    if Prop2$='Humidity Ratio' then

```

```

        w=value2
        pl=16
        Tdb=temperature(AirH2O,B=Twb,P=P,w=w)
        h=enthalpy(AirH2O,T=Tdb,P=P,W=w)
        v=volume(AirH2O,T=Tdb,P=P,W=w)
        Twb=wetbulb(AirH2O,T=Tdb,P=P,W=w)
        Tdp=dewpoint(AirH2O,T=Tdb,P=P,w=w)
        Rh=relhum(AirH2O,T=Tdb,P=P,w=w)
    endif
Endif

If Prop1$='Relative Humidity, 0 to 1' Then
    Rh=Value1
    pl=17
    If Prop2$='Relative Humidity, 0 to 1' then Call Error('Both properties cannot be Relative
Humidity, Rh=xxxF2',Rh)
    if Prop2$='Wet-bulb Temperature' then
        Twb=value2
        pl=18
        Tdb=temperature(AirH2O,B=Twb,P=P,R=RH)
        h=enthalpy(AirH2O,T=Tdb,P=P,B=Twb)
        v=volume(AirH2O,T=Tdb,P=P,B=Twb)
        Tdp=dewpoint(AirH2O,T=Tdb,P=P,B=Twb)
        w=humrat(AirH2O,T=Tdb,P=P,B=Twb)

    endif

    if Prop2$='Dew Point Temperature' then
        Tdp=value2
        pl=19
        h=enthalpy(AirH2O,R=Rh,P=P,D=Tdp)
        Tdb=temperature(AirH2O,h=h,P=P,R=Rh)
        v=volume(AirH2O,T=Tdb,P=P,D=Tdp)
        Twb=wetbulb(AirH2O,T=Tdb,P=P,D=Tdp)
        w=humrat(AirH2O,T=Tdb,P=P,D=Tdp)
    endif

    if Prop2$='Enthalpy' then
        h=value2
        pl=20
        Tdb=temperature(AirH2O,h=h,P=P,R=Rh)
        w=humrat(AirH2O,h=h,P=P,R=Rh)
        Twb=wetbulb(AirH2O,T=Tdb,P=P,H=h)
        Tdp=dewpoint(AirH2O,T=Tdb,P=P,w= w)
        v=volume(AirH2O,T=Tdb,P=P,R=Rh)
    endif

    if Prop2$='Humidity Ratio' then
        w=value2
        pl=21
        Tdb=temperature(AirH2O,R=Rh,P=P,w=w)
        h=enthalpy(AirH2O,T=Tdb,P=P,W=w)
        v=volume(AirH2O,T=Tdb,P=P,W=w)
        Twb=wetbulb(AirH2O,T=Tdb,P=P,W=w)
        Tdp=dewpoint(AirH2O,T=Tdb,P=P,w=w)
    endif
Endif

If Prop1$='Enthalpy' Then
    h=Value1

```

```

pl=22
If Prop2$='Enthalpy, kJ/kg' then Call Error('Both properties cannot be Enthalpy, h=xxxF2',h)
if Prop2$='Relative Humidity, 0 to 1' then
    Rh=Value2
    pl=23
    Tdb=temperature(AirH2O,h=h, P=P,R=Rh)
    v=volume(AirH2O,T=Tdb,P=P,R=Rh)
    Twb=wetbulb(AirH2O,T=Tdb,P=P,R=Rh)
    Tdp=dewpoint(AirH2O,T=Tdb,P=P,R=Rh)
    w=humrat(AirH2O,T=Tdb,P=P,R=Rh)
endif
if Prop2$='Wet-bulb Temperature' then
    pl=24
    Call Error('The properties cannot be Wet-bulb Temperature and Enthalpy,
h=xxxF2',h)
endif

if Prop2$='Dew Point Temperature' then
    Tdp=value2
    pl=25
    Tdptest=temperature(AirH2O,h=h,P=P,R=1)
    if Tdp>Tdptest then Call Error('These values of Dew Point Temperature and Enthalpy
are incompatible h=xxxF2', h)
    Pv = pressure(steam, T=Tdp, x=1)
    w=molarmass(steam)/molarmass(air)*Pv/(P-Pv)
    Tdb=temperature(airH2O,h=h,P=P,w=w)
    Twb=wetbulb(AirH2O,T=Tdb,P=P,D=Tdp)
    Rh=relhum(AirH2O,T=Tdb,P=P,D=Tdp)
    v=volume(AirH2O,T=Tdb,P=P,R=Rh)
endif

if Prop2$='Humidity Ratio' then
    w=value2
    pl=26
    wtest=humrat(AirH2O,h=h,P=P,R=1)
    If w>wtest then Call Error('These values of Humidity Ratio and Enthalpy are
incompatible, h=xxxF2', h)
    Tdb=temperature(airH2O,h=h,P=P,w=w)
    Twb=wetbulb(AirH2O,T=Tdb,P=P,w=w)
    Rh=relhum(AirH2O,T=Tdb,P=P,H=h)
    Tdp=dewpoint(AirH2O,T=Tdb,P=P,w=w)
    v=volume(AirH2O,T=Tdb,P=P,R=Rh)
endif
Endif

If Prop1$='Humidity Ratio' Then
    w=Value1
    pl=27
    If Prop2$='Humidity Ratio' then Call Error('Both properties cannot be Humidity Ratio,
w=xxxF3',w)
    if Prop2$='Relative Humidity, 0 to 1' then
        Rh=Value2
        pl=28
        Tdb=temperature(AirH2O,R=Rh,P=P,w=w)
        h=enthalpy(AirH2O,T=Tdb,P=P,W=w)
        v=volume(AirH2O,T=Tdb,P=P,W=w)
        Twb=wetbulb(AirH2O,T=Tdb,P=P,W=w)
        Tdp=dewpoint(AirH2O,T=Tdb,P=P,w=w)
    endif
    if Prop2$='Wet-bulb Temperature' then

```

```

        Twb=value2
        pl=29
        wtest=humrat(airH2O,B=Twb,P=P,R=1)
        If w>wtest then Call Error('These values of Wet-bulb Temperature and Humidity
Ratio are incompatible, w=xxxF3',w)
        Tdb=temperature(airH2O,B=Twb,P=P,w=w)
        h=enthalpy(AirH2O,T=Tdb,P=P,w=w)
        v=volume(AirH2O,T=Tdb,P=P,B=Twb)
        Tdp=dewpoint(AirH2O,T=Tdb,P=P,B=Twb)
        Rh=relhum(AirH2O,T=Tdb,P=P,B=Twb)
    endif

    if Prop2$='Dew Point Temperature' then
        pl=30
        Call Error('The properties Humidity Ratio and Dew Point Temperature are
incompatible, w=xxxF3',w)
    endif

    if Prop2$='Enthalpy' then
        h=value2
        pl=31
        wtest=humrat(AirH2O,h=h,P=P,R=1)
        If w>wtest then Call Error('These values of Humidity Ratio and Enthalpy are
incompatible, w=xxxF3',w)
        Tdb=temperature(airH2O,h=h,P=P,w=w)
        Twb=wetbulb(AirH2O,T=Tdb,P=P,w=w)
        Rh=relhum(AirH2O,T=Tdb,P=P,H=h)
        Tdp=dewpoint(AirH2O,T=Tdb,P=P,w=w)
        v=volume(AirH2O,T=Tdb,P=P,R=Rh)
    endif
Endif
end

"Input from the diagram window"
{P=101.3" kPa"
Prop1$='Dry-bulb Temperature'
Prop2$='Relative Humidity, 0 to 1'
Value1=24
Value1=0.5}
"For debugging, the variable pl gives the location in the procedure."
{UnitStr$='SI'}
{PUnits$='kPa'
TUnits$='C'
hUnits$='kJ/kg'
vUnits$='m^3/kg'}

Call Find(Prop1$,Prop2$,Value1,Value2, P:Tdb,Twb,Tdp,h,v,Rh,w,pl)
T[1] = Tdb; w[1] =w

```

\$TabWidth 0.5 cm

```

{$NC$ 324 324 324 324 324 324 324 324 325 326 327 328 329 330 325 325 }{$NC$ 325 325 325
325 325 325 325 325 326 327 328 329 330 331 326 326 }{$NC$ 325 325 325 325 325 325 325 325
326 327 328 329 330 331 326 326 }

```

Coding Simulation on Dehumidifier Section

example at 10 m/s

\$UnitSystem SI MASS RAD PA K J
\$TABSTOPS 0.2 0.4 0.6 0.8 3.5 in

```
#####
#####
#####"
```

"Simulation on determine parameters in each slots for dehumidification wheel"

```
#####
#####
#####"
```

```
*****
*****
*****"
```

"Parameters known:"

```
*****
*****
*****"
```

L=0.18[m] "(length of slot)"

l_s=0.004 [m] "(side length of square slot)"

p=l_s*4 [m] "(Square slot perimeters)"

miu=0.000019 [kg/ms] "(dynamic viscosity of air)"

U=10[m/s] "(air speed (face velocity))"

N=10 [-] "(number of element)"

epsilon= 0.8 [-] "(wheel porosity)"

A_h=(l_s*L*4)/N "(internal surface area of each cell)"

A_c=l_s^2[m^2] "(cross section area of each cell)"

rho_d=1908[kg/m^3] "(apparent density of desiccant)"

rho_f=946 [kg/m^3] "(apparent density of wheel)"

Q=2500*10^3 [J/kg] "(latent heat of evaporation)"

C_pf=1.92*10^3 [J/kg*K] "(specific heat of wheel wall)"

C_pd=2.800*10^3 [J/kg*K] "(specific heat of desiccant)"

C_pw=4.18*10^3[J/kg*K] "(specific heat of liquid water)"

C_pa=1.005*10^3 [J/kg*K] "(specific heat of dry air)"

C_pv=2*10^3 [J/kg*K] "(specific heat of water vapor)"

```
*****
*****
*****"
```

"time step:"

```
*****
*****
*****"
```

t_sim=1200 [s] "(simulation time)"

M=401 [-] "(number of time step)"

DELTA t= (t_sim/(M-1)) "(duration of time step)"

duplicate j=1,M

time [j]=(j-1)*t_sim/(M-1) "(time associated with each parameters)"

end

```
*****
*****
*****"
```

"Parameters calculation in each cell in wheel desiccant:"

```
*****
*****
*****"
```

H[1]=0.01912 [kg/kg] "(initial humidity ratio kg water vapor/kg dry air)"

x[1]=0.3 "(initial desiccant moisture content)"

Ts[1]=300 [K] "(initial temperature of desiccant in cell)"

Ta[1]=310 [K] "(initial temperature of air)"

duplicate j=1, (M-1)

$H[j+1] = H[j] + ((N \cdot U \cdot V[j] / L) \cdot (H[j] - H[j+1]) - (k[j] \cdot A_h \cdot N \cdot V[j]) / (L \cdot A_c \cdot \epsilon)) \cdot (H[j+1] - H_s[j]) \cdot \Delta t$
 "(the rate change of humidity - euler's solution)"

$V[j] = (0.00283 + 0.00456 \cdot H[j]) \cdot T_a[j]$ "specific volume of humid air"

$G[j] = U / V[j]$ "(the mass flow rate of humid air)"

$Re[j] = (G[j] \cdot 4 \cdot A_c) / (\rho \cdot \mu)$ "(Reynolds number)"

$k[j] = 0.704 \cdot G[j] \cdot (Re[j])^{(-0.51)}$ "(mass transfer coefficient)"

```
Ta[j+1]=Ta[j]+((N*G[j]/L)*(C_pa+C_pv*H[j])*(Ta[j]-Ta[j+1])-(hc[j]*N*A_h/(L*A_c*epsilon))*(Ta[j+1]-Ts[j])+(k[j]*A_h*N/(L*A_c*epsilon))*(H[j+1]-Hs[j])*(C_pw*Ts[j]+C_pv*(Ta[j]-Ts[j])))*DELTAAt/((C_pa+C_pv*H[j+1])/V[j]) "(Temperature of air in each cell)"
```

```
W[j+1]=W[j]+((k[j]*N*A_h)/(L*A_c*rho_d)*(H[j]-Hs[j]))*DELTAAt "(desiccant moisture content in cell no i)"
```

```
hc[j]=0.671*G[j]*C_pa*(Re[j])^(-0.51) "(heat transfer coefficient between gas and solid phase)"
```

```
Ts[j+1]=Ts[j]+((k[j]*A_h*N)/(L*A_c)*(H[j]-Hs[j])*(Q-C_pw*Ts[j])+(hc[j]*A_h*N)/(L*A_c)*(Ta[j]-Ts[j]))*DELTAAt/(rho_f*C_pf+rho_d*C_pd+rho_d*C_pw*W[j]) "(Temperature of desiccant in cell)"
```

```
x[j]=(1-x[j])/W[j] "(concentration of liquid)"
```

```
Pv[j]=exp((9.8043+11.7292*x[j])-(-265.906+2052.21*x[j])/(Ts[j]-273)+(-10743.8+54141.7*x[j])/(Ts[j]-273)^2) "(Vapor pressure on desiccant surface)"
```

```
Hs[j]=(0.622*Pv[j])/(1.0133*10^(5)-Pv[j]) "(Humidity on desiccant surface)"
```

```
end
```

```
*****
*****
*****"
```

Coding for the whole system

for potassium formate

known values of the system 2-3 (10% concentration)"

```
T_3=60 [C]
T_2=30 [C]
C_p1=3.803 [kJ/(kg*K)]
m_dot_sol= 0.15 [kg/s]
```

"calculate heat transfer from heating system"

```
Q_th=m_dot_sol*C_p1*(T_3-T_2)
```

"Energy input for heating system"

```
E_th=Q_th/COP_hs
```

COP_hs=6 "known value for COP_hs"

"in cooling system 4-1, known values are (concentration high=42.9%)"

```
C_p2=2.846 [kJ/(kg*K)]
T_4=40 [C]
T_1=10 [C]
COP_cool=7
```

$$Q_{cool}=m_{dot_sol}*C_{p2}*(T_4-T_1)$$
$$E_{cool}=Q_{cool}/COP_{cool}$$

"in evaporative cooler after dehumidification"

$$m_{dot_air}=0.16 \text{ [kg/s]}$$

$$T_b=10 \text{ [C]}; P_b=101 \text{ [kPa]}; Rh_b=0.3 \text{ " where } Rh=1=100\% "$$

$$h_b=enthalpy(AirH2O,T=T_b,P=P_b,R=Rh_b)$$

"total of energy consumption in kW (E_total)"

$$E_{input}=E_{cool}+E_{th}$$

$$E_{aux}=0.1*E_{input}$$

$$E_{total}=E_{input}+E_{aux}$$

"Cooling capacity of dehumidifier"

$$T_a=33 \text{ [C]}; P_a=101 \text{ [kPa]}; Rh_a=0.60$$

$$h_a=enthalpy(AirH2O,T=T_a,P=P_a,R=Rh_a)$$

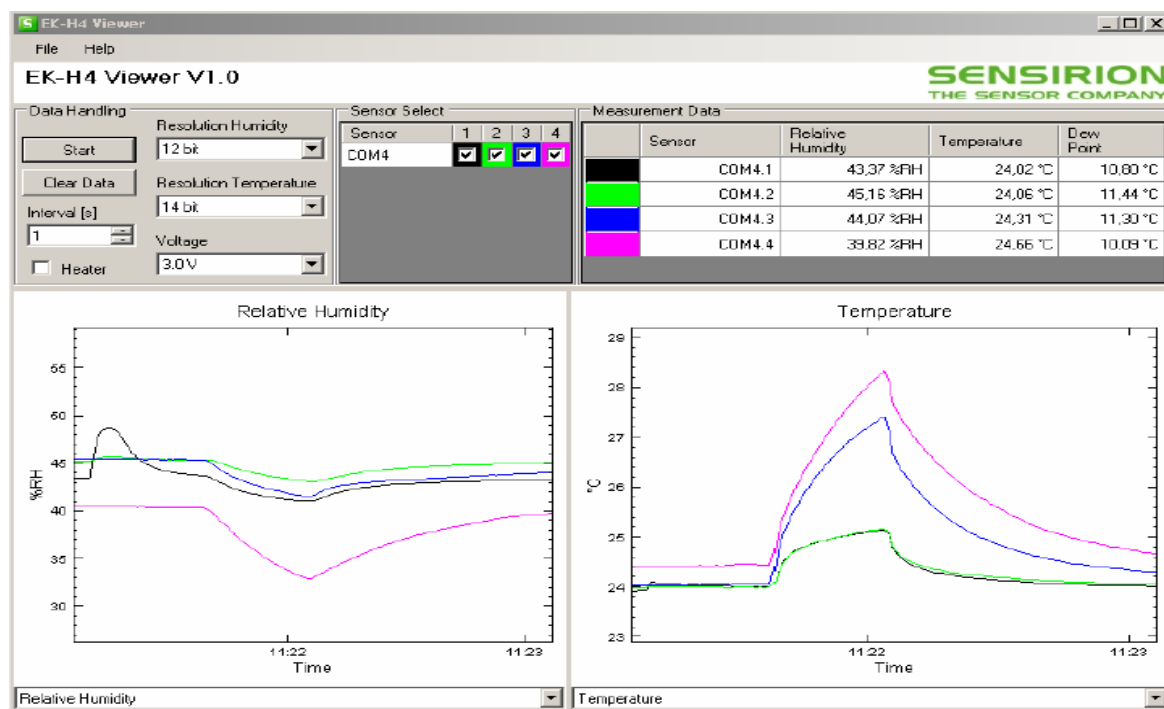
$$CC=m_{dot_air}*(h_a-h_b)$$

$$COP_{total}=CC/(E_{total})$$

Appendix B

Data Logger Setup and Data Collection Process

The data was taken using EK-H4 viewer interface. Select sensor 1, 2, 3 and 4 at the sensor select. In data measurement, it will display the relative humidity, temperature and dew point temperature at a real time. When the button start was press, the data collection begin. When we press the stop button, the data collection stopped and it will be saved the collected data.



The collected data then will be saved in notepad format. Besides that, we can convert the data to cvs format for further data analysis. From here, we analyse each data for humidity, temperature and so on according to time.

File Edit Format View Help

ta from EK-H4

M3.1: SHT2x, 36034904723030290
M3.2: SHT7x, 6548336631274
midity: 12bit
mperature: 14bit
ltage: 3.0V

COM3.1;;;COM3.2;;;COM3.3;;;COM3.4;;
te;Time;Relative Humidity;Temperature;Dew Point;Relative Humidity;Temperature;Dew Point;Relative Humidity;Temperature;Dew Point;Relative Humidity;Temperature;Dew Point
15-01-30;12:26:58;36.23;21.49;5.87;34.84;21.92;5.69;;;;;
15-01-30;12:26:59;36.23;21.49;5.87;34.77;21.92;5.66;;;;;
15-01-30;12:27:00;36.23;21.49;5.87;34.71;21.93;5.64;;;;;
15-01-30;12:27:01;36.23;21.49;5.87;34.64;21.94;5.62;;;;;
15-01-30;12:27:02;36.23;21.49;5.87;34.61;21.94;5.61;;;;;
15-01-30;12:27:03;36.26;21.49;5.89;34.57;21.92;5.58;;;;;
15-01-30;12:27:04;36.26;21.49;5.89;34.57;21.92;5.58;;;;;
15-01-30;12:27:05;36.29;21.48;5.89;34.57;21.92;5.58;;;;;
15-01-30;12:27:06;36.29;21.48;5.89;34.54;21.92;5.56;;;;;
15-01-30;12:27:07;36.26;21.49;5.89;34.54;21.91;5.55;;;;;
15-01-30;12:27:08;36.23;21.49;5.87;34.50;21.91;5.54;;;;;
15-01-30;12:27:09;36.26;21.48;5.88;34.50;21.90;5.53;;;;;
15-01-30;12:27:10;36.29;21.48;5.89;34.50;21.90;5.53;;;;;
15-01-30;12:27:11;36.33;21.48;5.90;34.50;21.90;5.53;;;;;
15-01-30;12:27:12;36.33;21.49;5.91;34.54;21.90;5.54;;;;;
15-01-30;12:27:13;36.29;21.48;5.89;34.54;21.90;5.54;;;;;
15-01-30;12:27:14;36.29;21.48;5.89;34.50;21.90;5.53;;;;;
15-01-30;12:27:15;36.26;21.48;5.88;34.50;21.89;5.52;;;;;
15-01-30;12:27:16;36.26;21.48;5.88;34.50;21.89;5.52;;;;;
15-01-30;12:27:17;36.26;21.48;5.88;34.50;21.89;5.52;;;;;
15-01-30;12:27:18;36.29;21.48;5.89;34.50;21.89;5.52;;;;;
15-01-30;12:27:19;36.29;21.48;5.89;34.50;21.88;5.51;;;;;
15-01-30;12:27:20;36.36;21.48;5.92;34.50;21.87;5.50;;;;;
15-01-30;12:27:21;36.39;21.48;5.93;34.57;21.86;5.52;;;;;
15-01-30;12:27:22;36.39;21.48;5.93;34.63;21.86;5.55;;;;;
15-01-30;12:27:23;36.39;21.48;5.93;34.70;21.86;5.58;;;;;

Activate Windows
Go to PC settings to activate Windows.

test1.csv - Excel (Product Activation Failed)

FILE HOME INSERT PAGE LAYOUT FORMULAS DATA REVIEW VIEW Sign in

Paste Clipboard Font Alignment Number Styles Cells Editing

A1 : Data from EK-H4

	A	B	C	D	E	F	G	H	I	J	K	L	M
1	Data from EK-H4												
2													
3	COM3.1: S	3.6E+16											
4	COM3.2: S	6.55E+12											
5	Humidity: 12bit												
6	Temperature: 14bit												
7	Voltage: 3.0V												
8													
9	;;COM3.1;;;COM3.2;;;COM3.3;;;COM3.4;;												
10	Date;Time;Relative Humidity;Temperature;Dew Point;Relative Humidity;Temperature;Dew Point;Relative Humidity;Temperature;Dew												
11	2015-01-30;12:26:58;36.23;21.49;5.87;34.84;21.92;5.69;;;;;												
12	2015-01-30;12:26:59;36.23;21.49;5.87;34.77;21.92;5.66;;;;;												
13	2015-01-30;12:27:00;36.23;21.49;5.87;34.71;21.93;5.64;;;;;												
14	2015-01-30;12:27:01;36.23;21.49;5.87;34.64;21.94;5.62;;;;;												
15	2015-01-30;12:27:02;36.23;21.49;5.87;34.61;21.94;5.61;;;;;												
16	2015-01-30;12:27:03;36.26;21.49;5.89;34.57;21.92;5.58;;;;;												
17	2015-01-30;12:27:04;36.26;21.49;5.89;34.57;21.92;5.58;;;;;												
18	2015-01-30;12:27:05;36.29;21.48;5.89;34.57;21.92;5.58;;;;;												
19	2015-01-30;12:27:06;36.29;21.48;5.89;34.54;21.92;5.56;;;;;												
20	2015-01-30;12:27:07;36.26;21.49;5.89;34.54;21.91;5.55;;;;;												
21	2015-01-30;12:27:08;36.23;21.49;5.87;34.50;21.91;5.54;;;;;												

test1