Study of a low temperature waste heat driven sorption chiller

by

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13618

Dissertation submitted in partial fulfillment of

the requirement for the

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Universiti Teknologi PETRONAS

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CERTIFICATION OF APPROVAL

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A project dissertation submitted to the
Mechanical Engineering Programme
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(MECHANICAL)

Approved by,

____

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TRONOH, PERAK
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CERTIFICATION OF ORIGINALITY

This is to certify that I am responsible for the work submitted in this project, that the original work is my own except as specified in the references and acknowledgements, and that the original work contained herein have not been undertaken or done by unspecified sources or persons.

NUR SAKINAH KHAIRUL SALLEH

ABSTRACT

With deeper concern regarding global warming problem, a growing interest has been given to adsorption chiller. It acts as a replacement to conventional air-conditioning unit that occupy high usage of electricity from fossil fuelled power plant. In order to make sure that the chiller is functional for all-year-round, the heat source need to be lowered down to compensate for the varying rainy and hot seasons in Malaysia. Therefore multi-staging the chiller is required so that the chiller is able to operate in a low temperature heat source. This paper aims to study the effects of two stages four-bed adsorption chiller of silica gel-water pair based on the cooling capacity, coefficient of performance (COP), cycle time, and temperature profiles. It is highlighted that the adsorbent-refrigerant pair used in this system is of silica gel and water for numerous advantageous reasons. Mathematical equation of the model comprises of adsorption isotherms, adsorption kinetics, isosteric heat of adsorption, energy balance for condenser, evaporator and the system beds is analysed respectively. Simulation method is applied to solve the problem using MATLAB programming in order to achieve the performance of the two-stage four-bed chiller. Results indicated that the system is able to operate below 60°C. With heat source temperature of 55°C, cooling water of 30°C and chilled water running at 14°C, the COP of the system achieved is around 0.24.

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NOMENCLATURE

Symbols	
A	area, (m^{2})
A_0	coefficient in Eq. (1), kg (kg of dry adsorbent)$^{-1}$
A_1	coefficient in Eq. (1), kg (kg of dry adsorbent, K)$^{-1}$
A_2	coefficient in Eq. (1), kg (kg of dry adsorbent K^{2})$^{-1}$
A_3	coefficient in Eq. (1), kg (kg of dry adsorbent K^{3})$^{-1}$
B_0	coefficient in Eq. (1), (-)
B_1	coefficient in Eq. (1), K^{-1}
B_2	coefficient in Eq. (1), K^{-2}
B_3	coefficient in Eq. (1), K^{-3}
C, C_p	specific heat, kJ kg^{-1} K^{-1}
COP	coefficient of performance
D_s	surface diffusion coefficient, m^{2} s^{-1}
D_{so}	pre-exponential constant, m^{2} s^{-1}
E_a	activation energy, J kg^{-1}
$k_s a_v$	mass transfer coefficient, s^{-1}
$\dot{m}$	mass flow rate, kg s^{-1}
L	latent heat of vaporization, kJ kg^{-1}
P	pressure (Pa)
q^*	equilibrium uptake, kg kg^{-1}
q	refrigerant amount adsorbed, kg kg^{-1}
Q_{st}	heat of adsorption, kJ kg^{-1}
R	gas constant, J kg^{-1} K^{-1}
R_p	average radius of adsorbent particle, m
T	temperature, K
t	time, s
U	heat transfer coefficient, kJ m^{2} K^{-1}
W	mass, kg

ABBREVIATIONS

Subscript	
ads	Adsorption
b	bed
c	condenser
chill	chilled water
conw	condensed water inside condenser
cw	cooling water
des	desorber
e	evaporator
ew	water inside evaporator
hex	heat exchanger
in	inlet
out	outlet
ref	refrigerant
sat	saturation
sg	silica gel
v	vapor
w	water

CHAPTER 1 : INTRODUCTION

This chapter shows the general descriptions that lays the broad foundation for the problem statement that directed the interest to study in this research area. The objective and scope of study is also identified to narrow down the search field.

1.1 Background Study

1.1.1 Adsorption Chiller System

Refrigeration process has been garnering a lot of attention over the years as it has numerous application in human life, such as in food processing sector and air-conditioning field [1]. However, the process consumes a lot of energy and the conventional vapor compression refrigeration cycles is not sustainable to the environment. Therefore, sustainable energy concept is taken into approach by constructing a chiller system that utilizes waste heat from industialization. Industrial facilities, power stations and biogas plants all make use of processes in which heat is essentially a waste product.

As of now, hardly any of this heat energy is put to use. Thermally driven cooling systems are one possible alternative to traditional air conditioning. These systems use the evaporation of fluids such as water at low pressure to remove heat from the environment – an energy-efficient cooling method [3]. Thermally activated cooling involves harnessing waste heat and using it for cooling applications. The low grade heat source can also be garnerred through solar means, thus making it into a solar thermal-driven sorption systems.

The term sorption can be divided into two; either absorption or adsorption. Adsorption is the term used for the adhesion process of atoms, ions, or molecules to a solid surface or film [3]. Practically this process causes accumulation of the adsorbate on the surface of adsorbent. It is categorized as an exothermic process as energy in the form of heat is released when molecules is adsorbed on the particular surface. Intriguingly, this process is reversible in the manner that as it can adsorb on the film of adsorbate, it also can be desorb from the surface film [3]. For this to be achieved, heat is required for the process of desorption to take place. The reversibility of this equation can be shown as below.

$$\text{Adsorbent} + \text{Adsorbate} \underset{\text{Heat gained}}{\overset{\text{Heat released}}{\rightleftharpoons}} \text{Adsorption}$$

It is important to note that adsorption is different from absorption, as the molecules undergoing absorption are taken up by the volume, not by the surface. The term sorption is a general expression used to incorporate both processes. The process occurrence can be related to the Van der Waals forces and electrostatic forces between molecules of adsorbent and adsorbate [1,3]. The phenomenon that occurs between adsorbent and refrigerant pairs is applied in nowadays adsorption chiller equipment.

This system is almost similar to a conventional air conditioner or refrigerator, only that the mechanical compressor driven electrically is substituted with thermal operated adsorption compressor. Generally the chiller uses fixed adsorbent beds which brings great advantages as no moving parts is necessary in the system, thus reducing power consumption. It also bring simplicity to its mechanical compartment, besides having a better lifetime and high reliability [1].

Although the system is a favourable option for an electric energy savers, it is undeniable that there are some drawbacks to the system as there is an intermittent cycle operation between the adsorbent beds, adsorption and desorption process will take place alternately on the beds. This decreases the COP of the system [1]. There are several factors affecting adsorption process [3].

They are the nature of adsorbate and adsorbent; surface area of adsorbent; adsorbent activation; and the experimental parameters such as temperature and pressure. Surface properties plays an important role in the process of adsorption. An adsorbent that has large specific surface area allows a higher adsorption capacity, which is preferable. A great number of small sized pores existed between the adsorption surfaces. The pore size distribution of these micropores are also substantial to determine the accessibility of adsorbate molecules to the internal adsorption surface. Certain adsorbate such as zeolite can be manufactured or specifically altered with precise pore size distributions. The process of adsorption cycle in refrigeration is discussed in detail in Chapter Two (2).

1.1.2 Adsorbent

The term adsorbent is used to represent the separating agent to express the difference between molecules in a mixture: adsorption equilibrium or kinetics [3]. Some similar characteristics can be found in adsorbent as they are commonly manufactured in the shapes of spherical pellets, rods, or moldings. Industrial adsorbents can be categorized in three classes, oxygen, carbon or polymer-based compounds [3].

For oxygen-containing compounds, they are normally hydrophilic and polar, such as silica gel and zeolites. In contrast, a carbon-based compounds have the characteristics that are commonly hydrophobic and non-polar, including materials such as activated carbon and graphite. On the other hand, polymer-based compounds can either be polar or non-polar functional groups in a porous polymer matrix [3].

Adsorbents like activated carbon, silica gel and zeolite are among the common used in the industry [1]. In terms of selecting a good adsorbent, some properties that need to be considered are having high abrasion resistance, high thermal stability and small pore diameters [3]. This are all the factors that can contribute to high adsorption capacity. The heat and mass transfer characteristics of the adsorbent beds also plays a leading role in giving a higher COP to the adsorption cycle [1].

Having a consolidated adsorbent is beneficial in improving the properties of heat and mass transfer. In this dissertation, silica gel is chosen as the adsorbent of the sorption chiller system to simulate the effects at the end of this study. Some characteristics of adsorption of silica gel will be discussed later in Chapter Two (2).

1.1.3 Refrigerant

A refrigerant is a substance, usually a fluid, used in a heat pump and refrigeration cycle [3]. Usually, it changes state from liquid to gas and inversely as it undergoes through the cycle. Refrigerants or adsorbate are widely used in air-conditioning systems for buildings, facilities, and industries such as chemical, pharmaceutical, automotive, and food production and storage.

In 20th century, ecological concerns developed on the broad usage of cholofluorocarbons (CFCs) and hydrochlorofluorocarbon (HCFCs) as a refrigerant [2]. These type of refrigerant causes severe environmental impact when it is release to the atmosphere as it depletes the ozone layer and adding into the greenhouse gasses effect [1,2].

In an adsorption refrigeration systems, water, methanol (ethanol) or ammonia are widely used as refrigerants [1]. Choosing an ideal refrigerant depends on several factors such as having a good thermodynamic properties, is noncorrosive to mechanical components, safe and free from toxicity and flammability [3]. The desired thermodynamic properties are a boiling point somewhat below the target temperature, a high heat of vaporization, a moderate density in liquid form, a relatively high density in gaseous form, and a high critical temperature. In this study, water is used as a refrigerant as it is non toxic, low cost, and widely available. Moreover, it is widely known to be useful as a coolant. The advantages of water as a refrigerant is further discussed in Chapter Three (3).

1.2 Problem Statement

There has been numerous researches based on the concept of adsorption chiller; however there is a limitation towards the range of temperature at which the chiller can operate. Taking into concern that solar power is to be used in generating the heat energy to the model, it is important to consider on the availability of the power generated throughout the year. Malaysia, which experiences monsoon season during the month of October until December; will undergo insufficient supply of driving heat source from the solar system during the said month as the Sun availability is rather low. It is expected that the heat source temperature will be below than 60°C.

Therefore, the idea of having a two-stage four-bed system rather than that of one-stage is to lower down the temperature required for hot water running in one of the bed to around 40°C -100°C. Usually for one-stage model, the operated temperature for running hot water ranges from 60°C -100°C. This will be beneficial for using the adsorption chiller throughout the whole year.

In this paperwork, silica gel-water will be used with the intention to evaluate the performance of the chiller model based on the cooling capacity, coefficient of performance (COP), cycle time, and temperature profiles. It is important to highlight that the model is for two-stages of four-bed system adsorption chiller. Thus, to develop the model from two-bed system into multiple-bed system requires deep understanding on the equations and parameters of the concept being used.

1.3 Objective(s)

To evaluate the performance of two stages four-bed adsorption chiller of silica gel-water pair based on the cooling capacity, coefficient of performance (COP), cycle time, and temperature profiles.

1.4 Scope of Study

This project focuses on the modelling of four-bed of two stages silica gel-water adsorption chiller. It focuses on acquiring waste heat from low temperature ranges (below 100°C) and covert them to become a useful sorption chiller. In this model, the sorption chiller is to function at sub-atmospheric pressure whereby silica gel-water is favours as the adsorbent-adsorbate pairs. The performance evaluation of the model will focus on coefficient of performance (COP), cooling capacity, cycle time, and temperature profiles. Mathematical equation of the model comprises of adsorption isotherms, adsorption kinetics, isosteric heat of adsorption, energy balance for condenser, evaporator and the system beds will be analysed and the model will be illustrated through MATLAB.

CHAPTER 2 : LITERATURE REVIEW

This chapter reviews some of the recent research works either experimentally or simulation analysis done on the adsorption chiller by numerous scholars. Firstly, the operation of a simple two-bed adsorption chiller is analysed to understand the principle of this refrigeration model. Various model of advanced chiller system that consists of more than two-bed is then examined comprehensively. Some study on the characteristics of adsorption using silica gel and water pair is also made to outline the advantages of the pair as adsorbent-refrigerant.

2.1 Adsorption Characteristics

Numerous studies had been conducted regarding the adsorption chiller performance and some are using different adsorbent-adsorbate pairs for example Activated Carbon-Hydrofluoro Olefin [14] and activated carbon Maxsorb III-R134a [15]. In this framework, silica gel-water pair is favourable due to several reasons. Conventionally, silica gel is found in the packing containers of clothing, food, medicines, and others. It is used widely to adsorb humidity inside the packing box or case to make sure the item is not defected when stored.

Silica gel have a distinctive internal structure whereby it is of a porous and amorphous form of silica (SiO_2) [12]. It has microscopic pores that is interconnected to form large network of porous system. Usually the diameter sizes of the pores varies between 5 Å and 3000 Å. This unique features of the material will be the key parameters to adsorb the large water content in the adsorption chillers. Silica gel possess a great latent heat of evaporation whereby 40% of its dry mass is able to hold the water content [5]. Large surface area of the silica gel will aid in uptake capacity of water vapour in the cycle. This consequently will enhance the cycle performance of the chiller.

Thu et. al. have made a significant study on the thermo-physical properties, surface characteristics and water vapour uptake capacity of silica gel [16]. The silica gel type specifically chosen in this study are of Type-RD 2560, Type-A5BW and Type-A++. In terms of surface characteristics, Type-A++ silica gel has been observed to show the largest surface area among the three which is approximately 863.6m^3/g.

Besides, micropore analysis of the structure had also been conducted and the results showed that type A++ has the largest external surface area of the micropore. In addition, the authors also determined the Pore Size Distribution (PSD), whereby it is applied to identify the pores population as a function of the pore width. The highest pore volume again can be found in Type-A++ silica gel. From these characteristics, it is found that Type-A++ silica gel displays the biggest equilibrium uptake capacity of around 537 cm^3/g especially at high relative pressure. In total, Type-A++ has better properties to be used as adsorbent as it has bigger surface area with more pore volume and thus enhances the water vapour uptake capacity.

On the contrary, the experimental study on thermo-physical properties of two Fuji Davidson type silica gels, namely type A and RD, showed a different results [17]. They showed that the monolayer capacities for type A and RD silica gels when in contact with water vapour is 0.4 and 0.45 $kg \cdot kg^{-1}$, respectively. In addition, the authors mentioned that between the two types, type RD is commonly used as chiller adsorbent in the industry.

Nevertheless, both of this study demonstrates that the sorption characteristics of the adsorbent plays an important role in determining the cycle performance. A larger specific surface area will be preferable as for adsorbent so that it holds more water vapour content in one cycle. It is important to choose the appropriate type of adsorbent-adsorbate pair to ensure that the cycle of the chiller is utilised to the maximum potential. On top of that, adsorption of water can occur without causing a structural change or volume expansion on the silica gel.

Moreover, another additional features of the adsorbent used is that it has low regeneration temperature. Regeneration temperature is the temperature required to activate the process of desorption of water vapour from the silica gel. In order to remove water vapour content from the surface area of silica gels, heat is supplied to stimulate the motion. In short, it is an endothermic process. The regeneration temperature for silica gel is relatively low (below 100°C) and averagely around 85°C [5]. Therefore, waste heat from the industry will be practical to operate the chiller. In this paperwork, the heat source generated will be focused on the solar power system and it has to be taken into account that the amount of power generated will be less than that of from electrical power plant that burned fossil fuels. Thus, having heat supplied of below 100 °C is actually a requirement, not an option.

Furthermore solar power depends on the weather of the country. In Malaysia specifically, it has a hot and humid climate as the country is located at the equatorial part of Earth. Thus, generating solar power will be practicable before the monsoon season strikes. In order to achieve an operating adsorption chiller that is functional for all year round is by lowering down the regeneration temperature of the system. This will be achievable by having a two-stage adsorption chiller.

On top of that, one can notice the behaviour of the regeneration temperature of the adsorption chiller of activated carbon Maxsorb III and refrigerant R134a pair studied by Loh et. al [4]. In terms of coefficient of performance (COP) of the system, asymptote graph is obtain as the regeneration temperature increases. This explains that there is a limiting behaviour on this relationship whereby COP increases greatly in the early increment of temperature until the differential gradient decreases and becomes uniform.

In contrast, a linear growth can be seen on the specific cooling effect (SCE) as the regeneration temperature rises. However, the value is almost zero when it is below 55 °C, which means that the cooling effect only becomes significant above the temperature. Loh et. al. clarify that by having multi-stage cycle, the regeneration temperature can be further reduced than the current state. A similar result of the graph is also obtained by Rezk and Al-Dadah [18].

Apart from that, another additional factors of choosing the silica gel as adsorbent-adsorbate pair is that because it is environmentally friendly. This thermally driven adsorption chiller which uses water as the adsorbate has zero ozone depletion potential (ODP). In the industry, R134a is the most commonly used refrigerant unit in cooling system. However it possess global warming potential (GWP) of around 1300, which is considerably high [19].

In contrast, water is used to replace the harmful refrigerant as it has no GWP. It is characterized as being natural, non-flammable and non-toxic refrigerant which is beneficial in the system. Moreover, water source is abundant and available almost everywhere especially in Malaysia.

2.2 Multi-stage Adsorption Chiller System

Numerous studies has been done to develop a higher performance of an adsorption chiller [7,8,9]. Improving COP value of the system has been shown by Liu et. al [7] whereby an experimental performance of an adsorption chiller which has no refrigerant valve is conducted. The purpose is to overcome the mass transfer resistance problem in a conventional system. The results obtained shows 34.4% of COP increment through heat recovery process. On the other hand, 18.3% of COP increases through the mass recovery method and the cooling power also rises by 13.7%.

Apart from that, Xia et. al [8] developed an improved prototype of conventional two-bed system adsorption chiller that utilizes only one vacuum valve. The performance positively shows that the COP value improves by 20% compared to the former model. Recently, Wang and Lu [9] demonstrated that heat recovery method can enhance COP significantly whereas the mass recovery can increase adsorption capacity of the refrigeration model.

Other than that, Saha et. al [10] introduced a chiller system which encompasses a multi-scheme bed. In this system, three-bed cycle is used with the expectation of that this system is able to perform as efficient as a single-stage adsorption chiller. The performance is evaluated through a cycle simulation computer program. . For this case, it can be observed that the waste heat recovery efficiency is about 35% higher than that of the two-bed system, and the COP value is 0.38 with a driving source temperature at 80 °C and the coolant inlet and chilled water inlet temperatures at 30 and 14 °C, respectively. However, the driving heat source for this model is still relatively high and does not meet the requirement of this project.

These researches mentioned as above highlights on the improvement made on the cooling capacity and COP of the system. However, utilizing a relatively low heat temperature source (below 60°C) is also an important element in developing the system. Therefore, staging the cycle will be necessary as to lower down the temperature required as for the heat source.

For instance, multi-stage system was designed such as the one studied by Hamamoto et. al [11] in a two-stage adsorption refrigeration chiller prototype. The result shown that the heat source temperature for the chiller is in the range of 53 to 61°C, with the evaporator and condenser temperatures at 7 and 30 °C, respectively. It has been demonstrated that by having an optimum adsorbent mass distribution to the lower sorption elements compared to the upper sorption elements, the cooling capacity shows a significant increase especially when the temperature source is higher. Nonetheless, COP increment is said to be less significant for this model.

In addition, Saha et. al. [12] also show a similar effort by developing a prototype of two stages four-bed refrigeration cycle for silica gel–water pair. The experimental performance reveals that by exploiting 55°C driving source temperature, the cycle can be successfully operated provided that 30°C cooling temperature is used. This prototype is suitable for tropical climate condition as the solar collector used is of flat plate type.

Another related work was done by Habib et. al. [13] on a dual-mode adsorption chiller. Intriguingly, this model can be operated as a single stage four-bed system for temperature above 60°C and two stages four-bed system for temperature below 60°C. This simulation study make use of a solar thermal data collected at Durgapur, India (23.48 °N, 87.32°E). Based on the climate of the coordinate, results obtained denote that the system is able to function effectively throughout the year.

These studies demonstrates that having a multi-stages system allows the driving heat source to be exploited at a relatively lower temperature. Hence, the concern of having a fluctuating weather condition in Malaysia can be eliminated once the system is able to operate using a low-grade heat source.

2.3 Working Principle: Two-Stage Four-bed Chiller System

This project focuses on building a two-stage four-bed system of refrigeration model in order to utilize a driving heat source of below 60°C which could not be attained using single-stage operation [12,13]. Based on Dühring diagram in Figure 5, the single stage silica gel–water cycle will not be functional with a 50°C driving source temperature if the cooling source is at 30°C or higher [12].

As the system is circulated, pressure of the refrigerant or water will increase from one evaporating pressure to condensing pressure. This will give lift on the regeneration temperature ($T_{des} - T_{cond}$) of the adsorbent beds. However, staging the system allows the pressure to be divided into two gradual phase and thus the regeneration temperature lift required on the beds can also be distributed into two smaller parts [12,13].

Figure 2.3.2 illustrates the schematic diagram of the system. An advanced two-stage four-bed chiller system comprises of four bed chambers that act as its sorption elements, namely SE1, SE2, SE3 and SE4. An evaporator and condenser are also applied like in a simple two-bed system. In total there are six elements in the system that acts as a heat exchangers.

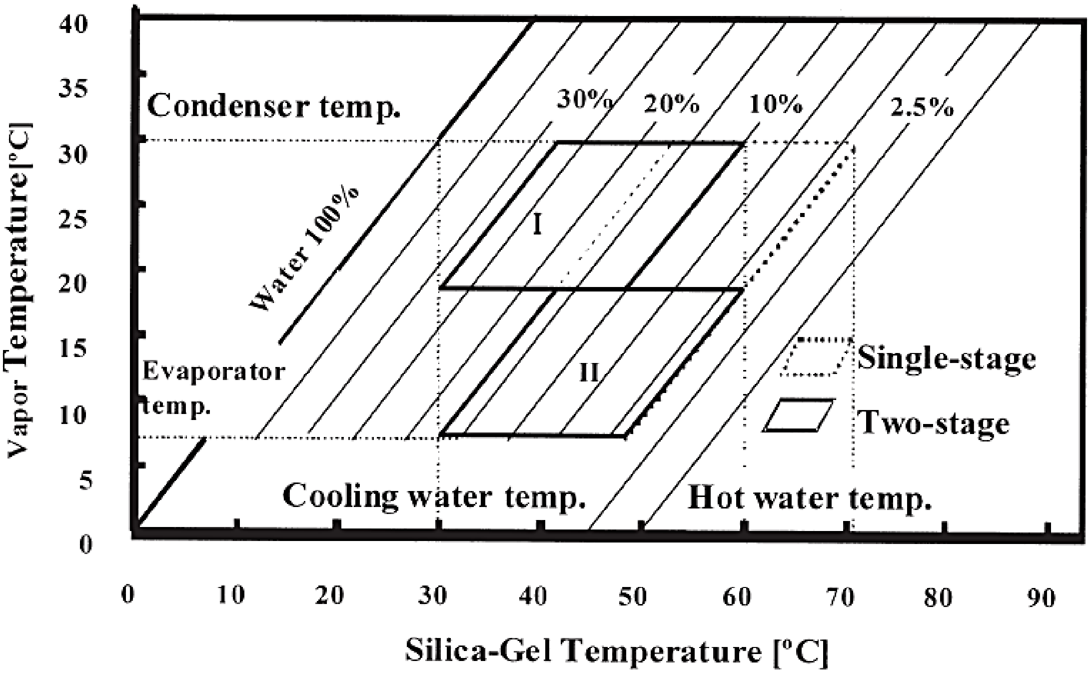


Figure 2.1 Conceptual P-T-X diagram both the conventional and two-stage cycles. [12]

For this system, the working operation is designed according to numerous study of a similar principle made by other scholars [12,13]. The operation for this system cycle can be configured into four modes, specifically A, B, C and D. In mode A, three valves namely valve 1, 3, and 5 are opened to let the water or refrigerant flow between the heat exchangers. SE1 and SE4 are circulated with hot water and acts as desorbing beds.

On the other hand, cool water are allow to flow in the SE 2 and SE3, thus acting them as the adsorbing beds. Evaporation of the refrigerant or water takes place in the evaporator and latent heat is taken up from the circulated chilled water. Opening valve 3 connects SE2 to the evaporator, therefore enabling the silica gel to adsorb the vapor refrigerant.

In SE4, hot water that flows causes the refrigerant to be desorbed by the silica gel and it will be channelled to SE3 via valve 5. This adsorption process will then takes place as cooled water is flowing through the bed. On the other hand, refrigerant at SE1 that experiences desorption will be directed to the condenser through valve 1, whereby the condensation of water will takes place at condensing temperature, T_{cond}.

The condensation heat Q_{cond} is removed by allowing cooling water to flow through the condenser. The condensed water is then circulated to the evaporator through a connecting tube and completes the cycle. The connecting tube is bent to ensure a pressure drop is achieved so that the refrigerant is in liquid phase as it enters the evaporator [12].

The flows of hot and cooling water is redirected between the operating beds as the concentration of the refrigerant in the adsorbed and desorbed bed has reach their equilibrium level. The valves are switched to enable the adsorbers to change into desorption modes and the desorbers to shift into adsorption modes (Mode C) as shown in Figure 2.3.3.

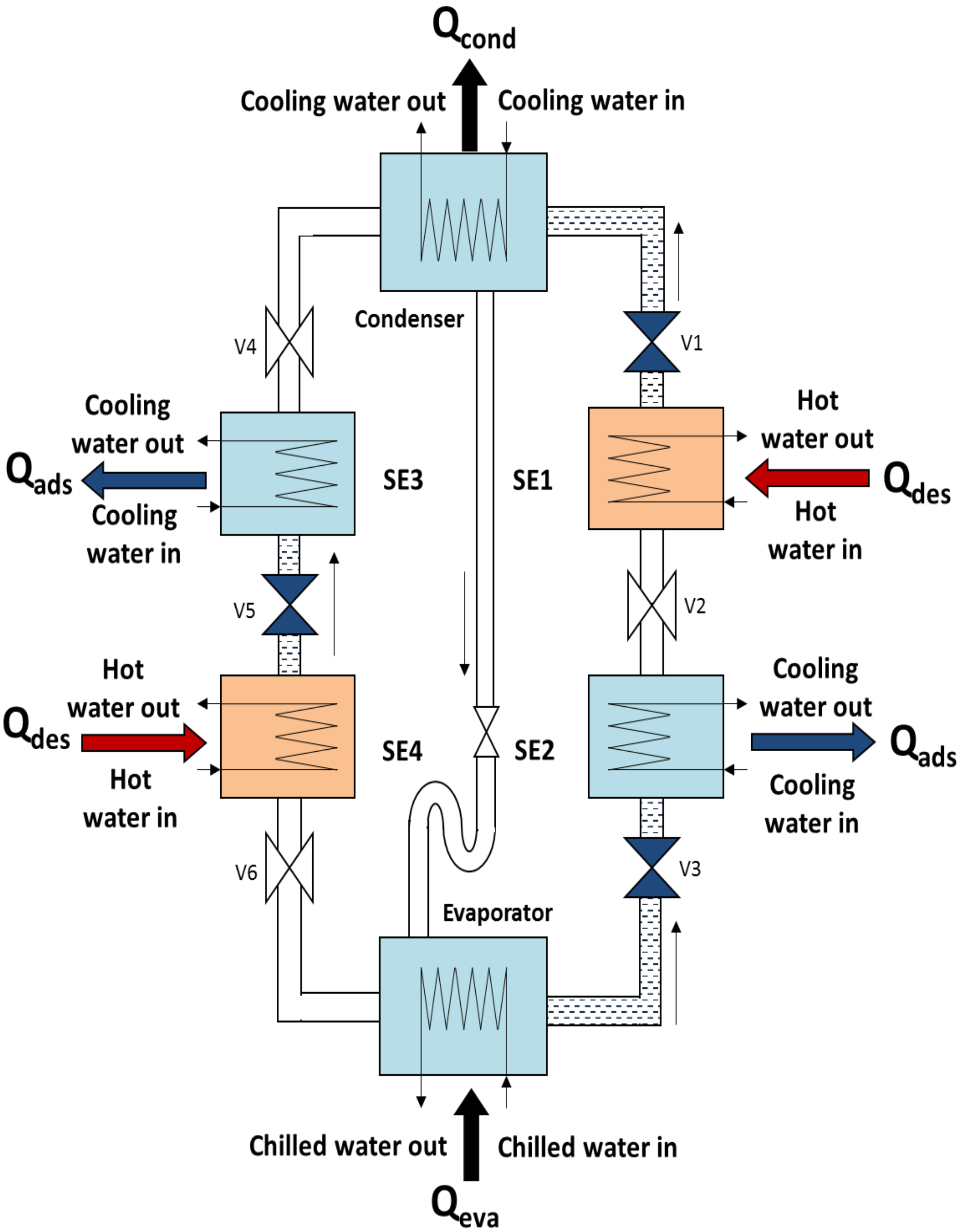


Figure 2.2 Two-stage Four-bed adsorption chiller system (Mode A).

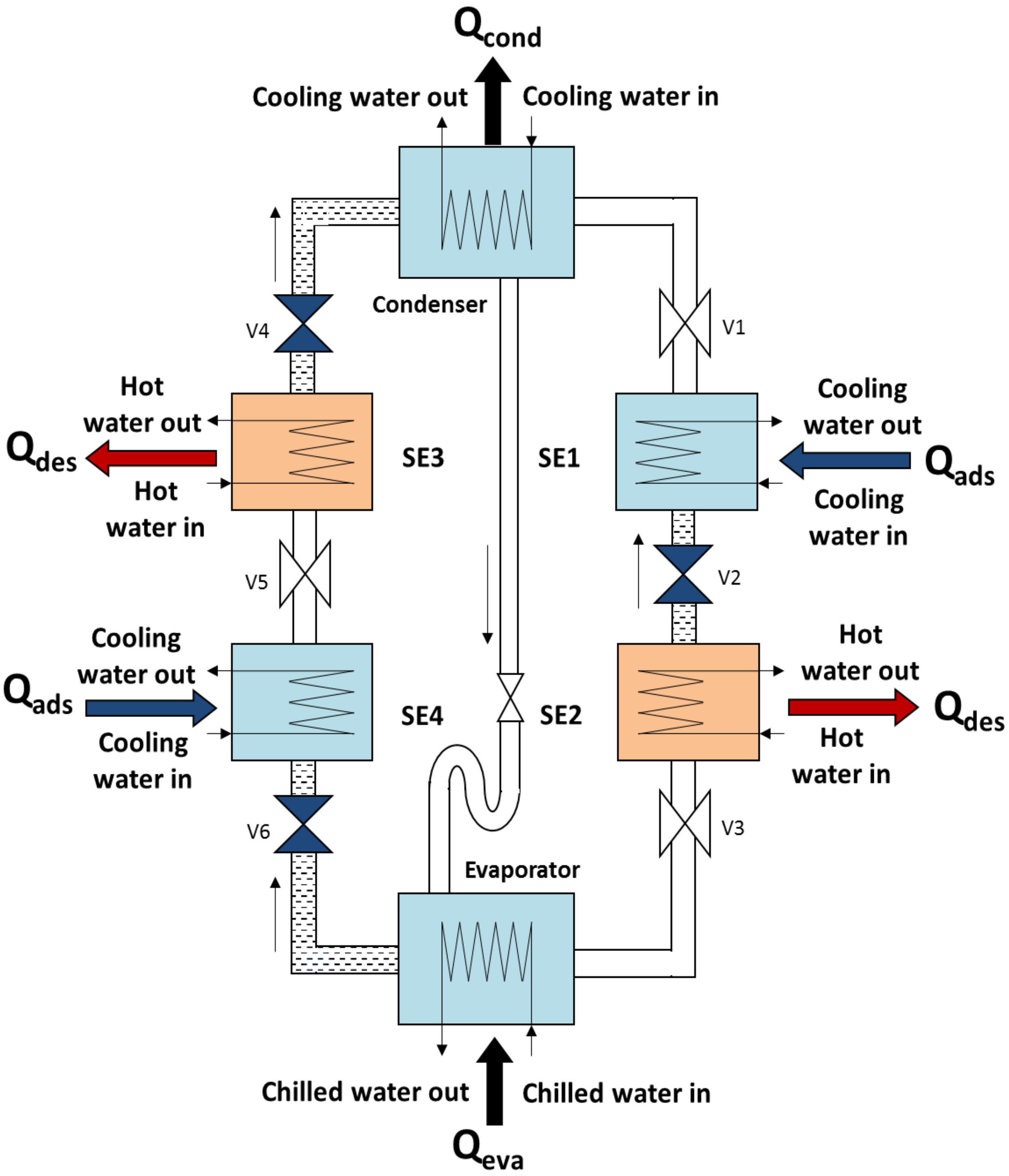


Figure 2.3 Two-stage Four-bed adsorption chiller system (Mode C).

The process will then continue accordingly. Mode B and D acts as an intermediate to allow short period of time to takes place so as to preheat the adsorbers and precool the desorbers, before the flow reversal follows. No adsorption or desorption process occurs during this interval. The operation schedule of the chiller system is shown in Table 2.3.1.

Table 2.1 Operation schedule of the chiller system.

Cycle (Duration)	**Mode A (Desorption)**	**Mode B (Pre-heating/ Pre-cooling)**	**Mode C (Adsorption)**	**Mode D (Pre-heating/ Pre-cooling)**
Valves 1, 3, 5	Open	Closed	Closed	Closed
Valves 2,4, 6	Closed	Closed	Open	Closed
SE 1, SE 4	Hot water	Cooling water	Cooling water	Hot water
SE 2, SE 3	Cooling water	Hot water	Hot water	Cooling water

CHAPTER 3 : PROJECT WORK

3.1 Project Methodology

In order to address the problem in this project, a sequence of tasks is developed to tackle the issue consecutively. This project is divided into two phase; Final Year Project (FYP) I and II. The method of performing this project is illustrated as in Figure 3.1.

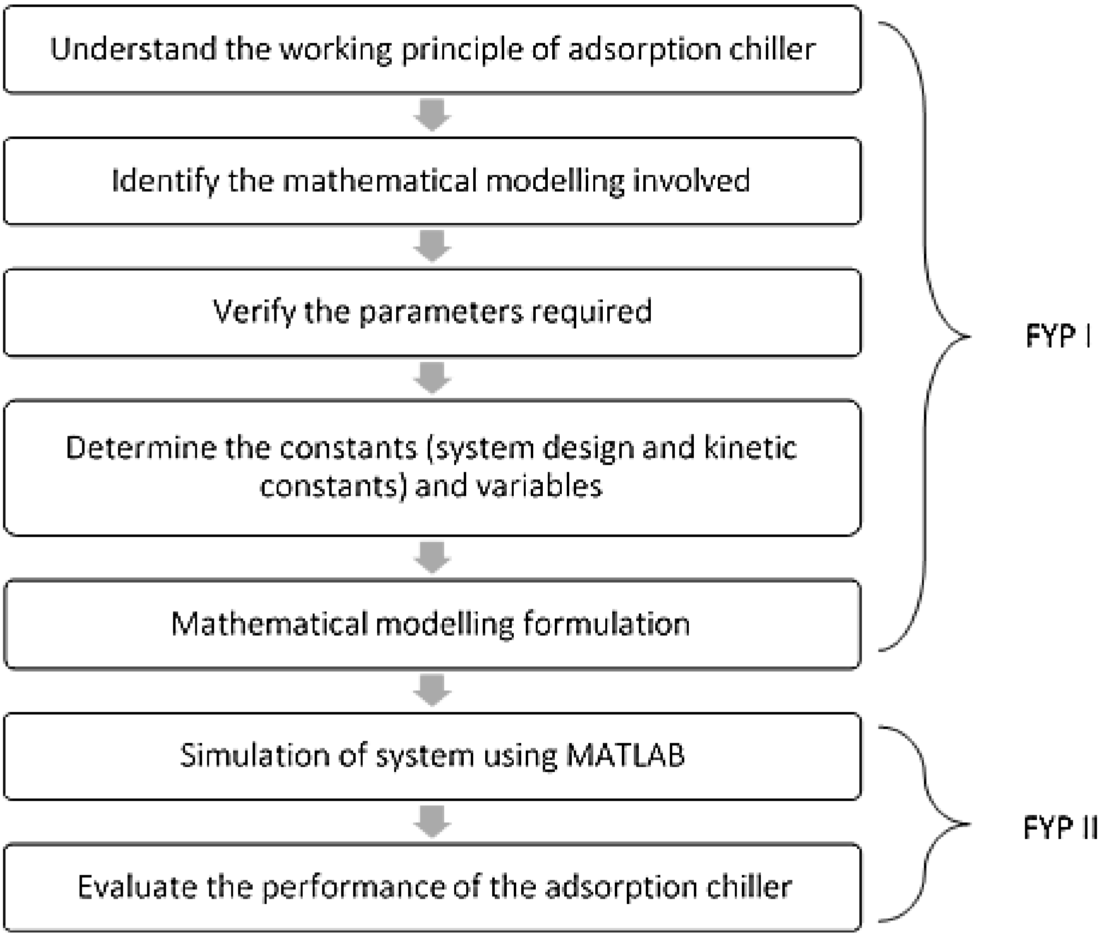
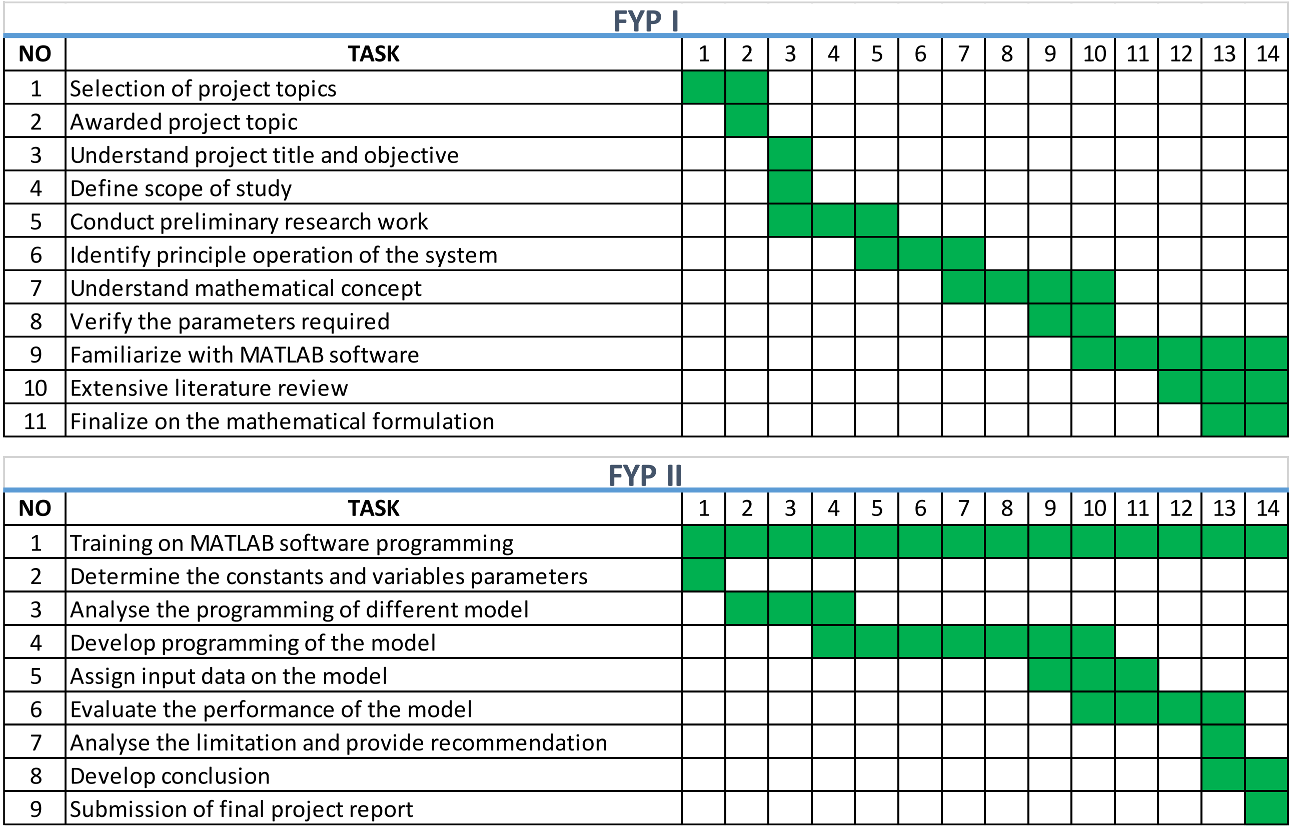


Figure 3.1 Methodology of the project.

3.2 Project Gantt chart

A Gantt chart is developed to manage the progression of the project. With this visual representation, it is easier to keep on track of the starting and finishing timeline of different specific tasks. Figure 3.2 shows the Gantt chart of this project based on the weeks of the project timeline. The total week of project completion is 28 weeks, each having 14 weeks allocated for FYP I and FYP II.



FYP I

NO	TASK	1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	Selection of project topics	🟩	🟩												
2	Awarded project topic		🟩												
3	Understand project title and objective			🟩											
4	Define scope of study			🟩											
5	Conduct preliminary research work			🟩	🟩	🟩									
6	Identify principle operation of the system					🟩	🟩	🟩							
7	Understand mathematical concept							🟩	🟩	🟩	🟩				
8	Verify the parameters required									🟩	🟩				
9	Familiarize with MATLAB software										🟩	🟩	🟩	🟩	🟩
10	Extensive literature review												🟩	🟩	🟩
11	Finalize on the mathematical formulation													🟩	🟩

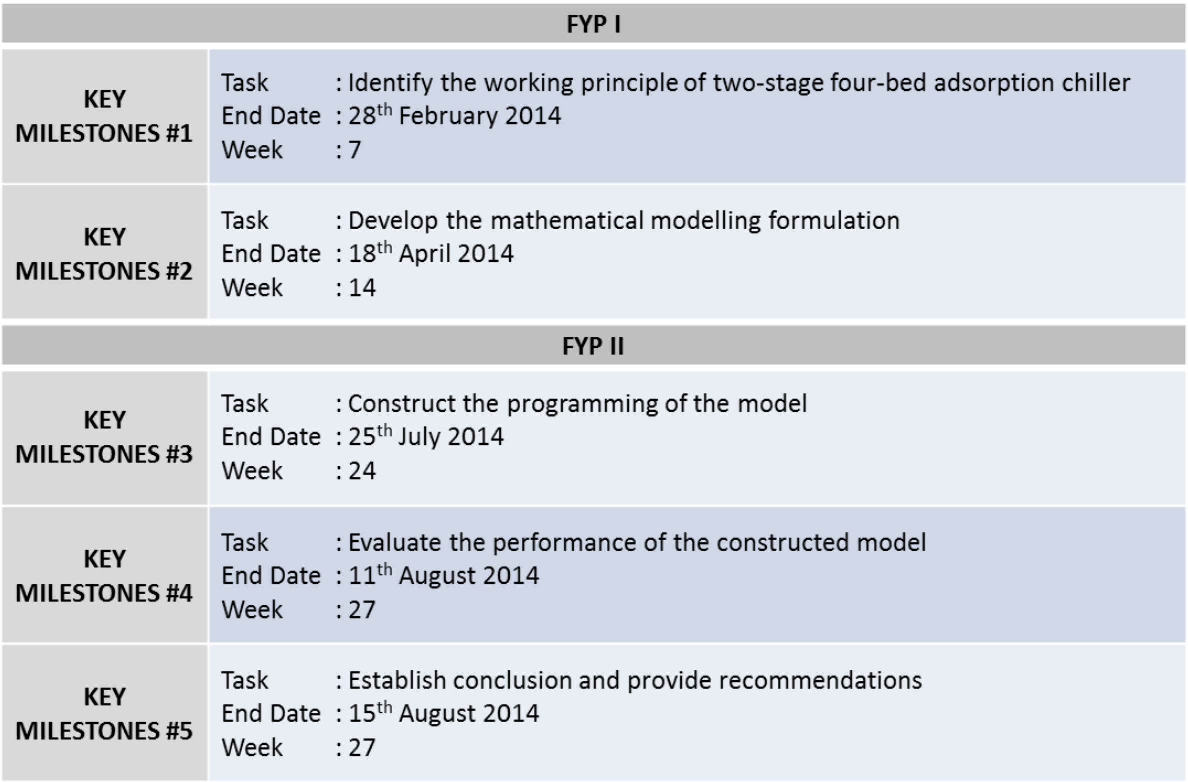
FYP II

NO	TASK	1	2	3	4	5	6	7	8	9	10	11	12	13	14
1	Training on MATLAB software programming	🟩	🟩	🟩	🟩	🟩	🟩	🟩	🟩	🟩	🟩	🟩	🟩	🟩	🟩
2	Determine the constants and variables parameters	🟩													
3	Analyse the programming of different model		🟩	🟩	🟩										
4	Develop programming of the model				🟩	🟩	🟩	🟩	🟩	🟩	🟩				
5	Assign input data on the model									🟩	🟩	🟩			
6	Evaluate the performance of the model										🟩	🟩	🟩	🟩	
7	Analyse the limitation and provide recommendation													🟩	
8	Develop conclusion													🟩	🟩
9	Submission of final project report														🟩

Figure 3.2 Project Gantt chart for FYP I and II.

3.3 Project Key Milestones

Based on the Gantt chart, some of the project key milestones are outlined to make sure the timescale-based tasks are completed accordingly. Moreover it allows one to measure the progressiveness of the project by achieving these distributed, specific tasks sequentially. The key milestones of this project can be seen as in Figure 3.3.



FYP I	
KEY MILESTONES #1	Task : Identify the working principle of two-stage four-bed adsorption chiller End Date : 28th February 2014 Week : 7
KEY MILESTONES #2	Task : Develop the mathematical modelling formulation End Date : 18th April 2014 Week : 14
FYP II	
KEY MILESTONES #3	Task : Construct the programming of the model End Date : 25th July 2014 Week : 24
KEY MILESTONES #4	Task : Evaluate the performance of the constructed model End Date : 11th August 2014 Week : 27
KEY MILESTONES #5	Task : Establish conclusion and provide recommendations End Date : 15th August 2014 Week : 27

Figure 3.3 Key milestones of the project.

3.4 Mathematical Modelling

This sub-chapter is an extended literature review that analyses the operation of the two stages four-bed refrigeration system as to gain deeper understanding on the system that will be designed and simulated. The mathematical equation that involves in the system is also illustrated in this section.

3.4.1 Adsorption Isotherm

Based on some studies reviewed [11,13], the model is developed mathematically. The adsorption isotherms which represents the equilibrium uptake of the silica gel-water pair is expressed in equation below.

$$q^* = A(T_{sg}) \left[\frac{P_{sat}(T_{ref})}{P_{sat}(T_{sg})}\right]^{B(T_{sg})} \tag{3.1}$$

where $A(T_{sg}) = A_0 + A_1 T_{sg} + A_2 T_{sg}^2 + A_3 T_{sg}^2$,

and $B(T_{sg}) = B_0 + B_1 T_{sg} + B_2 T_{sg}^2 + B_3 T_{sg}^2$

Based on equation 1, $P_{sat}(T_{ref})$ and $P_{sat}(T_{sg})$ are the saturation vapor pressure at temperatures T_{ref} (refrigerant or water vapor) and T_{sg} (silica gel), respectively. The parameters $A(T_{sg})$ and $B(T_{sg})$ are determined by the least-square fits of manufacturer's experimental data which is applicable in this study. The constants used in the equation are shown as in Table 3.4.

The refrigerant temperature in the vapor phase is defined by the temperatures of the evaporator (during adsorption) or condenser (in desorption). On the other hand, the adsorbent temperature is determined by the heat transfer fluid (hot or cooling water) temperature. Only for refrigerants in the adsorbed phase are the temperatures of refrigerant and adsorbent taken as equivalent.

The saturation vapor pressure and temperature are correlated by Antoine's equation [22], which can be written as follows:

$$P_s = 133.32 \times \exp 18.3 - \left(\frac{3820}{T - 46.1}\right) \tag{3.2}$$

From this adsorption isotherm, the rate of adsorption and desorption can be modelled through a linear driving force kinetic equation as shown in Eq. (3.2). The coefficients are established by Chihara and Suzuki [20].

$$\frac{dq}{dt} = k_s a_v (q^* - q) \tag{3.3}$$

where $k_s a_v$ is the term used to represent the overall mass transfer coefficient for adsorption [20] and is given by:

$$k_s a_v = 15 \frac{D_s}{R_p^2} \tag{3.4}$$

The rate of adsorption is considered to be varied by the surface diffusion inside a gel particle and surface diffusivity (D_s) [20]. This term is expressed as a function of temperature by:

$$D_s = D_{s0} \exp(-\frac{E_a}{RT}) \tag{3.5}$$

3.4.2 Energy balance of the bed

Energy balance equation for the sorption bed can be expressed as in Eq. (6) with the assumption of having uniform temperature, pressure and refrigerant concentration throughout the bed [11].

$$T_{out} = T_b + (T_{in} - T_b) \exp\left(-\frac{U_b A_b}{\dot{m}_w C_w}\right) \tag{3.6}$$

$$\begin{aligned} \frac{d}{dt}\{(W_{sg}C_{sg} + W_{sg}C_{p,w}q + W_{hex}C_{hex})T_b\} \\ = W_{sg}Q_{st}\frac{dq}{dt} - \delta W_{sg}C_v \left\{\gamma(T_b - T_e)\frac{dq}{dt} + (1-\gamma)(T_b - T_v)\frac{dq}{dt}\right\}T_b \\ + \dot{m}_w(T_{in} - T_{out}) \end{aligned} \tag{3.7}$$

The δ value vary from 0 or 1 and is governed by the adsorber or desorbed function of the bed. Value of γ is also either 0 or 1 and depends on the interaction of the bed; either on the evaporator or another bed. Eq. (3.7) is applicable for both condition of adsorption and desorption energy balances of the bed. In this equation, during adsorption the heat transfer fluid (water) temperature terms T_{in} and Tout is governed by the cooling water whereas during desorption, it is indicated by the hot water. T_b refers to the temperature of the bed.

The left hand side of the adsorber/desorber energy balance equations Eq. (3.7) presents the amount of sensible heat needed to cool or heat the silica-gel (*sg*), the water (*w*) contents in bed as well as metallic (*hex*) parts of the heat exchanger during adsorption or desorption. This term shows the input/output of sensible heat required by the batched-cycle operation. The first term on the right hand side of Eq. (3.7) represents the release of adsorption heat or the input of desorption heat, while the second and third terms represent for the sensible heat of the adsorbed vapor. The last term on the right hand side of Eq. (3.7) indicates the total amount of heat released to the cooling water upon adsorption or provided by the hot water for desorption. Equation (3.7) does not account for external heat losses to the environment as all the beds are well insulated.

3.4.3 Energy balance of condenser

In the case of condenser, the same approach is taken for energy balance equation [11]. For condenser:

$$T_{cw,out} = T_c + (T_{cw,in} - T_c)\exp\left(-\frac{U_c A_c}{\dot{m}_{cw} C_{cw}}\right) \tag{3.8}$$

$$\begin{aligned}\frac{d}{dt}\{(W_{conw}C_w + W_{chex}C_{chex})T_e\} \\ = -LW_s\frac{dq_{des}}{dt} - W_s C_v(T_{des} - T_c)\frac{dq_{des}}{dt} \\ + \dot{m}_{cw}C_w(T_{cw,in} - T_{cw,out})\end{aligned} \tag{3.9}$$

The left hand side of Eq. (3.9) indicates the sensible heat required by the metallic parts of heat exchanger tubes caused by the temperature differences in the condenser. On the right hand side, the first term represents the latent heat of vaporization (L) for the amount of refrigerant desorbed ($\frac{dq_{des}}{dt}$), the second term illustrates the sum of sensible heat necessary to cool down the incoming vapor from the desorber bed at temperature T_{des} to condenser at temperature T_c, and the last term provides the total amount of heat released to the cooling water.

3.4.4 Energy balance of evaporator

The heat transfer and energy balance equations for evaporator can be expressed as mentioned by Hamamoto et. al. [11].

For evaporator:

$$T_{chil,out} = T_e + (T_{chill,in} - T_e)\exp\left(-\frac{U_e A_e}{\dot{m}_{chill} C_w}\right) \tag{3.10}$$

$$\begin{aligned}\frac{d}{dt}\{(W_{ew}C_w + W_{ehex}C_{ehex})T_e\} \\ = -LW_s\frac{dq_{ads}}{dt} - W_s C_w(T_c - T_e)\frac{dq_{des}}{dt} \\ + \dot{m}_{chill}C_w(T_{chill,in} - T_{chill,out})\end{aligned} \tag{3.11}$$

Based on Eq. (3.11), the left hand side of the term shows the sensible heat required by the liquid refrigerant or water (w) and the metal of heat exchanger tubes in the evaporator. The first term of the equation denotes the latent heat of evaporation (L) for the amount of refrigerants adsorbed ($\frac{dq_{ads}}{dt}$), while the second term provides the sensible heat needed to cool down the incoming condensate from the condensation temperature T_c to evaporation temperature T_e. The last term on this equation shows the total amount of heat loss by the circulating chilled water.

3.4.5 Mass balance

In the condition that the temperature and the amount of refrigerant adsorbed in the adsorbent beds are both in a uniform state, mass and heat balances are assumed to occur. As the temperatures in an adsorption cycle are unsteady, the energy balance equations (Eqs. 3.7, 3.9, and 3.11) must account for sensible heat input and/or output during the cycle period. The mass equilibrium of the refrigerant in the evaporator can be expressed as below.

$$\frac{dm_{ew}}{dt} = -m_{sg}\left(\frac{dq_{des}}{dt} + \frac{dq_{ads}}{dt}\right) \tag{3.12}$$

where the subscripts des represents the refrigerant vapor flows from desorber to condenser and ads stands for the flows from evaporator to adsorber, respectively.

3.4.6 Measurement of system performance

In order to analyse the performance of the chiller system, two main parameters are calculated which are the coefficient of performance (COP) and the cooling capacity (CC). Both of the parameters can be expressed based on equations below [13].

$$Cooling\ capacity = \dot{m}_{chill}C_{p,w}\int_{0}^{t_{cycle}} \frac{\left(T_{chill,in} - T_{chill,out}\right)}{t_{cycle}}\, dt \tag{3.13}$$

$$\mathrm{COP} = \frac{Q_{chill}}{Q_{hot}} \tag{3.14}$$

where $Q_{chill} = \dot{m}_{chill}C_{p,w}\int_{0}^{t_{cycle}}\left(T_{chill,in} - T_{chill,out}\right)\, dt$ (3.15)

and $Q_{hot} = \dot{m}_{hot}C_{p,w}\int_{0}^{t_{cycle}}(T_{hot,in} - T_{hot,out})\, dt$ (3.16)

Values for the parameters used in the equation is shown as in Table 3.4.

Table 3.4 Values of parameters for the simulation.

Parameter	Value
A_0	- 6.5314 kg (kg of dry adsorbent) $^{-1}$ [11]
A_1	0.72452 × 10^{-1} kg (kg of dry adsorbent, K) $^{-1}$ [11]
A_2	0. 2395l × 10^{-3} kg (kg of dry adsorbent, K^2) $^{-1}$ [11]
A_3	0.25493 × 10^{-6} kg (kg of dry adsorbent, K^3) $^{-1}$[11]
B_0	l5.587 [11]
B_1	0.15915 K [11]
B_2	- 0.506l2 × 10^{-3} K^{-2} [11]
B_3	0.5329 × 10^{-1} K^{-3} [11]
D_{s0}	2.54 × 10^{-4} m^2 s^{-1} [11]
E_a	4.2 × 10^4 J mol^{-1} [11]
L	2.5 × 10^3 kJ kg^{-1} [11]
M_{Cu}	12.67 kg [22]
M_{Al}	5.33 kg [22]
M_{sg}	14 kg [22]
U_{bed}	1724.14 W/m^2K [22]
U_{cond}	4115.23 W/m^2K [22]
U_{evap}	2557.54 W/m^2K [22]
$\dot{M}_{hot}$	0.30 kg/s [22]
$\dot{M}_{cool}$	0.30 kg/s [22]
$\dot{M}_{cond}$	0.34 kg/s [22]

3.5 Simulation and Optimization Procedure

The system design constants and kinetics constants used in the simulation are furnished by the given parameter shown in Table 3.4. The simulation of the system transient response is carried out with MATLAB and the solving of the differential equations is carried with the built in function ODE45 for solving ordinary differential equations. Based on the equations presented in the previous section, the system of differential are solved simultaneously by numerical integration using finite difference substitution in the derivatives. The calculations are made with approximation of a time step of one second. The results shown in the study are from the cyclic steady state conditions.

Initially, the two-stage four-bed adsorption chiller started its operation with an unbalance condition. After a few cycles of operation it reached its balance condition. In order to fix the initial conditions of the cyclic steady state, iteration process is employed in the solution procedure. Therefore, in the beginning, assumptions are made on the initial condition and the values are adjusted later by the iteration process so that the cyclic steady state condition can be achieved.

When two beds are connected with each other, the vapor pressure is unknown. The parameters are required for the calculation of adsorption/desorption rate inside the adsorbent beds. Therefore, in the state vapor pressure is assumed initially and the amounts of vapor adsorbed/desorbed by the beds are calculated. Theoretically, the amount of desorbed vapor from one bed should be equal to the amount of adsorbed vapor by the other bed. If this condition is not achieved, the value are adjusted for next iteration. The convergence factor is taken as 10^{-5} for all parameters.

Adsorbent refrigerant properties, flow rates of heat transfer fluids and heat exchangers specifications are some of the input constant parameters which are initially specified so that the system cyclic operation can be continued. The simulation procedure can be seen in details based on Appendix. The procedure is adopted based on single-stage four-bed system [21], but some modifications are made to implement the two-stage four-bed behaviour for this process.

CHAPTER 4 : RESULT AND DISCUSSION

4.1 Critical Analysis

4.1.1 Chiller Transient Response

The cycle simulation runs are executed with the purpose of clarifying the performance of mass recovery cycle. The main concern for this study is to utilize the low grade waste heat as the driving source. Therefore, the analysis is done for hot water below 60°C. The base line parameters and standard operating conditions for the chiller operation are shown as in Table 4.1.

Table 4.1 Rated condition of the chiller system.

Refrigerant	Hot water inlet		Cooling water inlet		Chilled water inlet	
	Temperature (ºC)	Flow rate (kg/sec)	Temperature (ºC)	Flow rate (kg/sec)	Temperature (ºC)	Flow rate (kg/sec)
Water	55	0.30	30	0.30	13.8	0.11
Adsorption/desorption cycle time : 400 sec			Switching time for both cycles : 40 sec			

Based on figure 4.1, the temperature profiles of different components of the adsorption chiller in two-stage, four-bed mode is shown. The curve shown for SE1, SE2, SE3, and SE4, the four beds are operating within the temperature limit of below 55ºC, which represents the hot water inlet for this process. The four beds also functions above the inlet cooling water which is 30ºC. The behaviour of the beds in this case is according to the predicted results from literature studies [11].

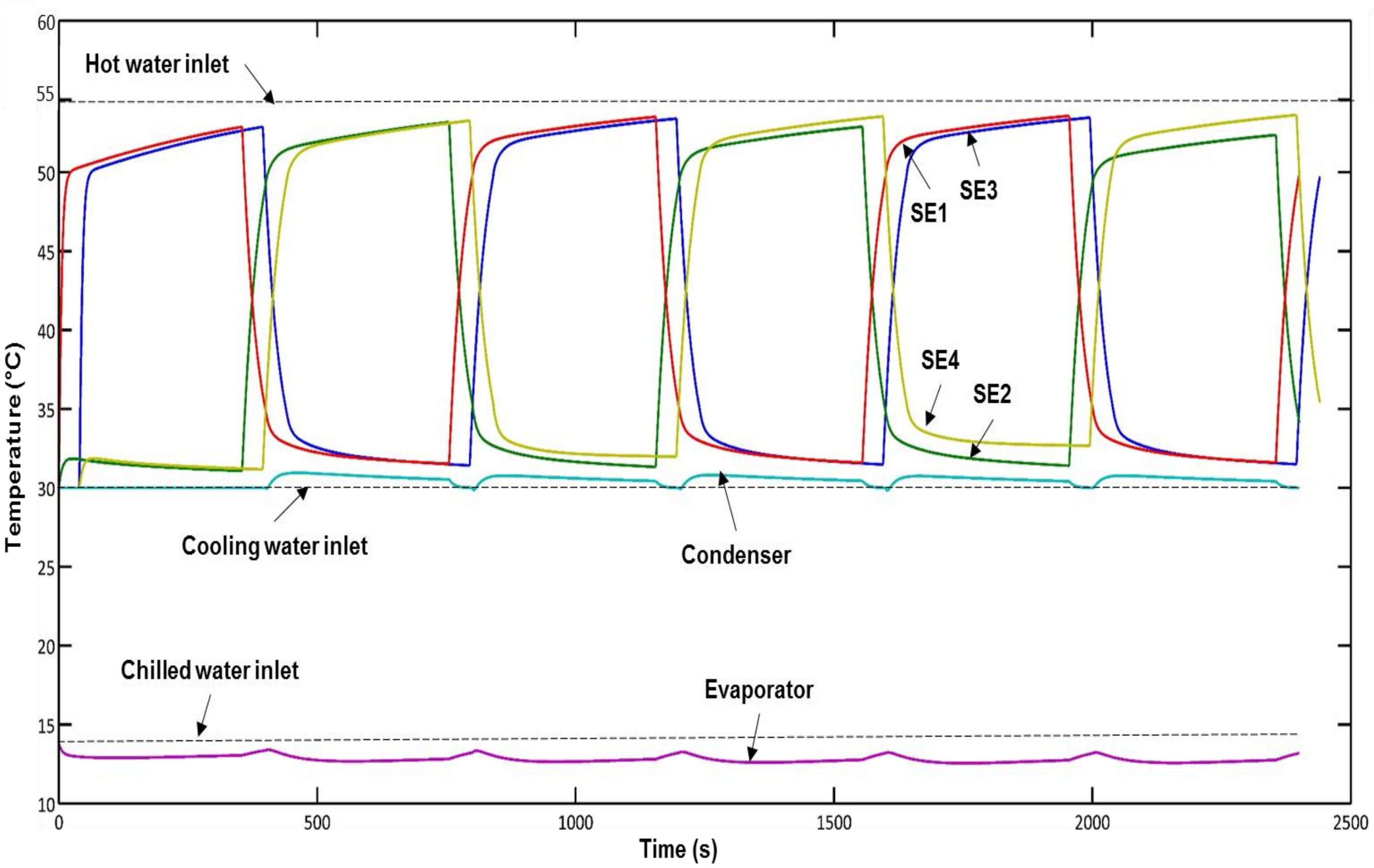


Figure 4.1 Temperature profiles of different components of the adsorption chiller in two-stage, four-bed mode.

4.1.2 Adsorption/Desorption Cycle Time

Figure 4.2 shows the effect of adsorption/desorption cycle time on both cooling capacity and COP. The highest cooling capacity values are obtained for adsorption/desorption cycle times between 400 and 500 s. For the cycle time below than 400 s, the cooling capacity increases by increases with the increase of the cycle time. This because is because the mass of refrigerant adsorbed and hence circulated increases with the increase of the cycle time.

On the other hand, for cycles longer than 500 s, the cooling capacity decreases this is because at the very large cycle time would saturate the adsorbed vapor onto the refrigerant water and cooling rate would decrease. Thus, there exists an optimal COP, which lies between these extreme cycle intervals for the advanced two-stage chiller.

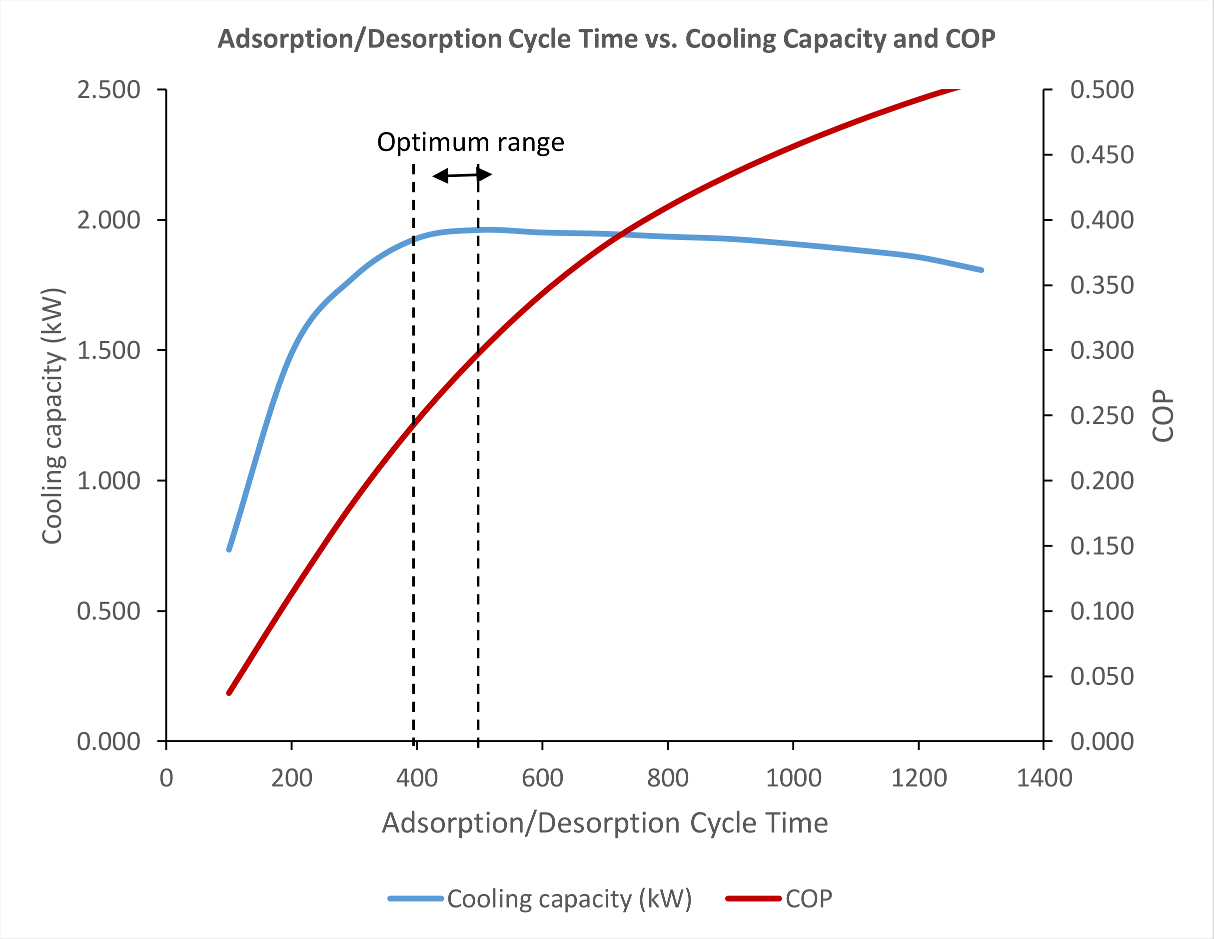


Figure 4.2 Effects of Adsorption/Desorption Cycle Time on Cooling capacity and COP.

4.1.3 Switching Time

The effects of switching time on the COP and cooling capacity is illustrated as in Figure 4.3. It can be seen that the cooling capacity increases gradually as the value of switching time is raised from 10 sec to 30 sec. Then the cooling capacity reaches its peak value of 1.65kW and become almost steady before it started to have a slight decrement at 70 sec and onwards. Moreover, COP value also shows the same pattern as the cooling capacity curve. When the switching time is raised from 10 sec to 30 sec, the value of COP grows significantly from 0.209 to 0.233. After that, value started to increase gradually and become steady until it reaches a peak value of 0.238. At switching time of 70 sec onwards, COP started to decrease in gradual manner. The optimum value of switching time for the cycle is around 40 s.

Determining the switching time is important as to ensure that the beds are not immediately connected to condenser or evaporator. It is important to allow a period of isolated, near-isosteric thermal swing to occur before changing the circulation of hot and cool water between the beds. The reason is because a premature connection between the regenerated hot bed and the evaporator would translate into momentary desorption of adsorbed refrigerant and undesirable reduction in instantaneous cooling power.

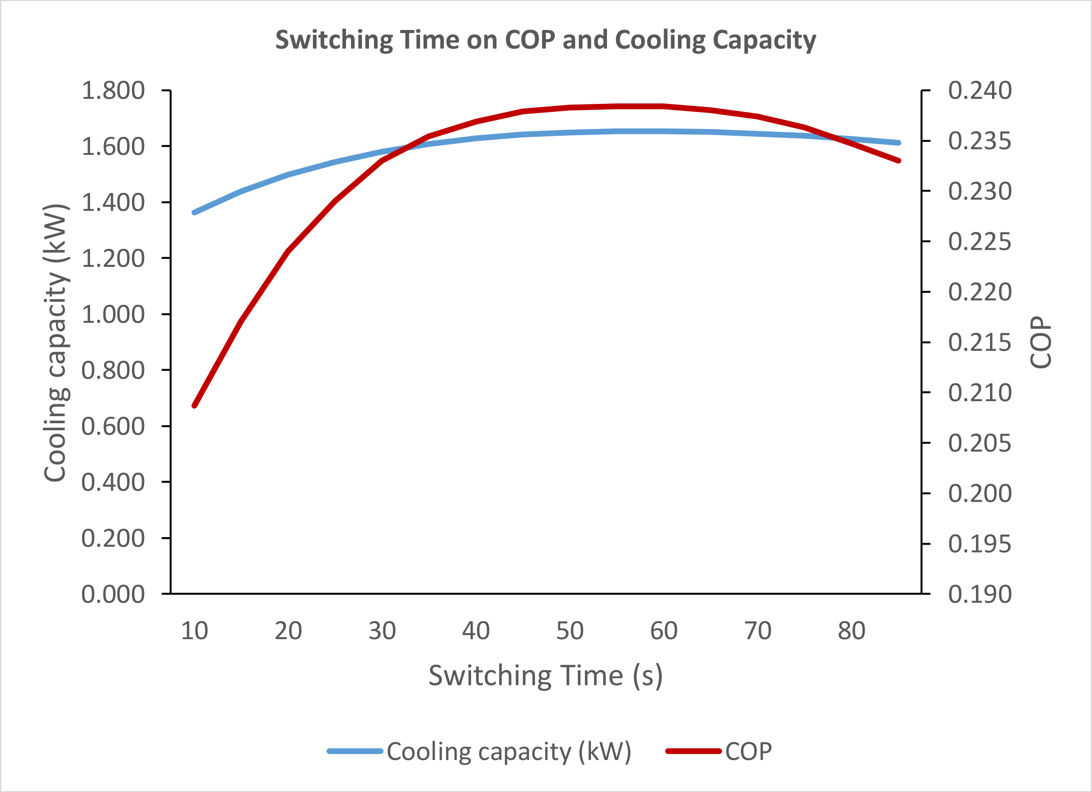


Figure 4.3 Effects of Switching Time on COP and Cooling capacity.

4.1.4 Heat Source Temperature

Figure 4.4 and 4.5 shows the effect of heat source temperatures on cooling capacity and COP, respectively. It is shown that as the regenerating temperature of heat source temperature increases, the cooling capacity and COP of the chiller also increases. The reason for this is that higher heat source temperature increases the amount of desorption for the refrigerant. Consequently, the amount of refrigerant being circulated also increases.

It can be observed from the figure, for two-stage operation, the cooling capacity increases almost linearly from 0.78kW to 2.78kW as the heat source temperature is raised from 40°C to 60°C. For single stage operation, as the heat temperature rises from 60°C to 95°C, the cooling capacity also increases significantly from 5.5kW to 15.4kW compared to the two-stage operation. The reason is because the amount of refrigerant desorption also escalates with a higher regenerating temperature used on the beds. In terms of COP value, for two-stage mode, there is a significant increase from 0.108 to 0.279 as the heat source temperature rises from 40°C to 60°C. On the other hand, for one-stage operation, the COP increases gradually.

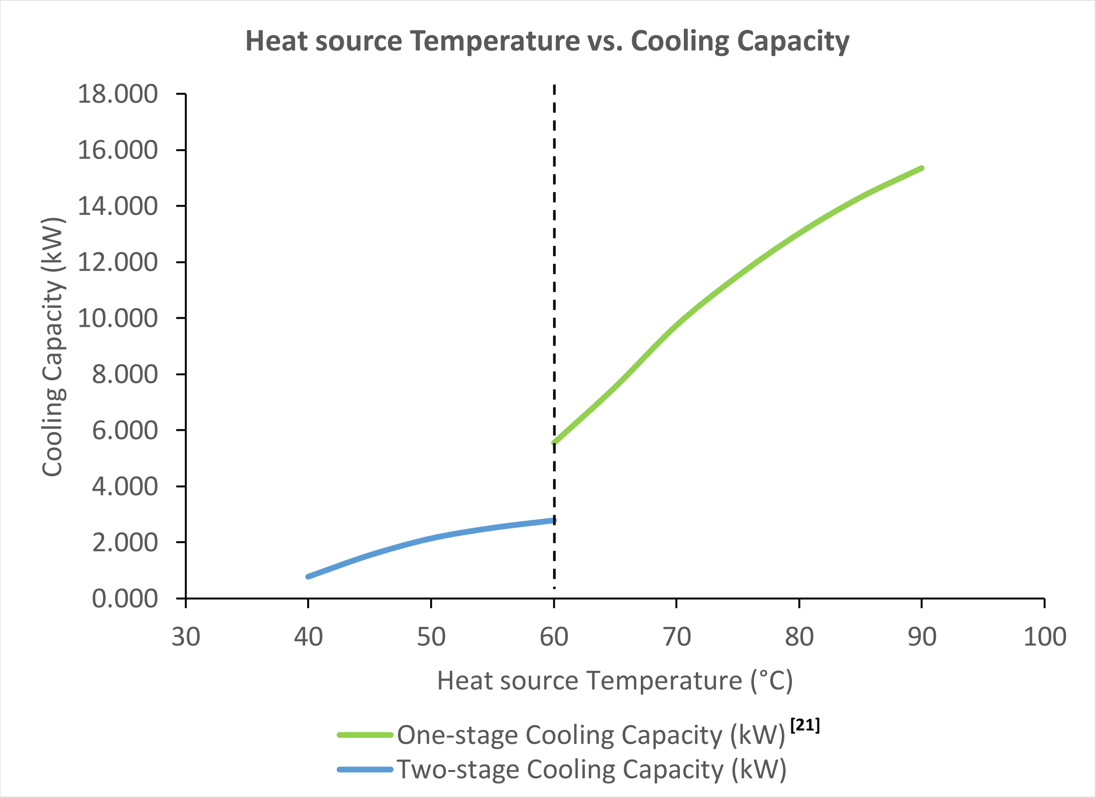


Figure 4.4 Heat source temperature effect on cooling capacity

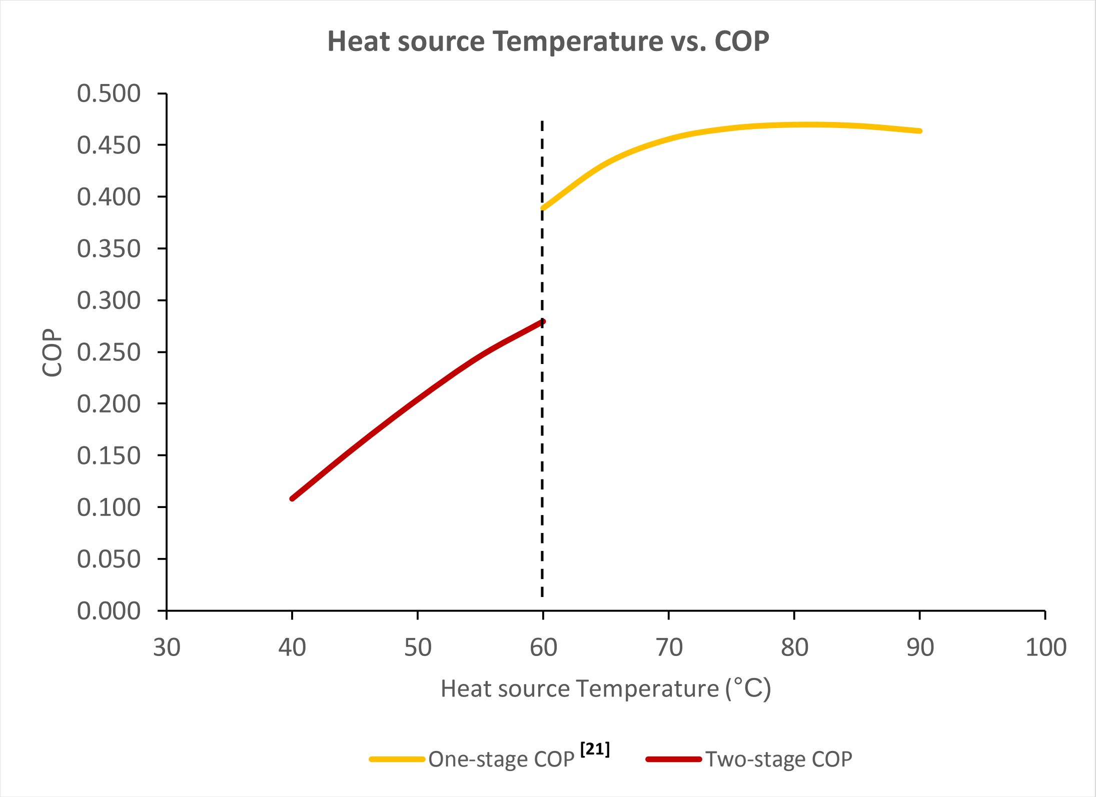


Figure 4.5 Heat source temperature effect on COP

4.1.5 Cooling Water Temperature

Figure 4.6 shows the effect of cooling water inlet temperatures on cooling capacity and COP, respectively. In the present simulation, cooling water mass flow rate into adsorber is taken as 0.4 kg/s, while for the condenser the coolant mass flow rate is taken as 0.34 kg/s. From the figure, it can be seen that the cooling capacity decreases steadily as the cooling water inlet temperature is raised from 10 to 40°C. This is due to the fact that higher adsorption temperature causes smaller amounts of refrigerant being adsorbed and desorbed during each cycle. Consequently, for a lower cooling water inlet temperature, the higher the amount of adsorption and desorption of refrigerant in the process.

Figure 4.6 also portrays that the simulated COP values also increase with lower cooling water inlet temperature. For the two-stages four-bed chiller the COP value reaches 0.24 with 55°C driving source temperature in combination with a coolant inlet temperature of 30°C.

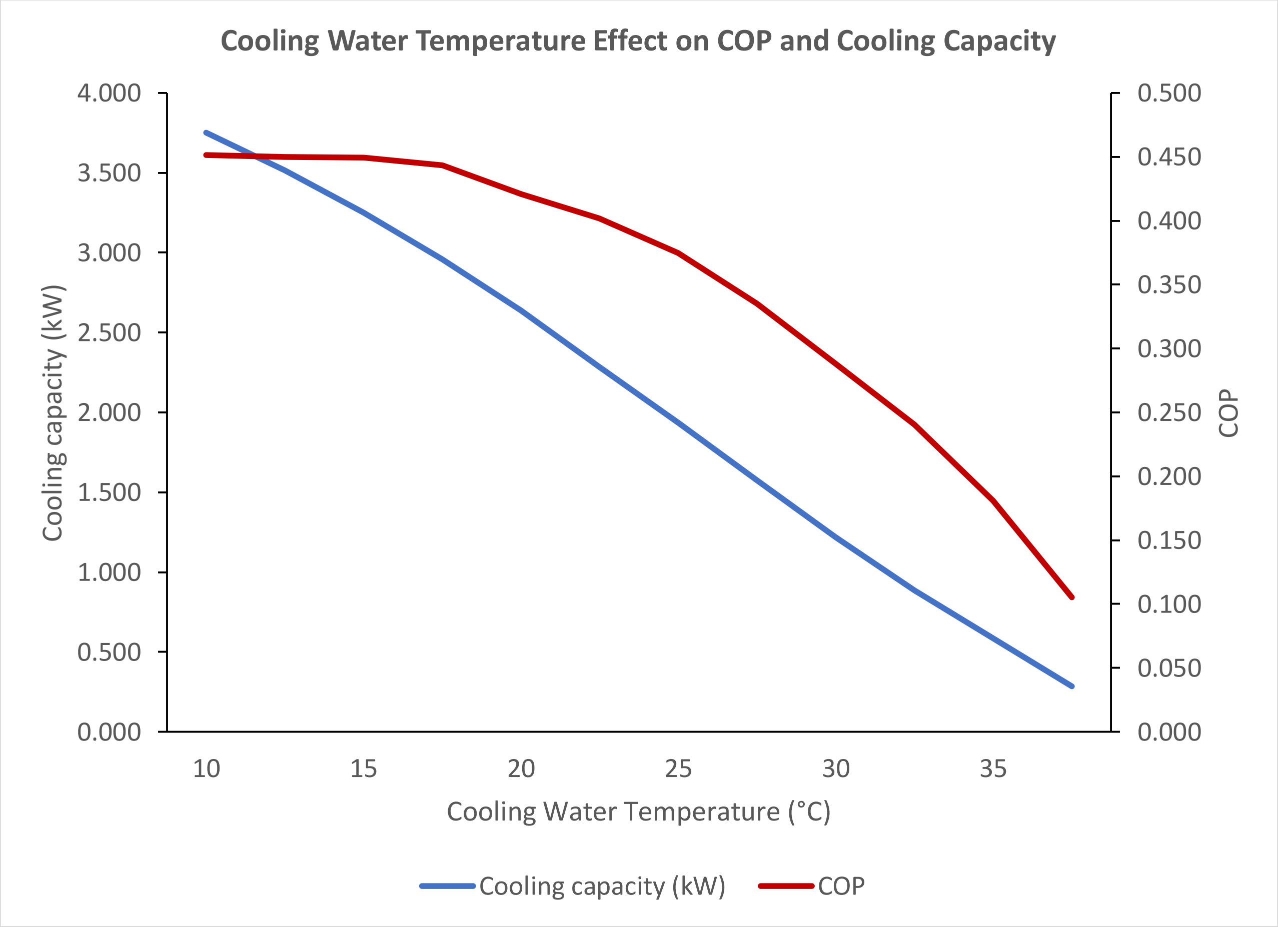


Figure 4.6 Cooling water temperature effects on cooling capacity and COP

4.1.6 Chilled Water Temperature

Figure 4.7 shows the effect of chilled water temperature variations on the cooling capacity and COP of the system. It can be observed that the cooling capacity increases gradually from 0.3kW to 7.6kW when the chilled water temperature is varied incrementally from 10°C to 45°C. In this curve, the growth of the cooling capacity seems to be almost similar to a linear growth.

On the other hand, the curve line for COP seems to grow in logarithmic pattern as there is a significant increase in COP when the chilled water temperature is varied from 10°C to 25°C. However, as the chilled water temperature increases from 25°C to 45°C, the COP value increases slightly from 0.41 to 0.56.

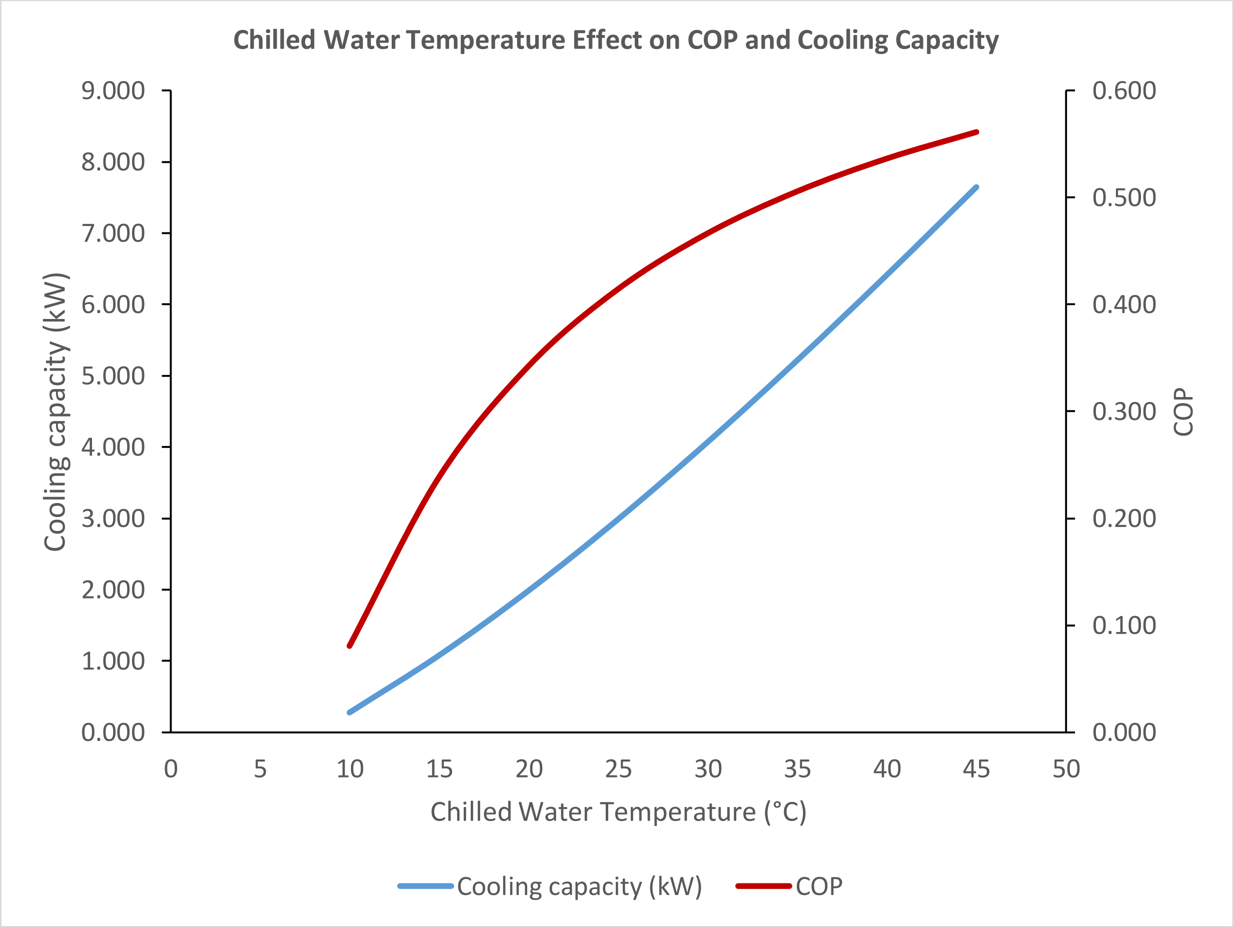


Figure 4.7 Chilled water temperature effects on cooling capacity and COP

4.1.7 Hot Water Flow Rates

Figure 4.8 shows the hot water flow rate effects on the cooling capacity and COP of the chiller system. It can be noticed that the curve of cooling capacity and COP are acting against each other. As the hot water flow rates is increased, the cooling capacity rocketed from 1.873kW to 1.878kW at 2.0 kg/s. After that, it started to increase steadily until it reaches 1.885kW at 8.0 kg/s. On the other hand, the value of COP drop steeply from 0.248 to 0.241 at 2.0 kg/s. Then the decrement started to become more gradual until it reaches 0.236 at 8.0 kg/s. In general, the cooling capacity increases whereas the COP decreases as the value of the hot water flow rate increases.

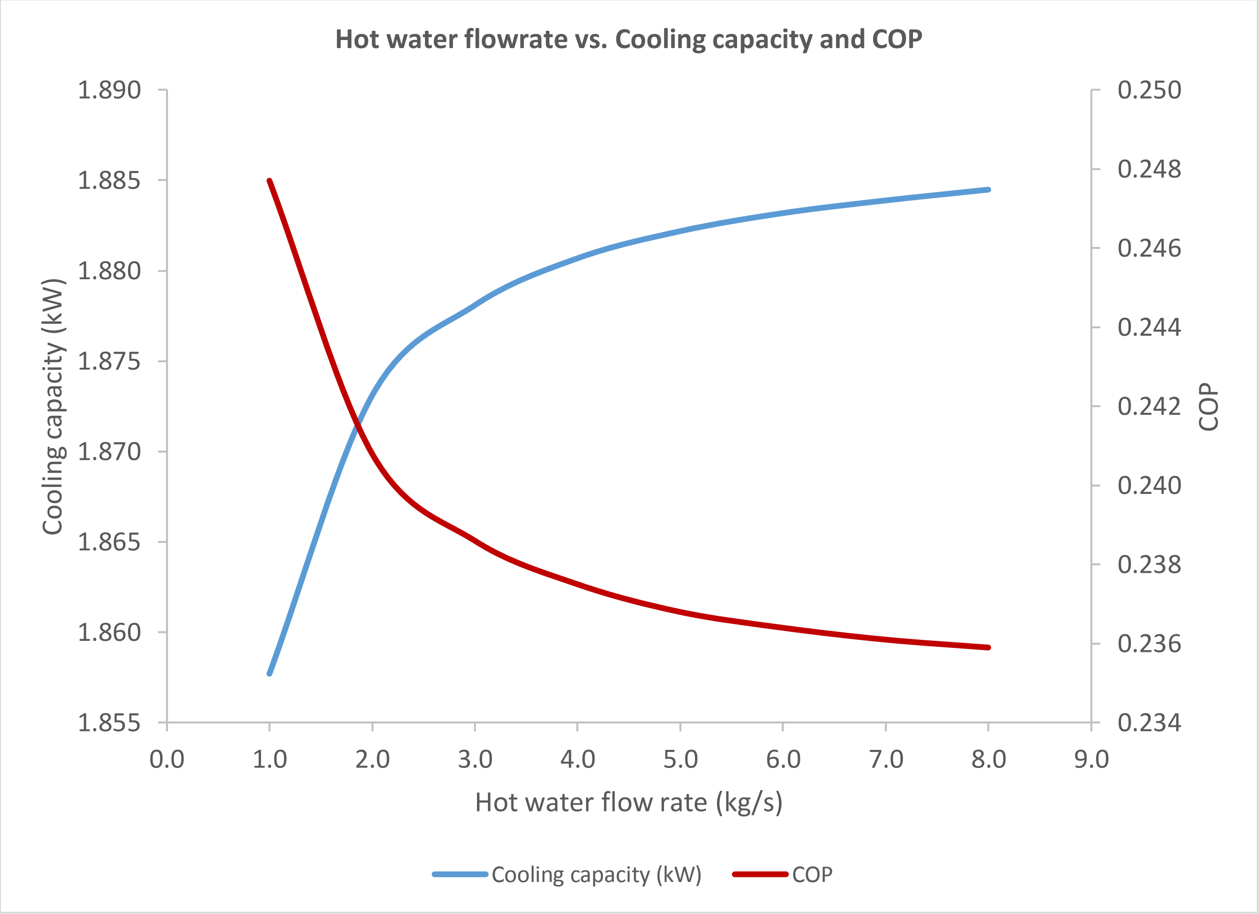


Figure 4.8 Hot water flow rate effects on cooling capacity and COP

4.1.8 Chilled Water Flow Rates

The chilled water flow rates effect on cooling capacity and COP for the cycle is shown in Figure 4.9. It can be observed that the curve for COP and cooling capacity are both similar. For cooling capacity, as the chilled water flow rates is raised from 0.1 kg/s to 0.3 kg/s, the value of cooling capacity surges up from 1.94kW to 2.2kW. After that, there is a slight gradual increase in the cooling capacity from 2.2kW to 2.3kW as the chilled water flow rates is also increased.

Similarly, as the chilled water flow rates increases from 0.1 kg/s to 0.3 kg/s, the COP value grow steeply from 0.26 to 0.289. Afterwards small incremental occurs on the COP as the chilled water flow rates is raised further until 1.0 kg/s.

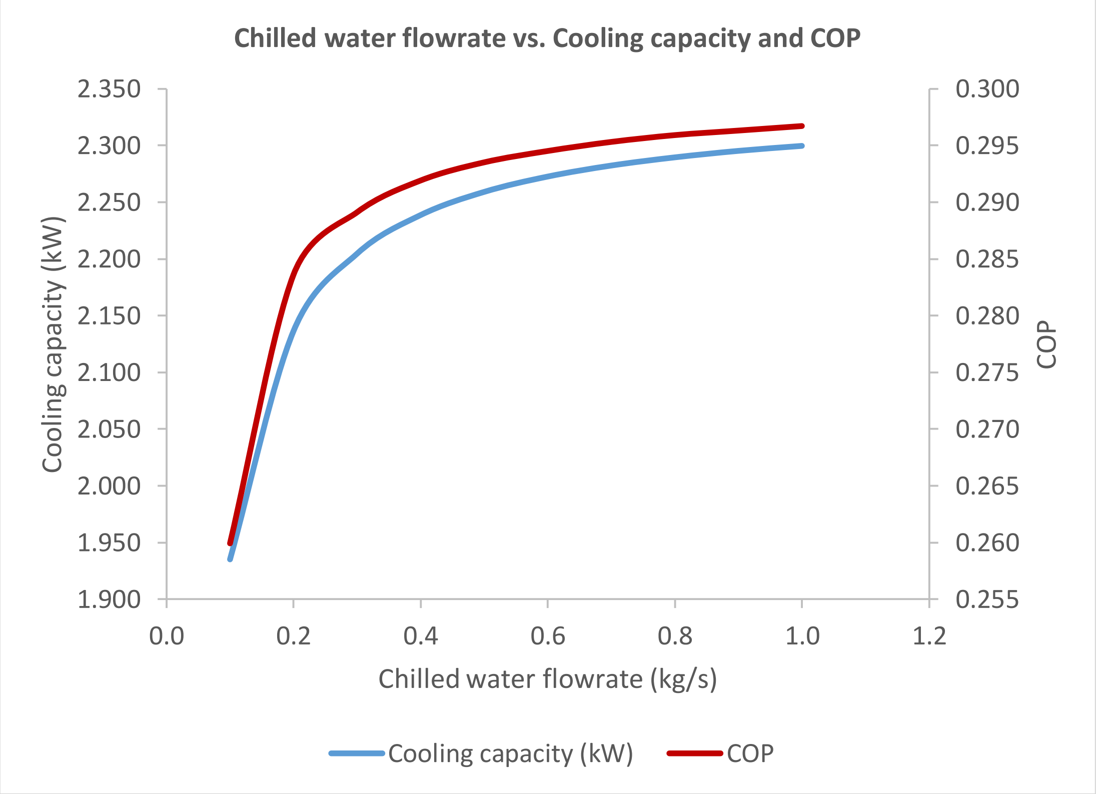


Figure 4.9 Chilled water flow rate effects on cooling capacity and COP

CHAPTER 5 : CONCLUSION AND RECOMMENDATIONS

5.1 Conclusion

Adsorption chiller is one of the effort made to reduce the reliance of mankind towards energy that utilizes fossil fuels. This system can be powered by renewable energy like solar energy. With a long operating life of the system, it brings no harm to the environment as the refrigerant used in this case is water, which has zero Global Warming Potential and Ozone Depletion Potential. In this paper, the working principle of the two-stage four-bed adsorption chiller has been outlined and the mathematical concept has been analysed.

The objective of this study is to investigate the round the effects of two-stages, four-bed adsorption chiller driven by solar heat in Malaysia. The main conclusions are as follows:

(1) The main advantage of the two-stage adsorption chiller cycle is its ability to utilize low range of driving heat source of temperature below 60°C. Therefore it feed the purpose to allow it to run especially when solar radiation is low in Malaysia.

(2) The higher the heat source temperature, the higher the cooling capacity and COP at which the adsorption chiller can deliver. In this study, since the temperature range is below than 60°C, the COP is lower compared to four bed single-stage adsorption chiller system. With heat source temperature of 55°C, cooling water of 30°C and chilled water running at 14°C, the COP of the system achieved is around 0.24.

(3) For the cycle time below than 400 s, the cooling capacity increases by increases with the increase of the cycle time. Long adsorption/desorption cycle times (above 400 s) result in COP gains and cooling capacity losses. The optimum range obtained for adsorption/desorption cycle times is between 400 and 500 s.

(4) From the above perspectives, the use of unexploited renewable energy at near-ambient temperature may offer attractive possibilities for improving energy efficiency.

5.2 Recommendations

Through this study, the behaviour of the cycle can be further understood and thus some innovations can be made for the manufacturers to improve their system. Most importantly, this two-stage four-bed refrigeration system feeds the purpose to which low-grade heat source from industrialization can be utilized to operate the system.

The chiller system can be improvised by developing a dual-mode adsorption chiller by pairing the two-stage operation with one-stage operation. During hot weather in Malaysia where high heat source temperature can be obtained, the system can operate using one-stage mode. On the other hand, during rainy seasons where solar radiation is relatively lower and the heat source temperature is below 60°C, the adsorption chiller can be operated using a two-stage mode. Having able to improve the current system will allow an increase on the cooling capacity and COP as a whole. This will make the chiller to function more efficiently during the whole year.

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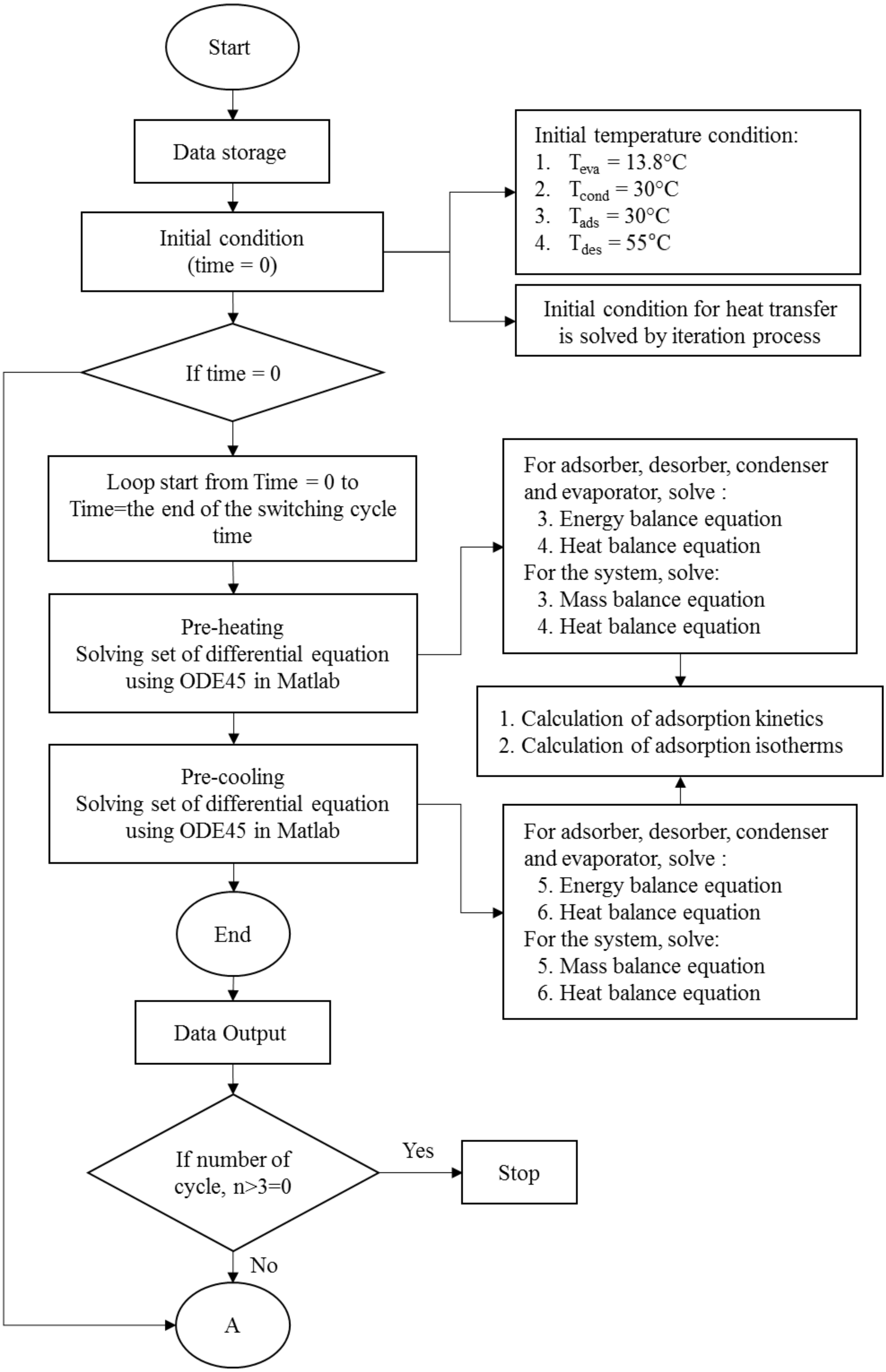
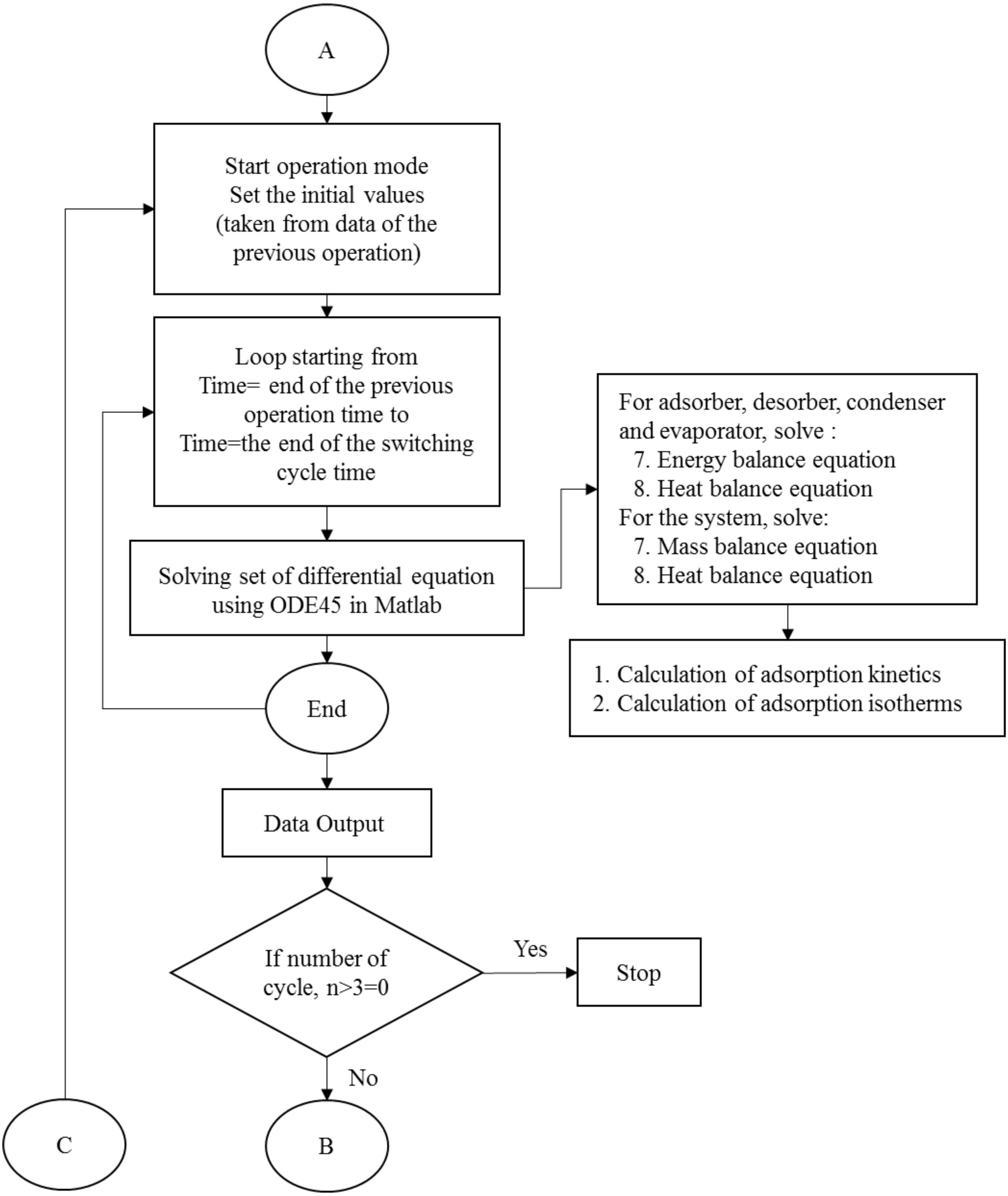
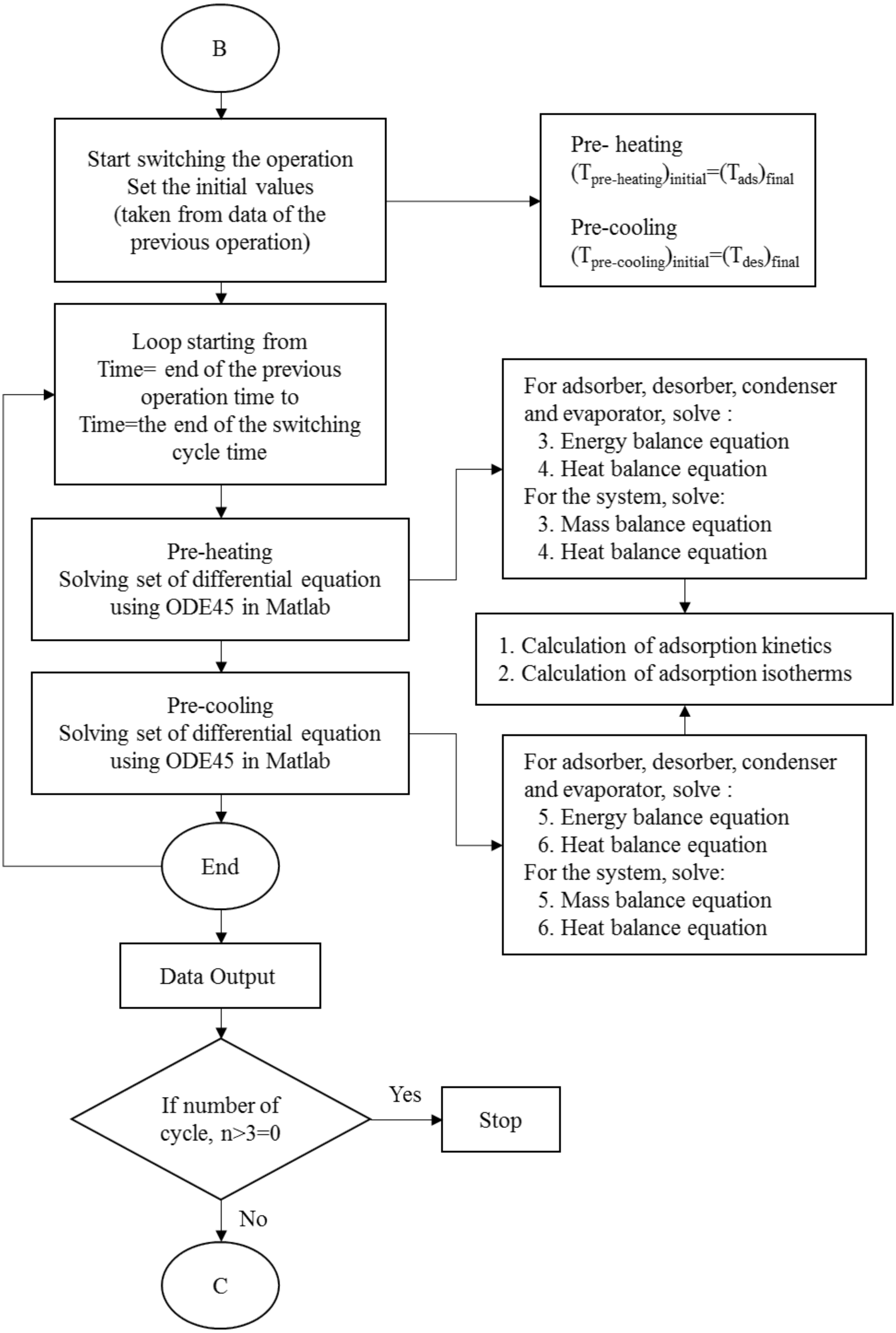
APPENDIX

Simulation flowchart of the system




A
Start operation mode
Set the initial values
(taken from data of the
previous operation)
Loop starting from
Time= end of the previous
operation time to
Time=the end of the switching
cycle time
Solving set of differential equation
using ODE45 in Matlab
For adsorber, desorber, condenser
and evaporator, solve :
7. Energy balance equation
8. Heat balance equation
For the system, solve:
7. Mass balance equation
8. Heat balance equation
1. Calculation of adsorption kinetics
2. Calculation of adsorption isotherms
End
Data Output
If number of
cycle, n>3=0
Yes
Stop
No
C
B


B
Start switching the operation
Set the initial values
(taken from data of the
previous operation)
Pre- heating
$(T_{pre\text{-}heating})_{initial}=(T_{ads})_{final}$
Pre-cooling
$(T_{pre\text{-}cooling})_{initial}=(T_{des})_{final}$
Loop starting from
Time= end of the previous
operation time to
Time=the end of the switching
cycle time
For adsorber, desorber, condenser
and evaporator, solve :
3. Energy balance equation
4. Heat balance equation
For the system, solve:
3. Mass balance equation
4. Heat balance equation
Pre-heating
Solving set of differential equation
using ODE45 in Matlab
1. Calculation of adsorption kinetics
2. Calculation of adsorption isotherms
Pre-cooling
Solving set of differential equation
using ODE45 in Matlab
For adsorber, desorber, condenser
and evaporator, solve :
5. Energy balance equation
6. Heat balance equation
For the system, solve:
5. Mass balance equation
6. Heat balance equation
End
Data Output
If number of
cycle, n>3=0
Yes
Stop
No
C