

**PERFORMANCE AND COST ANALYSES OF AN INTEGRATED
SYSTEM OF M-CYCLE-BASED COOLING AND ABLUTION
GREYWATER TREATMENT FOR MOSQUES IN HOT AND DRY
CLIMATIC ZONES**

BY

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Dedication

To my beloved parents, for their endless love, support, and encouragement

To my lovely brothers and sisters who taught me how to stand up

To my second half and son who helped me to stay stand

To my professors, for their help and guidance

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My dear prayers for this work to be beneficial for the humanity

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NOMENCLATURE

<i>Symbol</i>	Definition	Unit
C	Specific heat of air	kJ/kg K
d_h	Hydraulic diameter	m
D_{va}	Binary diffusion coefficient of water vapor in air	m ² /s
h	Convective heat transfer coefficient	W/m ² K
h_m	Mass transfer coefficient	kg/m ² s
$H_{w,fg}$	Latent heat of water	J/kg
K	Thermal conductivity	W/m K
$\dot{m}$	Mass flow rate	kg/s
M	Number of nodes	nodes
Nu	Nusselt number	-
Sh_w	Sherwood number	-
T	Temperature	°C
t	Thickness	m
U	Overall heat transfer coefficient	W/m ² K
W	Width of the heat exchanger	m
$\dot{W}$	Power input	kW

y	Channel spacing	m
Δx	Distant between adjacent nodes	m
ε	Effectiveness	-
ρ	Density	kg/m ³
ω	Humidity ratio of air, kg water/kg dry air	kg water/kg dry air
h	Convective heat transfer coefficient	W/m ² K
$\dot{V}$	Volumetric flow rate	m ³ /s
ΔP	Pressure drop	
f	safety factor	-
WC	Water consumption	m ³ /Kwh
IC	Initial cost	\$
EC	Electricity consumption	KW
SH	The static head of the fan	Pa

SUBSCRIPTS

d	Dry channel
w	Wet channel
WB	Wet bulb
DP	Dew Point
wf	Water film
i	Capital
F	Filter
O	Operating
C	Chemical

ABBREVIATIONS

COP	Coefficient of Performance
HVAC	Heating, Ventilation and Air Conditioning
CAD	Computer Aided Design
DEC	Direct Evaporative Cooling
EC	Evaporative Cooling
EES	Engineering Equation Solver
IEC	Indirect Evaporative Cooling
NTU	Number of Transfer Units
REC	Regenerative Evaporative Cooling
RH	Relative Humidity

ABSTRACT

Full Name : Faisal Bader Al-Mutairi

Thesis Title : Performance and Cost Analyses of an Integrated System of M-Cycle based Cooling and Ablution Greywater Treatment for Mosques in Hot and Dry Climatic Zone

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One of the most unique building types in Muslim communities are mosques where a substantial amount of waste ablution water with a low pollutant level is generated and a large amount of energy is consumed for cooling. There are 101,108 mosques in Saudi Arabia where Muslims perform their prayers five times a day. The energy use of mosques is within the 16.2% average energy consumed by government buildings in the Kingdom, which accounts for a total electricity bill of 700 million Riyals annually and an average estimated waste ablution water of 40 L/h.

Riyadh city represents 17.4% of the total number of mosques in Saudi Arabia and it is classified as hot and dry climate zone. Accordingly, mosques located in Riyadh totally rely on HVAC systems to provide acceptable thermal comfort for the occupants. Thus, this study proposes an integrated ablution greywater treatment and M-cycle based cooling system for mosques located in such climatic zone. The study recommends to redesign the ablution and toilet areas in mosques to separate and treat the ablution greywater to be used in mosque cooling.

A theoretical model was developed to predict the performance of the proposed cooling system using a variation of the Newton's method. The developed model was tested and validated with an experimental data available in the literature and the results show good agreement with an overall

error of 3.3%. The relative error is due to several factors, including but not limited to the non-uniform flow of air in the wet channel and the uneven water distribution in the wet channel.

The performance of the proposed system was also investigated and the result shows that the average monthly supplied air temperature ranges from 13.8 to 19.89 °C and the relative humidity ranges from 51% to 72%. Moreover, a wet-bulb effectiveness greater than unity will be achieved with a channel length greater than about 1.2 m .Furthermore, the working air to supply air ratio must be within the range of 0.30 to 0.63, as the efficiency and cooling effectiveness of the system will be affected by higher values. Additionally, the effectiveness of the system decreases significantly as the inlet velocity increases.

Based on the technical and economic analysis of the potential grey water treatment system , filtration and activated carbon adsorption (FA) based treatment system is the most feasible option to be utilized in the proposed integrated system with a total annualized estimated cost of 0.00085 \$/L. The economic analysis shows that the capital cost of the conventional cooling system is 54.2% less than the capital cost of M-cycle-based cooling system. On the other hand, M-cycle-based cooling system consumes 88% less electricity than that consumed by a conventional cooling system and consequently the payback period of the proposed system is about 18 months.

ARABIC ABSTRACT

الاسم الكامل : فيصل بدر المطيري

عنوان الرسالة : تحليل فني واقتصادي لإعادة استخدام مياه الوضوء المبددة لغرض التكييف باستخدام دورة ميزوتسينكو للتبريد و نظام معالجة مياه مبني على الترشيح وامتصاص الكربون المنشط للمساجد الواقعة في المناطق الحارة والجافة

التخصص : الهندسة الميكانيكية

تاريخ الدرجة العلمية : 26 ربيع الآخر 1442 هـ

تعد المساجد أحد أهم المعالم المميزة في المجتمع الإسلامي , حيث يتم إهدار الكثير من مياه الوضوء والتي تعد ملوثة تلويثاً ضئيلاً كما يتم في تلك البقاع الطاهرة إهدار كثير من الطاقة لغرض التبريد لضمان راحة المصلين اثناء تأدية العبادة , يبلغ عدد المساجد في المملكة العربية السعودية حوالي 101,108 مسجد حيث يتهافت المسلمون لأداء خمس صلوات في اليوم والليلة , تصنف المساجد ضمن المباني الحكومية والتي يبلغ استهلاكها للطاقة 16.2% من مجموع استهلاك الكهرباء في المملكة العربية السعودية , وقد تجاوز مجموع ما أنفقته المساجد على الطاقة سبع مائة مليون ريال سعودي , كما بلغت كمية مياه الوضوء المهذرة أربعون لتر/ساعة.

بلغ عدد المساجد في منطقة الرياض 17.4% من مجموع عدد المساجد في المملكة العربية السعودية, وكما هو معلوم فإن منطقة الرياض تعد ذات مناخ يتسم بالحرارة والجفاف في المجل وببناء على ذلك فإن المساجد التي تقع في منطقة الرياض تعتمد اعتماداً كلياً على أنظمة التبريد والتكييف لتزود المصلين بالبيئة الملائمة لأداء العبادة, ولذا سعت هذه الدراسة لاختبار إمكانية تصميم نظام جديد يستخدم مياه الوضوء المعالجة واستخدامه لتبريد الهواء بواسطة نظام تبريد مبني على دورة ميزوتسينكو , حيث تقترح هذه الدراسة إعادة تصميم أماكن الوضوء ودورات المياه في المساجد لفصل ومعالجة مياه الوضوء المستهلكة ليتم استخدامها لغرض التبريد , ولتحقيق هذه الغاية فقد تم تطوير النموذج النظري لتوقع أداء نظام التبريد المقترح. كما تم اختبار هذا النموذج و التحقق من صحته بمقارنة نتائجه بأخرى تجريبية متوفرة في الأبحاث المشابهة لموضوع البحث وقد أظهرت النتائج توافقاً جيداً مع خطأ إجمالي بنسبة 3.3% حيث يعود الخطأ النسبي إلى عدة عوامل بما في ذلك على سبيل المثال لا الحصر, التدفق غير المنتظم للهواء والتوزيع الغير متكافئ للمياه في القناة الرطبة.

كما أظهرت النتائج أن متوسط درجة حرارة الهواء الناتجة من النظام تتراوح ما بين 13.8 إلى 19.9 درجة مئوية كما تتراوح نسبة الرطوبة النسبية من 51% إلى 72%، بالإضافة إلى أن فعالية النظام قد تتجاوز 100% إذا بلغ طول قنوات النظام 1.2 متر. ومن جهةٍ أخرى، يجب أن تتراوح نسبة الهواء المرجع إلى القناة الرطبة ما بين 30% إلى 60% من مجموع الهواء الناتج من النظام ، حيث ستتأثر كفاءة وفعالية التبريد للنظام إن زادت القيمة عن ذلك.

بناءً على دراسة فنية واقتصادية لعدة أنظمة مقترحة لمعالجة مياه الوضوء المبددة، يعد نظام معالجة مياه الوضوء المبني على الترشيح وامتصاص الكربون المنشط أفضل الخيارات وأكثرها جدوى ليتم دمجها مع نظام التبريد المقترح ، حيث تبلغ تكلفة إنتاج المياه 0.00069 دولار/ لتر والتي هي أقل من تكلفة إنتاج المياه عن طريق وسائل التحلية المتاحة.

وختاماً فقد أظهر التحليل الاقتصادي للنظام أن التكلفة المالية لنظام التبريد التقليدي أقل بنسبة 54.2% من التكلفة المالية لنظام التبريد القائم على دورة ميزوتسينكو ولكن في المقابل فإن نظام التبريد القائم على دورة ميزوتسينكو يستهلك طاقة كهربائية أقل بـ 88% من تلك التي يستهلكها النظام التقليدي وبالتالي فإن فترة استرداد رأس المال المدفوع لإنشاء النظام المقترح هي 18 شهر.

CHAPTER 1

INTRODUCTION

1.1 Background

In the past few decades the consumption of energy has sharply increased mainly due to the increase in the world population, which is predicted to rise by eighty five percent by 2040 [1]. The largest percentage of global energy demand is met using fossil fuel resources, which are also depleting at a rapid rate. Their combustion also has detrimental effects on humans and the environment[2][3][4]. Studies have revealed that the energy consumption of systems used in HVAC (heating, ventilation, and air-conditioning) is more than 40% of the consumed total energy, particularly in hot climatic zones. In China, about fifteen percent of the total energy generated is used to power HVAC systems. About half of the total energy generated in Saudi Arabia is estimated to be used to power HVAC systems [5]. The peak load demand in Saudi Arabia in the year 2000 was 21,673 MW, which increased to 34,953 MW by the year 2007 and reached 56,547 MW by the year 2014 [6]. The HVAC systems in building sector are the largest consumer of electric energy. Hence, it is imperative that efforts are made to curb the enormous amount of energy used to power conventional HVAC systems. Such action will decrease the cost of maintaining comfortable environments in buildings as well as protecting the planet by reducing the amount of greenhouse gases (GHGs) released to the atmosphere.

1.2 Overview of Saudi Arabia

Saudi Arabia is a large country with a land area of 2,149,690 km² and a population of about 33 million with an annual growth rate of 1.8%[6][7][8]. The country consists of five major regions, which are distinct in terms of climatic and geographic zoning. However, generally the Kingdom of Saudi Arabia is classified as an arid country [9]. One third of the total daily oil production in Saudi Arabia is used for electricity generation as illustrated in Figure 1-1[10][11].

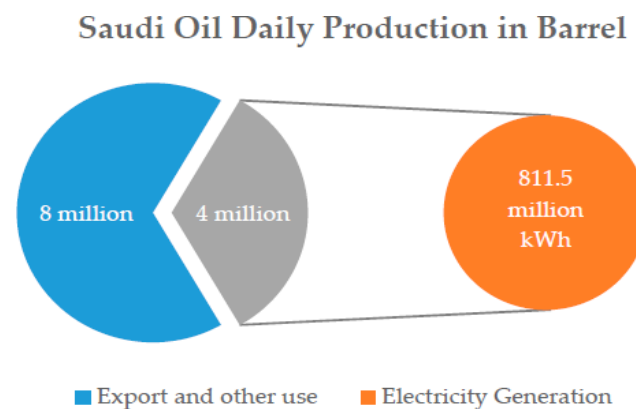


Figure 1-1: Daily oil production of Saudi Arabia in Barrels [8].

In Saudi Arabia, the building sector consumes 80% of the electricity generated daily, as shown in Figure 1- 2 [12]. HVAC systems, which are an integral part of buildings in arid countries such as the Kingdom of Saudi Arabia, account for 70% of the total electricity consumed by buildings [13]. Hence, developing alternative efficient air conditioning systems is essential to reduce energy use in buildings.

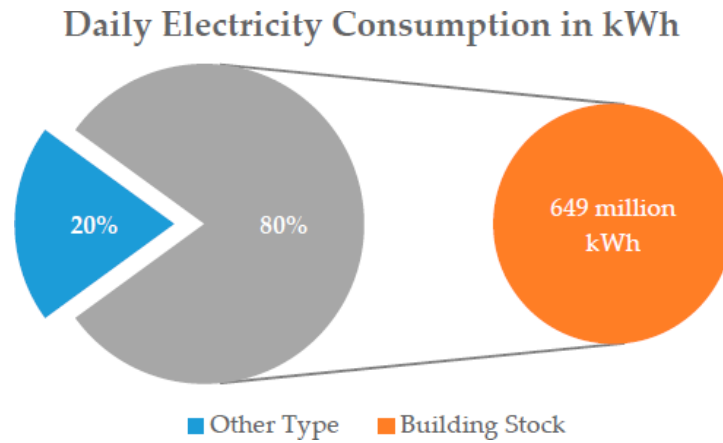


Figure 1-2: Electricity consumption by the building sector [13].

1.3 Mosques in Saudi Arabia

One of the major features of Saudi Arabia is the enormous number of mosques in the country. These mosques are designed to provide maximum human comfort for worshippers. Hence, HVAC systems are an integral part of virtually all the mosques in the country. Mosques in Saudi Arabia can be classified into two major categories based on who designs and constructs them. Some of the mosques are designed and built by the Ministry of Islamic Affairs, Endowments, Da`wah and Guidance (MIAEDG), while others are designed and built by private well-wishers. The mosques in the second category can be classified based on the number of occupants and the size of the mosque. They include small, medium, and large mosques, which are identified as musalla, mosque, and jamee, respectively. Classification of mosques into jamee, mosque, and musalla categories is based on a survey conducted by Alhamod et al. [14]. Musalla mosques can accommodate between 30 to 250 people, thus, this type of mosque is normally for daily prayer. Mosques with the classification of mosque can accommodate between 250-700 people and are frequently used for performing both daily and Friday prayers. Jamee is a mosque that can accommodate more than 700 people, which is used for Friday prayers, with a

section of the mosque used for daily prayers also. The characteristics of these mosques are similar in terms of the layout, shape, construction materials, the type of air-conditioning, and other energy systems used[14][15][16][17]. According to the MIAEDG annual reports, the number of mosques in Saudi Arabia in 2017 was about 99,000 with a total electricity bill of 700 million Riyals [16].

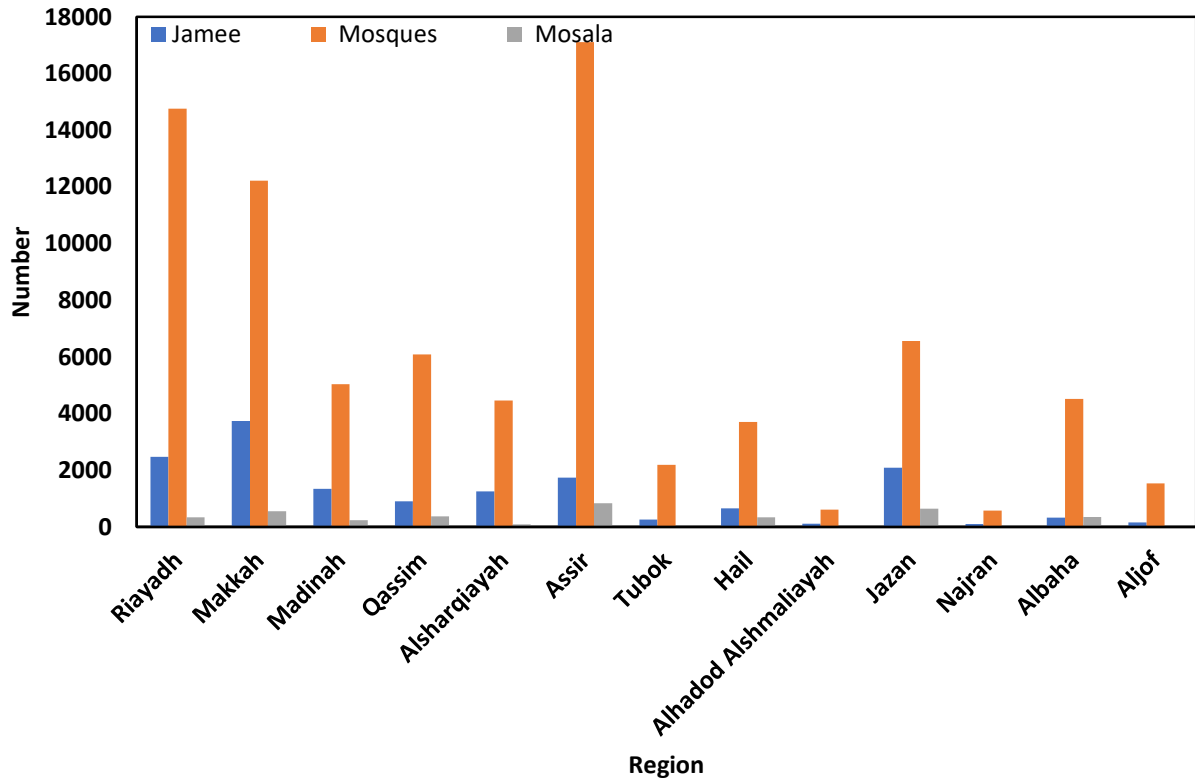


Figure 1-3: Distribution of mosques in Saudi Arabia [18].

The data presented in Figure 1-3 indicate that the region with the second highest number of mosques in Saudi Arabia is the country's capital (Riyadh), which will be the central focus of this study.

1.4 Thermal Comfort in Mosques

Due to the harsh hot climate of Saudi Arabia, mosques totally rely on HVAC systems to provide acceptable thermal comfort for the occupants. Different HVAC systems are used in mosques depending on the size of the mosque and its location. However, most of the mosques in Saudi Arabia use split or window AC systems or packaged systems. During the summer months, the HVAC systems typically operate the whole day to provide thermal comfort for the occupants. Only few studies on the thermal comfort requirements of mosques have been conducted. One of these is on the thermal comfort of worshippers during “Friday” prayer conducted in Riyadh by Al-Homoud et al [15]. The relatively high energy use by some mosques during the peak thermal load is not only due to providing better thermal comfort for the occupants, but also due to the un-insulated nature of the mosques. A comprehensive study has been conducted for six centrally air-conditioned mosques with ceiling-mounted air distribution systems in Kuwait characterized by a hot desert climate to evaluate the thermal conditions and the thermal comfort of the occupants. A total of 140 surveys were conducted to evaluate the thermal comfort of occupants and determine the environmental parameters. The mean values of the environmental parameters are as follows: (i) the mean indoor dry bulb temperature (IDBT) of air is 23 °C with a standard deviation (SD) of 0.53; (ii) the MRH is 44.19% with an SD of 0.85; (iii) the mean air velocity is 0.23 m/s with an SD of 0.08; (iv) the neutral operative temperature, from linear regression between the actual mean vote (AMV) and operative temperature, is 26.1 °C during prayer periods; (v) AMV is within -2.67 to 2.33 and the mean thermal impression of the occupants is 0.26; and (vi) the range of the predicted mean vote (PMV) is from 0.005 to +1.43 with a mean of +0.19. The mean air temperature in each mosque and its standard deviation are shown in Table 1-1 [15][19].

Table 1-1: The mean air temperature and standard deviation in each mosque [19]

Mosque	Mean Air Temperature (°C)	Standard Deviation
1	23.87	0.2
2	20	0.35
3	21.53	0.3
4	19.13	0.95
5	28	0.93
6	25.43	0.45

The results in Table 1-1 indicate that in most cases thermally comfortable conditions are not achieved, i.e., there is overcooling. Hence, intermittent operation of the HAVC system will improve the cooling efficiency and reduce the energy consumption for cooling, without compromising the quality of the indoor atmosphere.

1.5 Ablution Water in Mosques

Saudi Arabia is a country with a very high number of medium to mega size Mosques where a very large amount of water is consumed in public ablution rituals. Worshippers perform ablution before most of the religious rituals. Ablution is the act of washing selected parts of the body using clean water as shown in Figure 1-4. Normally, the ablution process is performed in facilities that have standard taps with a manual open/closure mechanism and typically they do not use any water saving features, such as aerators and motion sensors. Thus, a considerable amount of water is wasted during the ablution ritual as water will be continuously running from the opened tap while a worshippers are

washing their body parts [68]. Hence, treating the lightly polluted water and reusing it in the cooling process will have considerable economic and environmental benefits.

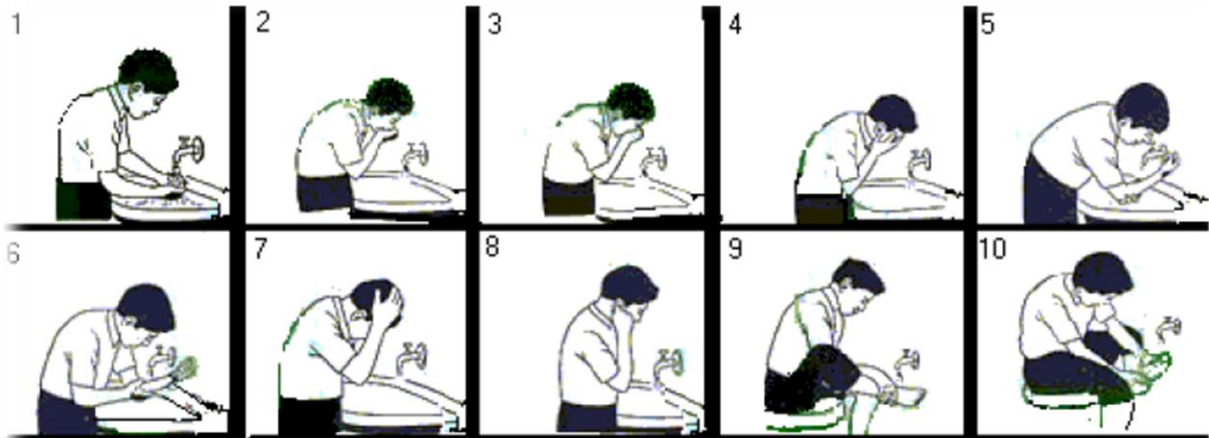


Figure 1-4: Sequence of the ablution process [67].

1.6 Climatic Zones in Saudi Arabia

Energy utilization, environmental performance of buildings and the climate conditions are closely related in terms of energy consumption and potential for the use of renewable energy [9]. Thus, investigating these conditions is essential for improving the energy performance of buildings, achieving building sustainability, and evaluating the potential for utilizing renewable energy. Identifying climatic zones with common climatic features is crucial, especially in Saudi Arabia where the variation of climatic characteristics is significant. Being the leading country in the Arab World Saudi Arabia is also a member of the Gulf Cooperation Council (GCC) Countries and the Middles East and North Africa (MENA) region. Many classifications of the climate of Saudi Arabia classify

the country as in either a desert or an arid region (i.e., as a single climatic zone). However, Alrashed [9] to identified the most appropriate climatic classification method by evaluating 16 different methods based on a weighted criteria assessment matrix. Saudi Arabia is classified into five inhabited distinctive climatic zones, namely, northern, eastern, western, southern, and central regions according to the resulting decision-matrix method. Northern region consists of Guraiat, Hail, Aljof, and Alhadod Alshmalayah. Eastern region consists of Dammam, Dhahran, Khobar, Jubail, Abqiq, and Ahsa city. Western region consists of Jeddah, Makkah, Madinah, and Tubok city. Southern region consists of Khamis Mushait, Assir, Jazan, Najran, and Albaha city. Central region consists of Riyadh and Qassim city. However, these regions are identified by the main cities in the regions, namely, Guraiat, Dhahran, Jeddah, Khamis Mushait, and Riyadh [9]. Figure 1-5 depicts the climatic zones and Table 1-2 shows the climatic parameters of the five climatic zones.

Table 1-2: Climatic parameters for various climatic zones [9].

Climatic Represented Location	Geographic coordinates			Air temperature			Relative humidity			Wind speed			Global Solar Radiation		
	Latitude (°N)	Longitude (°E)	Elevation (m)	Minimum (°C)	Maximum (°C)	Mean (°C)	Minimum (%)	Maximum (%)	Mean (%)	Minimum (m/s)	Maximum (m/s)	Mean (m/s)	Minimum (W/m ²)	Maximum (W/m ²)	Mean (W/m ²)
Dhahran	26.3	50.1	22	5.0	45.7	25.8	19	99	57	0	5.7	0.9	0	990	195
Gurait	31.3	37.4	502	-3.3	43.9	19.8	12	100	40	0	16.3	4.2	0	1053	235
Riyadh	24.7	46.8	583	2.2	43.7	25.1	10	91	32	0	11.9	3.1	0	1049	252
Jeddah	21.5	39.2	33	13.9	41.7	27.9	37	100	65	0	11.2	2.6	0	1035	257
Khamis Mushait	18.3	42.7	2051	2.7	34.3	18.9	17	100	51	0	12.4	3.1	0	1127	289



Figure 1-5: Climatic zones in Saudi Arabia [9].

The central region of Saudi Arabia, where the climate is hot and dry, is associated with a high sensible load and a low latent load. In regions with such climate, the need for cooling is high, while the need for dehumidification is not high. Thus, cooling based on the Maisotsenko-cycle (M-cycle), which is an efficient indirect evaporative cooling technology, is well suited for such climatic conditions. In addition, the abundant solar radiation can be effectively utilized to power this system, for which the energy requirement is low. Table 1-3 presents a comparison between conventional and M-cycle-based cooling systems [20].

Table 1-3: Comparison between a conventional cooling system and an M-cycle based AC system [21].

Factor	Vapor compression AC	M-Cycle based AC
Operational cost	High	Low
Energy source	Electricity, Natural gas	solar energy or any low-grade heat
Efficiency	Low	High
Air quality	Average	High
Installment of the system	Compact and easy	bulky
Effect on environment	Harmful	Eco-friendly

This study proposes to decrease the power consumption of mosques in Riyadh by introducing a novel method of cooling based on the Maisotsenko Cycle, which is not only cost effective but also saves energy and environment. This cooling system operates by combining the thermodynamic processes of dew point evaporative cooling with heat transfer from the latent heat of the evaporation of water using the available psychometric renewable energy. A mathematical model will be developed first to evaluate the effectiveness of the novel method of cooling in the proposed region. Thereafter, the results will be validated using published experimental results of previous studies.

1.7 Motivation and Significance

The world is facing a challenging time in fulfilling its energy requirements due to the depletion of the fossil fuel resources. In the Kingdom of Saudi Arabia, cooling systems account for about 50% of the total energy utilized by building sector, amounting to about 20% of the total energy consumption nationwide[8][9][10][11]. Many countries experience increasing in electrical energy demand during the hot months of the year use vapor compression air conditioning (VCAC) as the dominant method to meet their cooling needs. However, the refrigerants used in VCAC equipment are harmful to the atmosphere, in addition to the process being energy intensive. Thus, developing cooling system that is environmentally friendly and energy efficient becomes very important area of research. During these challenging times, this study is conducted to achieve the following:

1. Develop a sustainable air conditioning system for mosques to decrease the enormous amount of energy consumed by the large number of mosques.

2. Help sustain the limited amount of available energy resources for a longer duration by using them in an efficient manner.
3. Decrease the use of fossil fuel to protect the environment from the harmful gases emitted in their combustion.

1.8 Study Objectives

The overall objective of this study is to reduce energy consumption by AC systems using cooling technology based on M-cycle. In this study, the feasibility of using M-cycle based air cooling system will be evaluated to examine the system performance and positively contribute to reduce the emission of harmful gases to the atmosphere. The specific objectives of this study are the following:

1. Lower the energy consumption in mosques located in the hot and dry climate of the Kingdom of Saudi Arabia.
2. Predict the performance of the proposed cooling system using a mathematical model.
3. Evaluate the extent of water conservation by proposing a sustainable ablution greywater treatment system
4. Assess the economics of the proposed system and its feasibility.

1.9 Research Approach

The work described in the following sections will be used to achieve the stated objectives:

Phase-I. Literature Review:

1. Acquire a comprehensive overview of cooling systems based on the M-cycle.
2. Review studies conducted in the field of thermodynamics, building simulation, and its relevant design elements.
3. Gather information on the most common design practices used in the construction of the mosques, including the operation strategies, types of HVAC systems used, and electricity consumption.

Phase-II. Formulation and Validation of Base Case Model:

1. Model a mosque building in the hot and dry climatic zone of Saudi Arabia using EnergyPlus software.
2. Employ the characteristics of buildings found in the literature survey to model the base case of mosque structure.
3. Simulate the base case model and study the performance in terms of energy consumption and the cooling load of mosques.
4. Verify the energy pattern of the base case model using an actual data from the selected mosque.

Phase-III. Investigation of the Feasibility of Using an M-cycle-based HVAC

System:

1. Evaluate the performance and cost of an M-cycle-based HVAC system.
2. Verify whether the proposed HVAC system can handle the required cooling load of the mosques fully or partially.
3. Determine the overall energy reduction that the proposed system would achieve in the mosques.

Phase-V. Conclusions and Recommendations:

1. Provide conclusions and recommendations on a sustainable HVAC system that will reduce the energy consumption for a typical mosque, resulting in the decrease of harmful gas emissions.

CHAPTER 2

LITERATURE REVIEW

A detailed literature review to identify the gaps in the subject is presented in this chapter. Published studies in the following areas were reviewed: (i) the fundamentals of evaporative cooling systems, (ii) the principle upon which M-Cycle cooling systems operates, and (iii) review of the studies focusing on the M-cycle-based cooling system.

2.1 Air Conditioning Systems based on Evaporative Cooling

At present, the market is dominated by HVAC systems based on mechanical vapor compression refrigeration systems. Based on the working fluid that serves as the major cooling agent, refrigeration systems can be divided into two major categories, namely, absorption cooling (AC) systems and mechanical vapor compression (MVC) systems. However, air conditioning systems based on evaporative cooling (EC) are available currently[22][23]. AC systems based on refrigeration are characterized by their high energy requirement relative to the ECs, which consume about one-fourth the energy requirement of MVCs [21]. An MVC system consists of four main components, namely, a condenser, an evaporator, a compressor, and an expansion valve as shown in Figure 2-1 [7]. Among all these components, the compressor consumes the most significant amount of electrical power.

Although these air conditioning systems have clear advantages such as low cost, long life cycle time, good stability in performance, and reasonable good energy performance, as indicated in Table 1-3, they have major disadvantages such as high demand for electricity generated mainly using fossil fuels. Thus, these systems are neither sustainable nor environmentally friendly [29]. AC systems based on EC are more environmentally friendly compared to those based on MVC because they consume less fossil fuel, curbing the release of GHGs. The EC systems can be divided into direct ECs (DECs) and indirect ECs (IECs) [25]. In the former, air come into direct contact with the water and convert sensible heat of air into latent heat of water. The latter (IECs) can be further divided into systems based on a wet or dry surface [26].

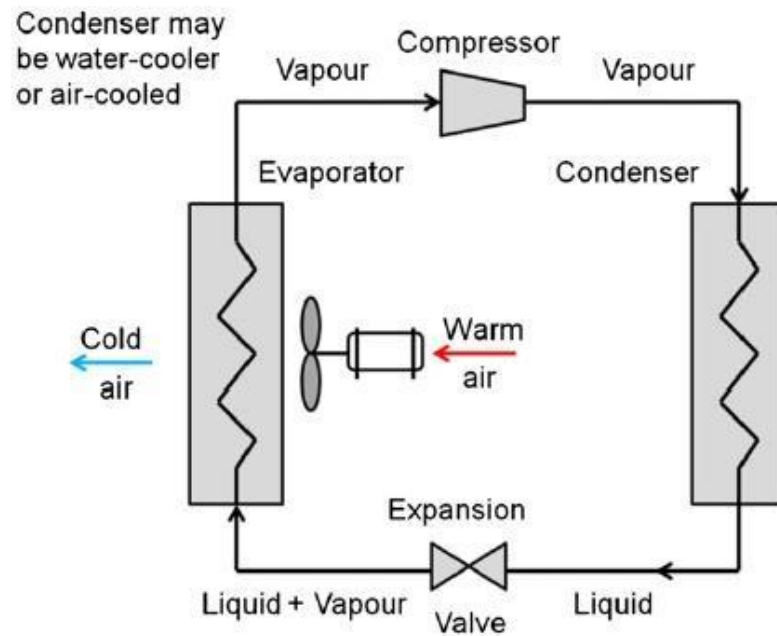


Figure 2-1: Conventional Vapor Compression Refrigeration Cycle [7].

The use of air conditioning systems based on evaporative cooling as an alternative to compressor based systems provides a solution to the high power consumption by the traditional air cooling systems, and hence, is more environmentally friendly [27]. Furthermore, as shown in Table 2-1, their coefficient of performance (COP) is much higher compared to other cooling systems.

Table 2-1: Values of the COP of some air conditioning system [28].

System type	Evaporative Cooling	Absorption /Adsorption	VMC Cooling	Thermoelectric Cooling
COP	15-20	0.6-1.2	2-4	0.2-1.2

Evaporative cooling is a process which utilizes the sensible heat of air to evaporate water. The substantial amount of heat that is transmitted from air to water in the evaporation process, significantly decreases the temperature of air. Although evaporative cooling systems have remarkable advantages, they are heavily dependent on the weather conditions. The wet and dry ambient air temperature difference is the driving force in this cooling process. This difference is at its highest value in hot and dry climatic zones. Hence, evaporative cooling systems are very efficient in such geographic regions. Figure 2-2 depicts the main types of the evaporative cooling systems [7][24][25][26][27].

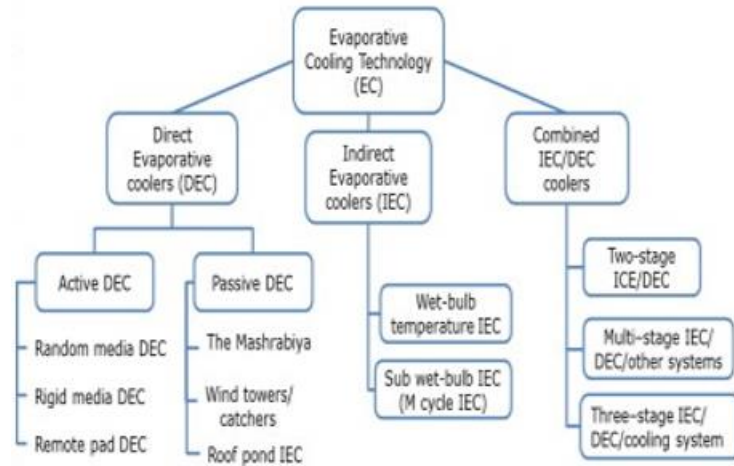


Figure 2-2: Types of systems based on evaporative cooling used in air conditioning applications [28].

2.1.1 Systems based on Direct Evaporative Cooling (DEC)

Direct evaporative cooling (DEC) is one of the simplest and oldest evaporative cooling systems. In this process, the intake air comes into direct contact with water, resulting in heat and mass transfer between the working fluids. Direct contact evaporative cooling is more suitable for desert regions, as rust and mildew can form on susceptible materials in humid climatic zones. In DEC systems, dry bulb temperature (DBT) of air is decreased, while its humidity is increased due to direct contact with water. The maximum achievable (theoretical) air flow temperature is the wet-bulb temperature of the inlet air [20][28]. Figure 2-3 shows a schematic representation of the basic structure of a DEC and its psychometric chart [29].

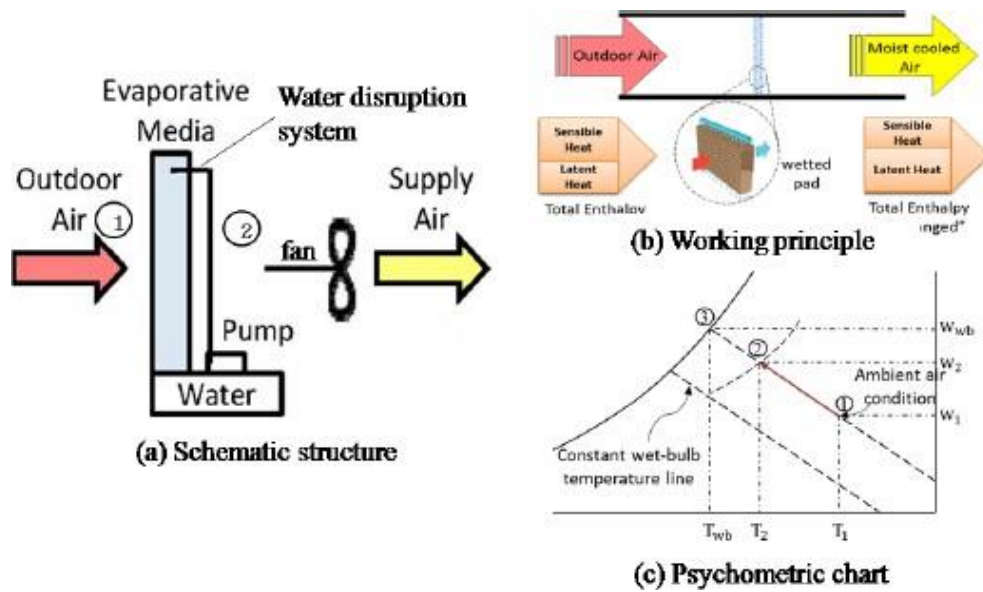


Figure2-3: Schematic representation of an air conditioning system based on direct evaporative cooling and its psychrometric chart [30]

2.1.2 Systems based on Indirect Evaporative Cooling (IEC)

Primary and secondary air streams of an indirect evaporative cooling (IEC), which consists of a dry channel and a wet channel, pass through the system as shown in Figure 2-4. The primary air stream flows through the dry channel, while the secondary air stream passes through the wet channel. Only heat transfer occurs in the dry channel, while both heat and mass transfer occur in the wet channel. Air in the primary channel cools, while the air in the secondary channel saturates due to these processes. IEC based systems are classified into two main types based on the flow direction, cross and counter flow. Studies reveal that the wet bulb effectiveness of IEC based systems ranges between 40% to 80%. The notable advantage of the IEC based systems is that moisture is not added into the air, thus, enhancing air quality. Owing to the poor heat transfer rate of IEC systems, which does not justify the manufacturing cost, these systems have not been commercialized[7][30][31][32][33][34].

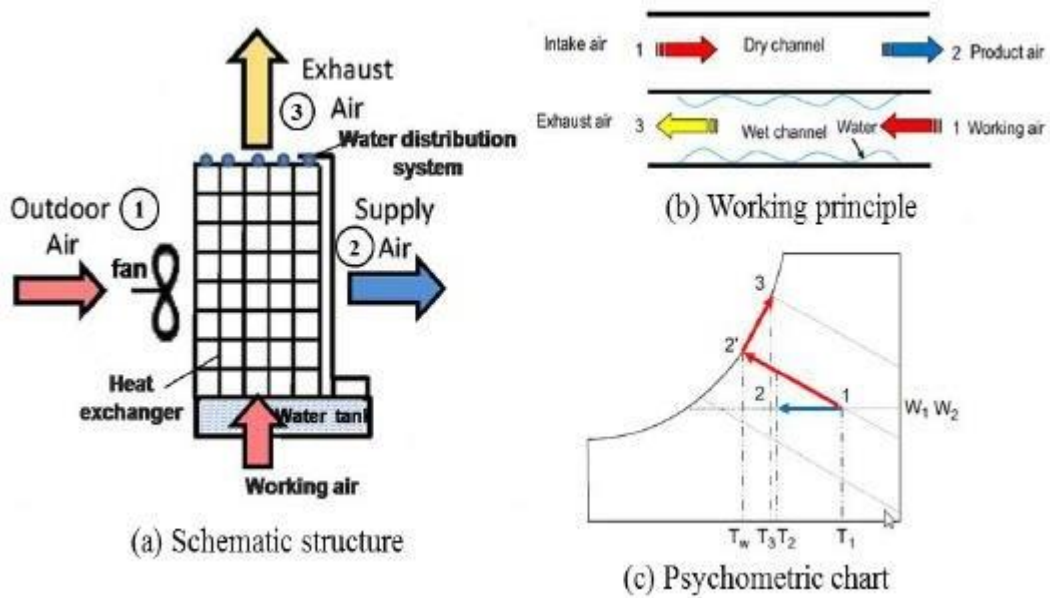


Figure 2-4: Schematic representation of an air conditioning system based on indirect evaporative cooling and its psychometric chart [8].

A novel IEC type cooling system with an improved wet bulb effectiveness and without direct contact between air and water, based on the M-cycle, has been introduced[35][36]. The cooling system based on the M-cycle is a modified IEC system with a portion of the primary air supplied as a secondary air stream, which further enhances the heat transfer with the air in the dry channel. The modified technique makes the primary air temperature approach the dew point temperature at the inlet, thus giving this method the technical name of dew point indirect evaporative cooling. The wet bulb effectiveness of the system ranges between 110% - 122% and its dew point effectiveness ranges between 55%-85%[37][38]. The M-cycle-based air conditioning systems have many advantages. The most important advantages are the following:

1. The temperature of the supplied air in the air conditioning system based on the M-cycle is lower than that supplied by both direct and indirect evaporative cooling systems and moisture is not added to the supplied air [39][40].
2. The possibility of causing any health issues by contaminated water is eliminated, as there is no direct contact between water and primary air [37].
3. The required water to run the cycle is reduced by 50% compared to conventional IEC systems [39].
4. The electrical power required to operate M-cycle based cooling systems is much less than that required to operate the traditional air conditioning unit. Thus, it can be supplied by solar energy using solar panels [38].
5. The air supplied by M-cycle based air cooling system is 100% fresh air [39].

2.2 Basic Principle of Maisotsenko Cycle

M-Cycle based cooling systems operate based on thermodynamics processes. In this method, ambient energy presents as psychometric renewable energy and it is captured in the form of latent heat of water, evaporating into the atmosphere in the process. The heat transfer and evaporative cooling processes are combined to ensure that the temperature of supplied air approaches the dew point temperature of the atmospheric environment. Hence, it is an advanced evaporative cooling system, which cools air down to its dew point temperature instead of its wet bulb temperature, as illustrated in Figure 2-5 [41][42][43].

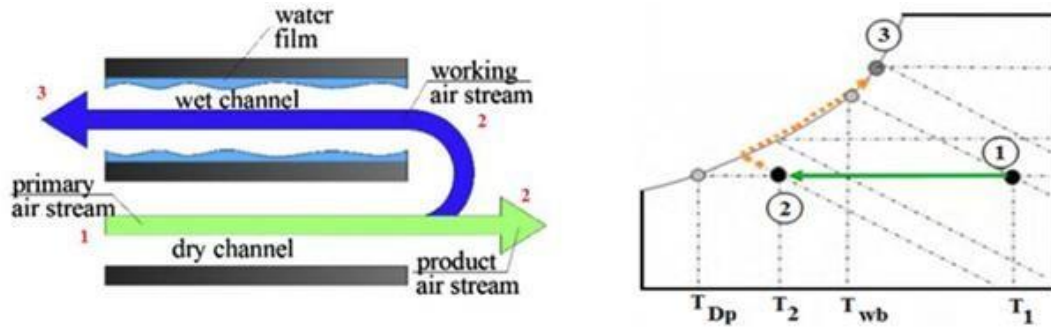


Figure 2-5: Schematic diagram of the working principle of an M-cycle cooling system and its psychrometric chart [44].

Generally, the cooling effectiveness of ECs is within the range of forty to ninety-five percent. The DEC systems are considered more efficient than IECs and are often utilized in hot and dry areas such as many places in Saudi Arabia, Australia, and China [26][44]. The cooling efficiency of DEC systems ranges from about seventy to ninety-five percent, while that of IECs is within the range of forty to sixty percent [6][45]. Hence, the major objective of researchers is to improve ECs to meet comfort standards in buildings (i.e., a temperature of twenty-five degrees and about fifty to sixty percent relative humidity), while reducing the global energy consumption [23][46]. Dew point evaporative cooling systems can be adopted to achieve both objectives of decreasing the energy consumption by buildings and achieving comfortable conditions for occupants. The M-Cycle is an example of a dew point EC. As the M-Cycle system cools ambient air to a temperature between its dew point temperature and the wet-bulb temperature with no humidity increase, it belongs to the wet surface category of IECs. Depending on the arrangement of the flow of intake and working air, M-Cycle systems can be categorized as parallel-flow, cross-flow, and counter-flow systems. The last two are the most famous arrangements [47][48].

2.3 Development of M-Cycle-based Air Conditioning Systems

Several studies have been conducted to evaluate the performance of M-cycle based cooling systems based on different methods of evaluation. The outcome of those studies as well as their key performance measures such as the dew point and wet-bulb efficiencies are summarized in Table 2-2 and 2-3. Moreover, these studies highlight the influence of factors such as the ambient air velocity, channel width, ambient air temperature, channel height, and water temperature. In this section, results of both experimental and numerical studies are presented.

2.3.1 Experimental Work

Recently, numerous experimental studies have been conducted to examine a range of performance parameters of the air conditioning system based on M-cycle. The most highly ranked and innovative papers that cover this topic experimentally are summarized in Table 2-2. Some of these studies will be discussed in more detail to highlight the most important studies.

Table2-2: Experimental Studies.

Study	Year	Method	Configuration	Research	Wet Bulb Effectiveness	Dew Point Effectiveness
Elberling [49]	2006	Experimental	Crossflow	Studied the performance of M-cycle based air conditioning system and evaluated based on air flow, cooling effectiveness, and power consumption.	81-91%	50-63%
Riangvilaikul et al. [50][51]	2010	Experimental	Counter-flow	- Examined the performance of the proposed system with different inlet air conditions. - Developed a mathematical code to predict the mass and heat transfer that occur within the cycle. - Optimized the system parameters.	92-114%	58-84%
Bruno [52]	2011	Experimental	Counter-flow	- Conducted experimental tests of the thermal performance of M-cycle based air conditioning systems for different types of buildings dominated by residential buildings - Applied the analysis for different operation conditions.	Residential 124%	Residential 106%
Boukhanouf et al. [53]	2013	Experimental	Counter-flow	Evaluated the system cooling capacities to predict the performance of the proposed system.	117%	N/A
Lee et al. [54]	2013	Experimental	Counter-flow	Evaluated the effectiveness of the proposed system for different inlet conditions.	118% - 122%	75-90%

Rogdakis [55]	2014	Experimental	Cross-flow	- Studied the performance of an M-cycle based air conditioning system based on Greek climate conditions. - The results show that the system can be applied for most Greek cities with low power requirements and water.	97% -115%	N/A
Jradi and Riffat [56]	2014	Experimental and Simulation	Cross-flow	Studied a system with a channel with dimensions of 500 mm in length and 5 mm in height.	112%	78%
Duan et al. [58]	2016	Experimental	Counter flow	Investigated the performance of the system by applying different inlet air conditions.	55%-106%	N/A

Rogdakis et al. [55] examined the efficiency of an M-cycle based air conditioning system for the Greek climatic zone experimentally and theoretically. They conclude that the system is very effective in most Greek cities. Moreover, the system consumes less electricity than the conventional system as well as it consumes less water than other evaporative cooling cycles. The effectiveness of the system lies between 97% and 115%, with water consumption ranging between 2.5 and 3.0 l/kW/h.

In addition, Riangvilaikul and Kumar [51] [52] have shown that using parallel rectangular channels would result in a dew point effectiveness and wet-bulb effectiveness between 65% - 86% and 106%–114%, respectively. In order to achieve a wet bulb effectiveness greater than 100%, the velocity of inlet air should be less than 2.5 m/s, the gap between the channels should be less than 5 mm, the height of the channels must be greater than 1.0 m, and working-air-to-inlet-air ratio must be between 35 % and 60%. The results show that the proposed design and operating conditions are optimized.

Duan et al. [58] have investigated the influence of several parameters on the overall efficiency of M-cycle based cooling system. They studied the system with a range of operating conditions involving different feed water temperatures and evaporation rates. The experimental results show that the wet bulb effectiveness ranges from 55% to 106%. Furthermore, wet bulb effectiveness can be enhanced by increasing either the inlet wet bulb depression or the working-air-to-supply-air ratio. Moreover, the cooling capacity can be increased by increasing the intake air velocity or inlet wet-bulb depression or by reducing the working-air-to-supply-air ratio.

Riffat and Jradi [56] have investigated the efficiency of a cross flow M-cycle system numerically and experimentally. The results show that the dew point effectiveness and the wet bulb effectiveness of the proposed system are 78% and 112%, respectively.

Lee et al. [66] and Bruno [64] have done many modifications on the geometries of the proposed channel which resulted in wet bulb effectiveness as high as 122% and dew point effectiveness of 106%.

2.3.2 Numerical Studies

Recently, several numerical studies have been conducted to examine the most important performance measures of M-cycle-based air cooling system. The most highly ranked and innovative studies are summarized in Table 2-3. Some of the investigated studies are discussed in more detail to highlight the most significant results.

Table2-3: Numerical Studies.

Study	Year	Method	Configuration	Research	Wet Bulb Effectiveness	Dew Point Effectiveness
Zhao et al. [57]	2008	Simulation	Counter-flow	- Proposed a mathematical model for novel counter-flow M-cycle-based system. - Optimized the channel dimensions for different operating conditions.	54-130%	36-82%
Riangvilaikul et al. [50][51]	2010	Simulation	Counter-flow	- Developed mathematical code for the proposed system to describe the mass and heat transfer phenomena within the channels. - Optimized the proposed system parameters.	92-114%	58-84%
Hasan [66]	2010	Simulation	Four different configurations	- Mathematically modeled the heat and mass transfer processes which describe the physics of the proposed system. - Analyzed different configurations of the system and their performance characteristics.	109-131%	N/A
Zhan et al. [59]	2011	Simulation	Cross flow and Counter-flow	- Compared both counter-flow and cross flow M-cycle-based air conditioning systems. - Evaluated the performance	Cross flow: 114% Counter-flow: 136%	Cross flow: 76% Counter-flow: 91%

				indicators, such as the dew point effectiveness and wet-bulb effectiveness and the system cooling capacity under similar operating conditions.		
Cui et al. [64]	2014	Simulation	Counter-flow	- Evaluated the efficiency of the proposed air conditioning system under several operating conditions. - Studied the effect of channel surface roughness on the overall efficiency of the system. - Examined the effect of using return air as working air and determined its impact on the overall efficiency.	122-132%	81–93%
Khalid [61]	2017	Simulation	Counter-flow	Studied the effect of a range of inlet conditions, such as the inlet humidity, air temperature, feed water temperature, velocity, and working air ratio on an M-cycle cooling system.	100-120%	70-80%
D Levchenko et al. [62]	2017	Simulation	Counter-flow	Examined the effect of the inlet operating conditions, such as the air velocity and the results recommend the following values based on the proposed geometry: (i) velocity lower than 1.5 m/s, (ii) the channel height less than 10 mm, and (iii) the channel length 200 times the working channel height.	N/A	N/A

Zhao et al. [57] investigated a novel counter flow M-cycle based cooling system. In their design, holes are located at end of the proposed channels as illustrated in Figure 2-6, unlike the cross flow M-cycle heat exchanger. The ambient air passes through the dry channel while discharging sensible heat into the wet channel. A portion of the cooled air is supplied to the conditioned area, while the remainder is diverted to the wet channel to act as a secondary air flow. Secondary air flow removes sensible heat from primary air or the supply air in the dry channel, transferring latent heat to the water in the wet channel. The results show that the wet bulb effectiveness can reach 130%, while the dew point effectiveness rises to 90%.

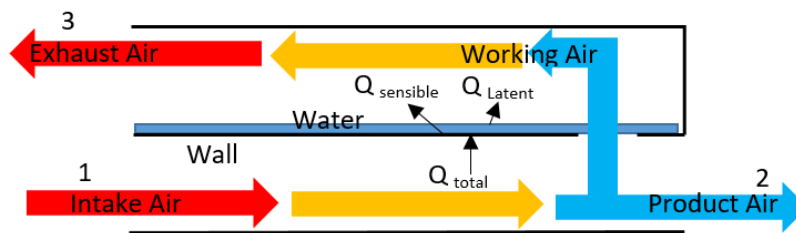


Figure 2-6: Basic structure of the design used by Zhao et al. [35].

Zhan et al. [59] performed a numerical analysis to compare counter flow and cross flow M-Cycle based cooling systems. They considered cross flow systems as that have parallel stud supporters while counter flow systems that constructed with triangular supporters. The results indicate that the cooling capacity of the counter flow heat exchanger is 20% higher than that of the cross flow heat exchanger. The counter flow heat exchanger has 15–23% higher dew point effectiveness and wet bulb effectiveness for similar operating conditions.

Zhao et al.[62]numerically evaluated the Maisotsenko cycle assuming United Kingdom summer conditions and using triangular counter flow arrangement for the cooling system. The wet bulb effectiveness and the dew point effectiveness of the system are 130% and 90%, respectively, and the water consumption is 2.1 - 3 liters/kWh. Furthermore, the cooling capacity of the studied system is expected to be between 3.1 - 4.3 W/m³/hour of air flow. The results indicate that the proposed system is promising for desert and mild climatic zones.

Khalid et al. [61] theoretically evaluated a counter flow M-cycle system and investigated the effects of several operating factors on the system performance. The factors evaluated are inlet air temperature, velocity, humidity, feed water temperature, and the working-air-to-supply-air ratio. The results indicate that the system is appropriate for desert climatic zones where the temperature is in the range of 45°C and the inlet air humidity is less than 12.7 g/kg.

Cui et al. [64] numerically evaluated a unique cooling system based on evaporative dew point principle with a closed loop arrangement in the counter flow mode with individual working and product channels. The results show that the wet bulb effectiveness and dew point effectiveness of the system are 122%–132% and 81%–93%, respectively. Furthermore, the predicted effectiveness is obtained when the supplied-air-to-working-air ratio is less than 1.5 and the inlet air velocity is less than 1.5 m/s with the height of the product channel less than 10 mm and the length of the channel 200 times the channel height [65].

Bakeem et al. [63] have conducted a feasibility study of using a novel counter flow M-cycle based cooling system that consists of fourteen channels for Arab Gulf cities. They developed heat and mass transfer model for the system and found that the power required by the system is less than that of the conventional air conditioning system with a similar cooling capacity. Moreover, the output of the proposed system is better than that provided by cooling systems based on DEC and IEC. This system is suitable for all areas of the Arab Gulf region.

To sum up, the literature review indicates that the M-cycle based cooling systems with a counter flow arrangement is the most popular type used for cooling purposes. Hence, M-cycle based air cooling systems are potential alternative cooling systems, especially in Arab Gulf countries. The results show that the highest wet bulb effectiveness is 136%, while the highest dew point effectiveness is 93% [47][63]. Thus, such a system can be strongly recommended as an alternative for conventional HVAC systems especially in countries that are located in hot and dry climatic zone.

2.4 Ablution Grey Water Treatment System

In this section, a literature review is presented to select the most appropriate greywater treatment system that can be utilized in this application to achieve the goal of this study. The details of the literature review are organized as follows: estimated volume of water used in the ablution process, characteristics and health aspects of reusing ablution water and review of the studies focusing on Ablution Grey Water Treatment System.

2.4.1 Estimated Volume of Water used in the Ablution Process

The ablution process normally takes several minutes with running clean water. The average ablution quantity per person is between 2.5 and 4.5 liters [68]. The volume was estimated to be about 5.0 litres from a field survey conducted by Suratkon [75], in which the number of prayers were monitored together with the ablution process in number of mosques in Malaysia at different prayer times of the day. Moreover, Abu-Rizaiza [68] conducted a comprehensive survey and the result shows that in 2002 about 32,403 m³/day of ablution greywater (AGW) was wasted. As data on wasted ablution water is not available for 2019, the data reported for 2002 summarized in Table 2-4 was extrapolated based on the annual growth rate of mosques to estimate the wasted ablution water in 2019. The number of mosques increased from 32,781 to 101,108 from 2002 and 2019 with an annual growth rate of 6.85%. Thus, the average estimated wasted ablution water in 2019 is 11.44 mL/s. Furthermore, Al-Harbi [71] has estimated the rate of production of ablution greywater by collecting samples from a mosque located in Qassim in 2019. The amount of ablution greywater collected during a single prayer was divided by the total number of worshippers to calculate the rate of production of ablution greywater (L/prayer/person). The average production rate was 11 mL/s which is in very good agreement with the estimated ablution greywater production in 2019 that was calculated utilizing the 2002 data with an error of 4%. Moreover, based on the water bill for the Hekmah mosque, an average of 422,000 liters of water is consumed in both toilets and ablution activities. Assuming that 30% of the wasted amount is from the ablution process, the rate of production of ablution greywater would be 195 L/h [69]. Hence, treating this amount of wasted greywater to be used in the cooling process of a mosque would result in tremendous environmental and economic benefits.

Table 2-4: Nationwide ablution water consumption [68].

Location	# of mosques	Average Mosque Area (m ²)	Total Area (m ²)	Number of people attending prayers	Number of prayers conducted/day	Number of Ablutions/day	Ablution Water Quantity in 2002 (m ³ /day)	Ablution Water Quantity in 2019 (m ³ /day)
Cities	10,916	590	6440440	966,066	4,830,330	1,449,099	16,906	33,669
Villages	21,865	270	5903550	885,533	4,427,663	1,328,299	15,497	66,274
Total	32,781	760	12343990	1,851,599	9,257,993	2,777,398	32,403	99943

2.4.2 Characteristics and the Quality of Ablution Water

Knowledge of the concentration of primary contaminants in raw ablution greywater is the key in its characterization and the average concentrations are shown in Table 2-5. Average values of pH, turbidity, TSS, COD, and BOD are 7.9, 14.8 (NTU), 24.9 (mg/L), 63.2 (mg/L) and 37 (mg/L), respectively. Average value of turbidity is higher than the value of 10 NTU specified in KSA reuse standard for all greywater reuse applications [69] [70]. The value of BOD at 10 mg/L exceeds the value specified in the standards for unrestricted irrigation, toilet flushing etc. Average value of COD (63.2 mg/L) exceeds the value specified in the KSA standard (50 mg/L) for all reuse applications. Nevertheless, TDS levels are acceptable for all potential reuse applications. However, levels of fecal coliform (FC) are higher than that specified in the standards on reuse for unrestricted irrigation (Standard A), which includes the reuse for cooling systems. The overall water quality indicates that raw ablution greywater must be treated prior to reuse [70].

Table 2-5: Quality of ablution greywater generated in Al-Ajaji mosque and the limits of parameters specified in the KSA standard for recycling greywater [69] [70].

Parameter	Unit	n	Mean (Min-Max)	STD	Re-use Standard		Acceptable	
					A ¹	B ²	A	B
pH		10	7.9 (7.7-8.1)	0.1	6.0-8.4	6.0-8.4	Yes	Yes
TSS	mg/l	9	24 (19.0-35.0)	6.1	10	40	No	Yes
alkalinity	mg/l	10	77.1 (67.2-86.0)	8.4	-	-	-	-
turbidity	NTU	10	14.8(11.2-18.1)	2.1	5	5	No	No
COD	mg/l	10	63 (38-88)	19.1	-	50	No	No
BOD	mg/l	9	37 (19 - 51.0)	12.6	10	40	No	Yes
TDS	mg/l	10	213 (192-245)	17.5	2500	2500	Yes	Yes
Oil & grease	mg/l	10	0.004 (0.001-0.009)	0.003	nil	nil	Yes	Yes
PO ₄ -P	mg/l	10	0.2 (0.07-0.28)		-	-	-	-
NH ₄ -N	mg/l	10	0.34 (0.27-0.44)	0.1	-	-	-	-
Free Cl ₂	mg/l	10	0 (0.0-0.0)	0.0	0.2	0.5	-	-
Fecal coliform	MPN/100 ml	11	884 (756-982)	74.4	0.2	1000	No	Yes

¹A= KSA Standard for unrestricted irrigation, toilet flushing, street cleaning and firefighting

²B= KSA Standard for restricted irrigation

2.4.3 Proposed Systems

Mohammad Ibrahim [82], developed a plumbing and ablution water reuse system to collect, filter, and disinfect ablution water to be reused subsequently for flushing of toilets or the irrigation of public gardens in a mosque located in Riyadh. He investigated the effect of the treatment on microbial and chemical quality of the recycled water. The system consisted of: (A) four PVC sand filters, one activated charcoal filter, an ultraviolet (UV) unit, tanks for collecting greywater and final purified water, valves, and regulators as illustrated in Figure 2-7. The output of the system was analyzed all year round and the results show that the concentration of all ions is similar to that of tap water, especially after adding UV unit. Moreover, the chemical composition of the treated water was very close to that of tap water with safe microbial content.

Therefore, the reused water can be safely used for unrestricted irrigation purposes which includes the cooling systems.



Figure 2-7: Sand and activated charcoal filters [82].

Prathapar et al. [73] developed a system with dual plumbing to separate greywater from black water followed by an ablution water treatment system. In their system the ablution water passes through a sand trap and collects in a collection tank. Then the ablution water is fed into a filtration unit, which is followed by a chlorination chute, and the treated water is collected in a collection tank. In their system, feeding of ablution water into the treatment system and release for reuse are automated using submersible pumps and timers. Even though most of the water quality parameters were within acceptable limits, pathogenic parameters such as the coliforms and *E. coli*, were above acceptable limits. Hence, ablution water treated using this system needs

to be further filtered and chlorinated to be ready for domestic uses without compromising the health of users.

Suratkon et al. [74] proposed a Smart WUDHU system to collect and filter ablution water at a university mosque in Malaysia. The system contains a simple system designed to collect, treat, and reuse ablution water for recycling. The system supplies treated greywater (TGW) for both potable and non-potable uses, including toilet flushing, general washing, watering plants as well as drinking. They employed filtration to remove sediments and other fine particles, followed by pumping the filtered greywater to a raised storage tank where sedimentation and disinfection via chlorination took place. Subsequently, the water was recycled for various purposes other than human consumption, as shown in Figure 2-8. The results of the analyzed samples show that all the quality parameters were in compliance with permissible levels of the Malaysian National Drinking Water Quality Standards.

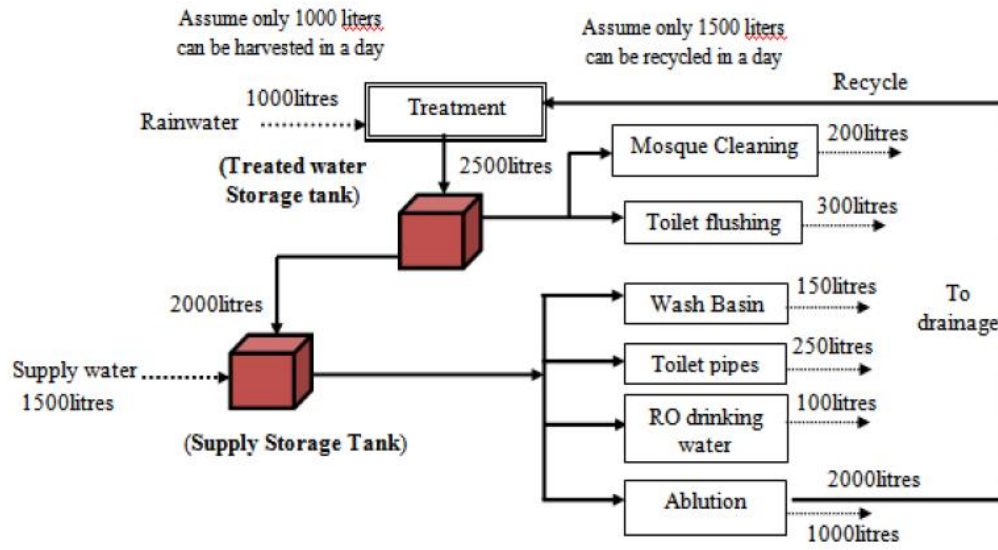


Figure 2-8: Proposed water circulation network of a University mosque, incorporating the Smart WUDHU system [74].

Five ablution greywater treatment options employing low cost ceramic filtration, coagulation with alum and adsorption using activated carbon (AC) were evaluated by Alharbi [70]. The five treatment options are the following: coagulation-sedimentation (CS), coagulation-sedimentation-AC adsorption (CSA), coagulation-sedimentation-ceramic filtration (CSF), direct ceramic filtration (DF), and filtration-AC adsorption (FA). All the options were investigated through monitoring of the quality of the treated water and an economic analysis. Moreover, the treatment efficiencies of these five treatment options were examined and monitored experimentally. The results indicate that filtration combined with adsorption, i.e., FA, is the most feasible option to treat greywater for unrestricted reuse for irrigation, toilet flushing, reuse in cooling systems, street cleaning and pedestrian area cleaning, and firefighting, provided that a disinfection stage is added. The treated water quality is compared to the maximum pollution limits for unrestricted application as per the wastewater reusing standards in Figure 2-9.

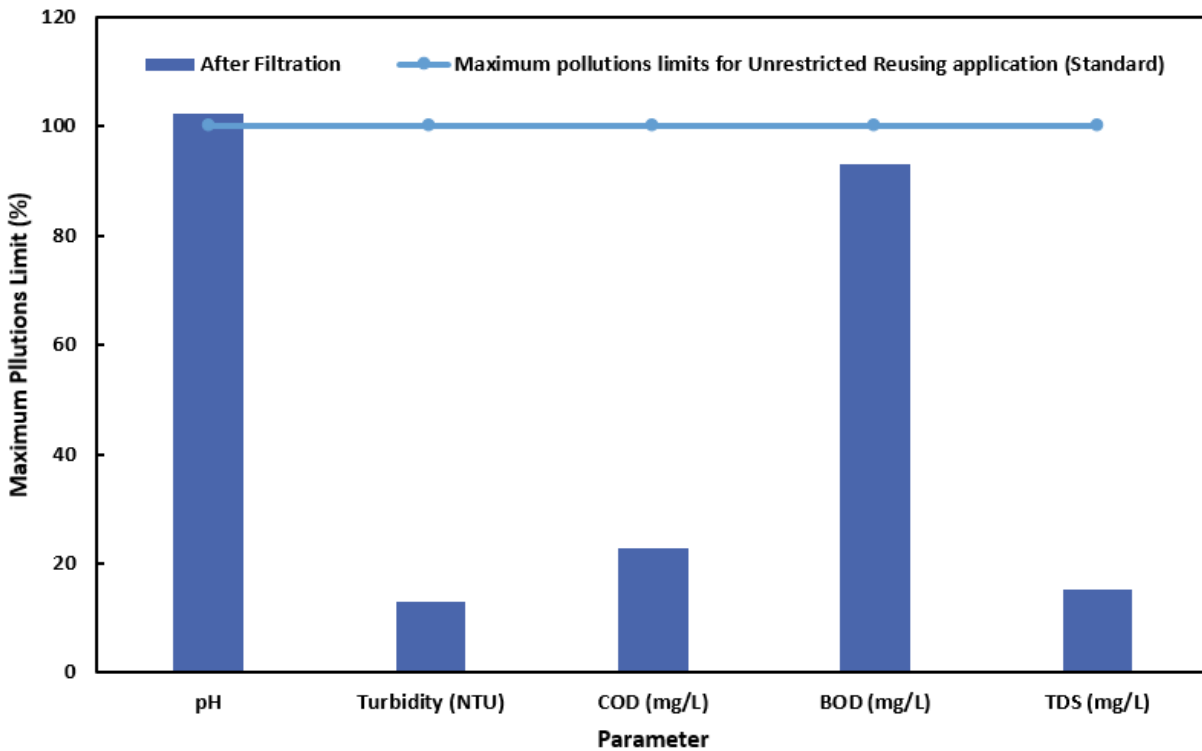


Figure 2-9: The technical performance of FA based greywater treatment system.

To sum up, the raw ablution water is very lightly polluted greywater, which can be treated to obtain water of acceptable quality. Therefore, the ablution and toilet areas must be redesigned to separate the greywater and black water by a proper plumbing design as shown in Figure 2-10. Then, ablution greywater treatment system based on filtration and activated carbon adsorption (FA) will be implemented to treat the collected ablution greywater to be used in the proposed cooling system. The treatment system will include a ceramic filter, followed by an activated carbon absorption column. Additionally, the system will include disinfection by chlorination and UV treatment,

which is an effective method for removing bacteria from water as illustrated in Figure 2-10. The treated and untreated ablution water can be circulated in a controlled manner using a fully automated system consisting of sensors and pumps. The treated ablution water must comply with the permissible levels specified in the Saudi national drinking water quality standards. The ablution water generation is variable and the peak value is expected during Friday prayer. Thus, the treated water collection tank and its connected piping system is designed with a level sensor to accommodate a situation where the level of water is always above the level that provides the required minimum quantity needed to run the cooling unit as shown in Figure 2-10.

2.5 Summary

A detailed literature review on the M-cycle based cooling system with different configurations along with an ablution greywater treatment system were performed. The literature review indicates that treated ablution water has been reused for irrigation and flushing toilets. However, the reuse of treated ablution water for cooling systems for mosques has never been considered and hence this study is original. An integrated M-cycle based cooling system coupled with an ablution greywater treatment system would lead to a considerable environmental and economic benefits. The current study proposes and demonstrates the integration between M-cycle based cooling system and an ablution water treatment system for mosques as depicted in Figure 2-10.

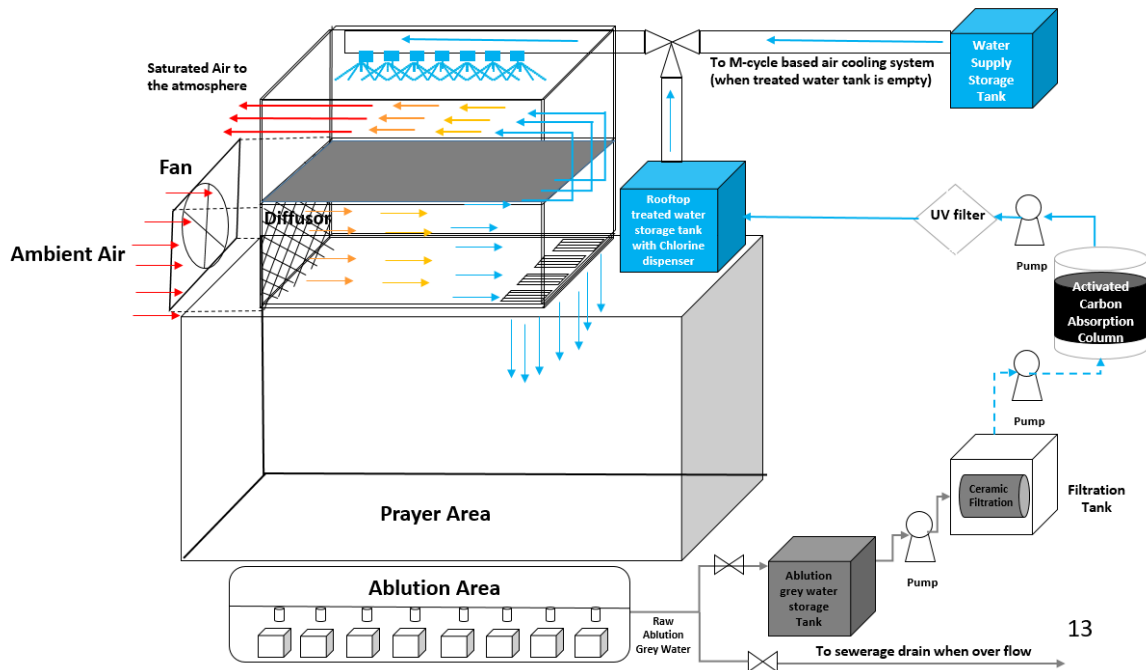


Figure 2-10: Conceptual model of an integrated greywater based cooling system for mosques.

CHAPTER 3: MATHEMATICAL MODELLING

This chapter presents the modeling of a counter flow M-cycle based cooling system, where the governing heat and mass transfer equations are obtained and the numerical method used to solve the equations is introduced. Furthermore, the cooling of the selected mosque is simulated using real data to validate the accuracy of the simulation.

3.1 Model Development

Figure 3-1 schematically depicts the operation of an M-cycle system in which ambient air flowing in the dry channel loses its sensible heat in the wet channel at State (1). The air at State (2) is cooled until the temperature reaches a value closer to its dew point temperature without changing its humidity and it is split into working and supplied air streams. The supplied air is used for cooling purposes. Heat and mass transfer occur during evaporation of some of the water sprayed into the working air injected into the wet channel. Typically, the temperature of the working air exiting at State (3) will be closer to the intake temperature and the relative humidity of the air will be closer to 100%. The flow chart of the model is illustrated in Figure 3-2.

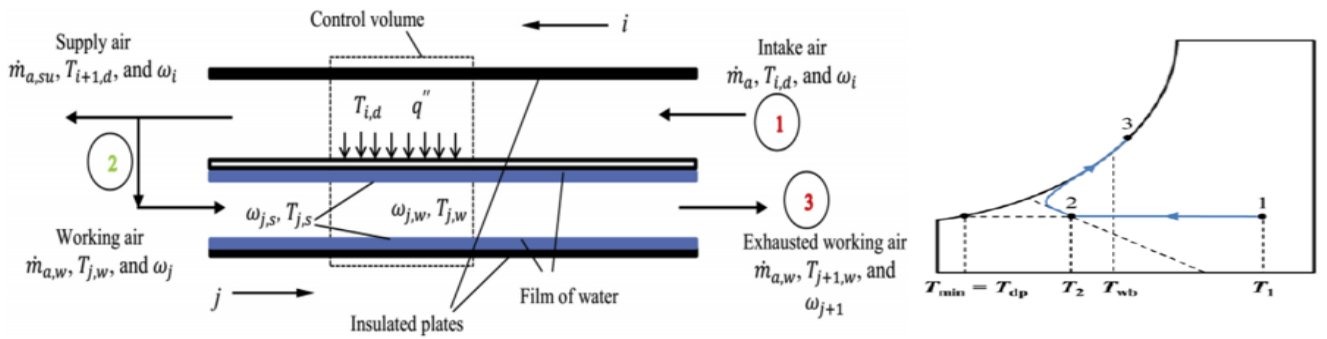


Figure 3-1: Schematic depiction of the principle of an M-cycle cooling system and the psychrometric chart.

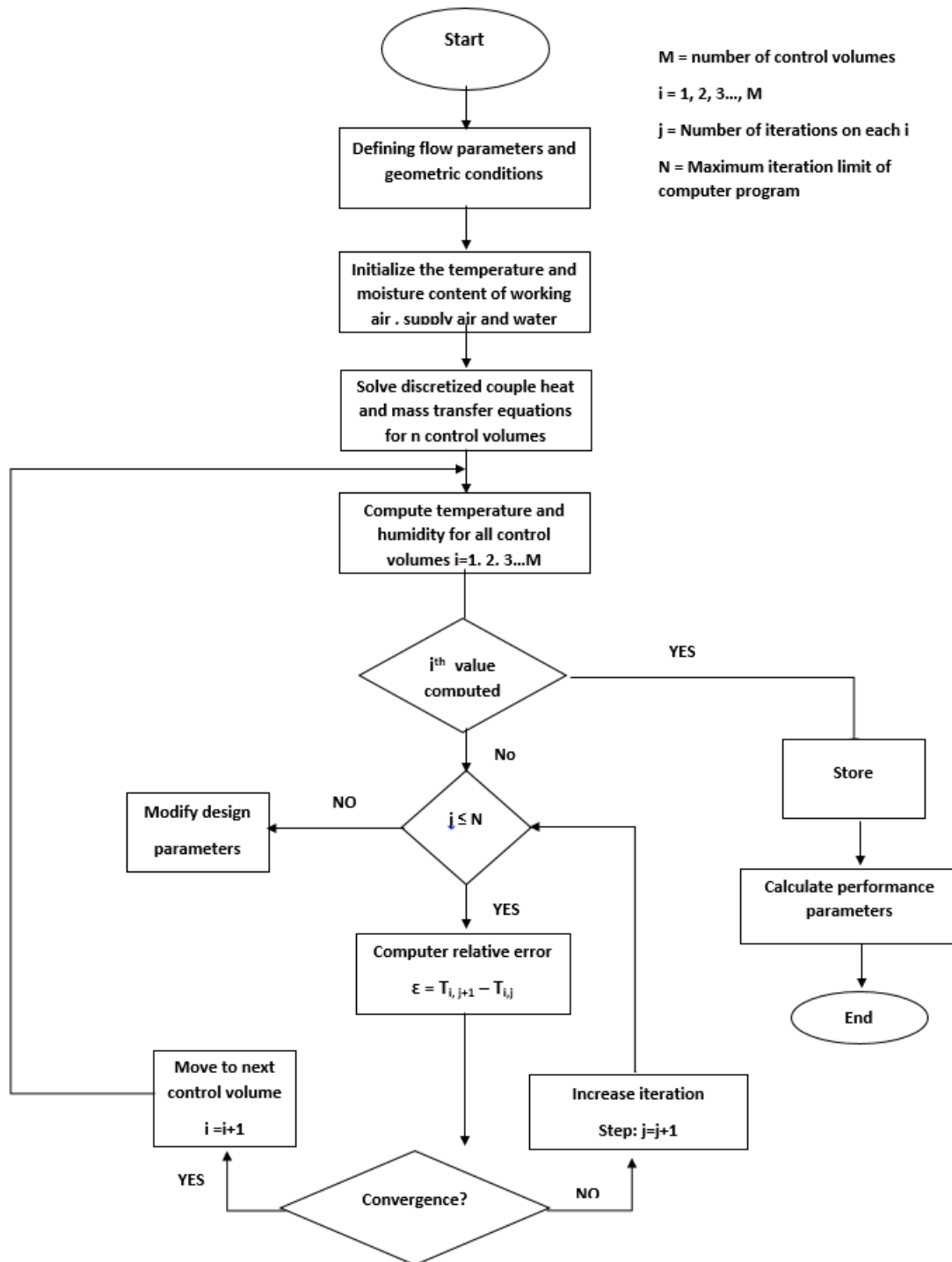


Figure 3-2: Methodology diagram.

Simultaneous steady state energy and continuity equations govern the flow in the heat exchanger. Figure 3-1 shows the differential control volume used for deriving the conservation equations. The following assumptions were made based on previous studies describing similar systems to develop the model:

1. Flow is steady and incompressible and there is no internal heat generation in the channels.
2. The system is well insulated and hence, heat transfer between the device and the surrounding space does not take place.
3. Channel height is negligible compared to the width of the channel. Thus, the system can be assumed to be one dimensional, which allows the temperature changes normal to the flow to be considered as negligible.
4. Viscous dissipation and body forces are negligible.
5. The flow inside the channels is laminar and fully developed.
6. The mass flow rate in each channel is constant.
7. The specific heat of water vapor and dry air does not vary appreciably and is constant.
8. The coefficients of heat and mass transfer too do not vary appreciably and are constant.
9. The temperature of the wall and the water film are similar and thus are equal.
10. The humid air in the wet channel is treated as an ideal incompressible gas mixture.

Given the stated assumptions, the conservation of energy equation in the dry channel along the x direction for the control volume shown in Figure 3-1 can be written as indicated in Equation

(3-1).

$$-\dot{m}_d c_d \frac{dT_d}{dx} = UW(T_{wf} - T_d) \quad (3 - 1)$$

Where U is the overall dry channel heat transfer coefficient, which can be calculated using Equation (3-2) and c_d is the specific heat of dry air.

$$U = \left(\frac{1}{h_d} + \frac{t_{plate}}{k_{plate}} + \frac{t_{paper}}{k_{paper}} \right)^{-1} \quad (3-2)$$

Where h_d is the dry channel convective heat transfer coefficient, k_{plate} is the thermal conductivity of the plastic plate and k_{paper} is the thermal conductivity of the wicking paper.

Similarly, the conservation of energy equation applicable to the wet channel is given by Equation (3-3).

$$\dot{m}_w c_w \frac{dT_w}{dx} = h_w W (T_{wf} - T_w) \quad (3-3)$$

Where c_w is the water vapor specific heat, and h_w is the heat transfer coefficient in the wet channel. Likewise, the conservation of mass for the differential control volume in the wet channel can be expressed by Equation (3-4).

$$\dot{m}_w \frac{dT_w}{dx} = h_m W (\omega_{wf} - \omega_w) \quad (3-4)$$

Where h_m is the wet channel mass transfer coefficient.

Furthermore, the conservation of energy near the wall between the wet and dry channels is given by Equation (3-5).

$$\begin{aligned} h_w W (T_w - T_{wf}) + h_d W (T_d - T_{wf}) + H_{w,fg} h_m W (\omega_w - \omega_{wf}) + k_{plate} W L \frac{d^2 T_{wf}}{dx^2} \\ = 0 \end{aligned} \quad (3-5)$$

Where $H_{w,fg}$ is the latent heat of water and h_m is the mass transfer coefficient.

Moreover, the Antoine equation (Equation 3-6) is used to calculate the dew point temperature of the ambient air at a given pressure [77].

$$T_{\text{dew}}(\text{K}) = \frac{3984.92}{16.5699 - \ln(p(\text{kpa}))} + 39.724 \quad (3 - 6)$$

Furthermore, Equation (3-7) is used to calculate the wet bulb temperature of the air according to reference [77].

$$T_{\text{wet}}(^{\circ}\text{C}) = 2.265 \sqrt{1.97 + 4.3T(^{\circ}\text{C}) + \omega 10^4 - 14.84} \quad (3 - 7)$$

The performance of the M-cycle based cooling system can be estimated using the dew point effectiveness which is estimated by dividing the temperature difference along the dry side by the difference in the dew point temperature of the inlet dry bulb and the inlet as expressed by Equation (3-8).

$$\varepsilon_{\text{wb}} = \frac{T_1 - T_2}{T_1 - T_{1,\text{wb}}} \quad (3 - 8)$$

Where ε_{wb} is wet-bulb effectiveness, $T_{1,\text{wb}}$ is the wet-bulb temperature at the inlet of the dry channel, and T_1 and T_2 which are the inlet and supply temperatures of the dry channel, respectively.

Moreover, the evaporative cooling performance can also be obtained using the expression in Equation (3-9).

$$\varepsilon_{dp} = \frac{T_1 - T_2}{T_1 - T_{1,dp}} \quad (3 - 9)$$

Where ε_{dp} is the dewpoint effectiveness, $T_{1,dp}$ is the dew point temperature at the inlet of the dry channel, and T_1 and T_2 are the inlet and supply temperatures of the dry channel respectively.

The COP of the system is defined by Equation (3-10) [78] [80].

$$COP = \frac{Q_{cooling}}{\dot{W}} \quad (3 - 10)$$

Where $Q_{cooling}$ is the cooling capacity of the system and is expressed by Equation (3-11) [78][80].

$$Q_{cooling} = m_{d,su} c_d (T_{d,2} - T_{d,1}) \quad (3 - 11)$$

Where $m_{d,su}$ is the mass flow rate of supplied air. The energy utilized by the cooling system (W) is defined by Equation (3 – 12) which is the sum of the energy consumed by the board, pump, fan, thermostat and the solenoid provides the electrical energy consumed by the cooling system [86, 87]. As the energy consumed by the board, pump, thermostat, solenoid, and soap is about 5% of the energy consumed by the fan [75], they can be neglected. The electric energy consumption of the fan is expressed by Equation (3-12) [75].

$$W_{fan} = W_{fan,base} \left(\frac{\rho}{\rho_{base}} \right) \left(\frac{\dot{V}}{\dot{V}_{base}} \right) \left(\frac{SH}{SH_{base}} \right) \quad (3 - 12)$$

Where $\dot{V}$ is the volumetric flow rate through the fan, ρ is the air density, and SH is the static head of the fan. As the air density do not change significantly, density ratio in Equation (3 – 12) is almost 1.

Static head of the fan can be expressed by Equation (3 – 13).

$$SH = f x \Delta P \quad (3 - 13)$$

Where f is a safety factor for both the base case and the operating case [75] [76]. Hence, the energy consumed by the fan can be expressed by Equation (3-14).

$$W_{fan} = W_{fan, base} \left(\frac{\dot{V}}{\dot{V}_{base}} \right) \left(\frac{\Delta P}{\Delta P_{base}} \right) \quad (3 - 14)$$

In addition, the total pressure drop through the system is expressed by Equation (3-15) [76].

$$\Delta P = \Delta P_w + \Delta P_d \quad (3 - 15)$$

Where ΔP_d is the pressure drop through the dry channel and ΔP_w is the pressure drop through the wet channel. They are defined by Equation (3-16) and Equation (3-17) [75]:

$$\Delta P_w = 1.12 \rho_a V^2 + 0.125 \rho_a V^2 \quad (3-16)$$

$$\Delta P_d = 1.12 \rho_a V^2 + 0.04 \rho_a V^2 \quad (3-17)$$

Finally, water consumption per kWh of cooling capacity can be expressed by Equation (3-18).

$$WC \left(\frac{m^3}{Kwh} \right) = 3600 \frac{\dot{m}_w (\omega_1 - \omega_3)}{\rho_w Q_{cooling}} \quad (3 - 18)$$

Where ρ_w is the water density.

In the above equations subscripts d and w refer to the dry and wet channels, while wf refers to the water film. The width and length of the channels are designated as W and L , respectively. Thus, the equations 1, 2, 3, 4, and 5 describe the heat and mass transfer of the system under the stated assumptions.

3.1.1 Transport Coefficients

Convective heat transfer coefficients are described by Equation (3-19) corresponding to for the dry and wet channels.

$$h = \frac{Nu k_{air}}{d_h} \quad (3 - 19)$$

Where k_{air} is the thermal conductivity of air, Nu is the Nusselt number, and d_h is the hydraulic diameter of the channel, defined by Equation (3 – 20) [83]:

$$d_h = \frac{4 * \text{cross sectional area}}{\text{perimeter}} \quad (3 - 20)$$

Approximate values of the Nusselt number must be obtained for the dry and wet channels to determine the heat transfer coefficient. The flow through the rectangular flutes in the dry channel is assumed to be laminar, while the heat transfer occurs through only the wall to which the wicking paper is attached. The Nusselt number for fully developed laminar flow between two infinite plates with one wall experiencing heat transfer, according to [78], is $Nu_d = 8.14$. The dry channel convective heat transfer coefficient is then defined by Equation (3-21).

$$h_d = \frac{Nu_d k_{air}}{d_{h,d}} \quad (3 - 21)$$

Where Nu_d is the Nusselt number in the dry channel and d_h is the hydraulic diameter of the dry channel.

Air flows between parallel plates in the wet channel, where heat transfer is through one wall. The Nusselt number for fully developed laminar flow between parallel plates with one wall experiencing heat transfer, according to [78], is $Nu_w = 5.385$. Heat and mass transfer analogy can be used to determine the mass transfer coefficient in the wet channel. Accordingly, the heat and mass transfer relationships of a specific surface shape are interchangeable making $Nu_w = Sh_w$, where Sh_w is the Sherwood number, which represents the dimensionless mass transfer coefficient [84]. Therefore, the heat and mass transfer coefficients in the wet channel can be expressed by Equation (3-22) and (3-23), respectively.

$$h_w = \frac{Nu_w k_{air}}{d_{h,w}} \quad (3-22)$$

$$h_m = \frac{Sh_w \rho_{air} D_{va}}{d_{h,w}} \quad (3-23)$$

Where D_{va} is the binary diffusion coefficient of water vapor in air.

3.1.2 Boundary Conditions

Equation (3-24) and Equation (3-25) provide the air temperature and humidity ratio at the inlet of the dry channel ($x = L$), respectively.

$$T_d = T_{in} \quad \text{at } x = L \quad (3-24)$$

$$\omega_d = \omega_{in} \quad \text{at } x = L \quad (3-25)$$

And as the wet channel inlet is at the outlet of the dry channel ($x = 0$) and a portion of the air in the dry channel is fed into the wet channel, the temperature and humidity ratio are assumed to be equal at this point.

$$T_w = T_d \quad \text{at } x = 0 \quad (3-26)$$

$$\omega_w = \omega_d \quad \text{at } x = 0 \quad (3-27)$$

As the plastic plate dividing the dry and wet channels is expected to be insulated at the inlet and outlet of the cooling system ($x = 0$ and $x = L$), the temperature gradient at the inlet and outlet is assumed to be zero and are expressed by Equation (3-28) and Equation (3-29), respectively.

$$\frac{dT_{wf}}{dx} = 0 \quad \text{at } x = 0 \quad (3-28)$$

$$\frac{dT_{wf}}{dx} = 0 \quad \text{at } x = L \quad (3-29)$$

Equations 26 to 29 are the boundary conditions of the cooling system.

3.1.3 Discretized Model

The first and second derivatives are approximated using backward and centered finite divided difference formulas, respectively, to numerically solve the governing differential equations of the cooling system (equations 1, 2, 3, 4, and 5). The finite difference formulas used are expressed by Equation (3-30) and Equation (3-31) [79].

$$f'(x_i) = \frac{f(x_i) - f(x_{i-1}))}{h} \quad (3-30)$$

$$f''(x_i) = \frac{f(x_{i+1}) - 2f(x_i) + f(x_{i-1}))}{h^2} \quad (3-31)$$

Where h is the step size.

The computational domain of a combination of dry and wet channels is depicted in Figure 3-3.

Nodes (the positions at which the temperature is calculated) are positioned throughout the

computational domain to obtain the numerical model. As shown in Figure 3-3, the position of any node i along x is given by Equation (3-32).

$$x_i = \frac{(i - 1)L}{M - 1} \quad (3 - 32)$$

Where M is the number of nodes used in the x direction. The distance between adjacent nodes is expressed by Equation (3-33).

$$\Delta x = \frac{L}{M - 1} \quad (3 - 33)$$

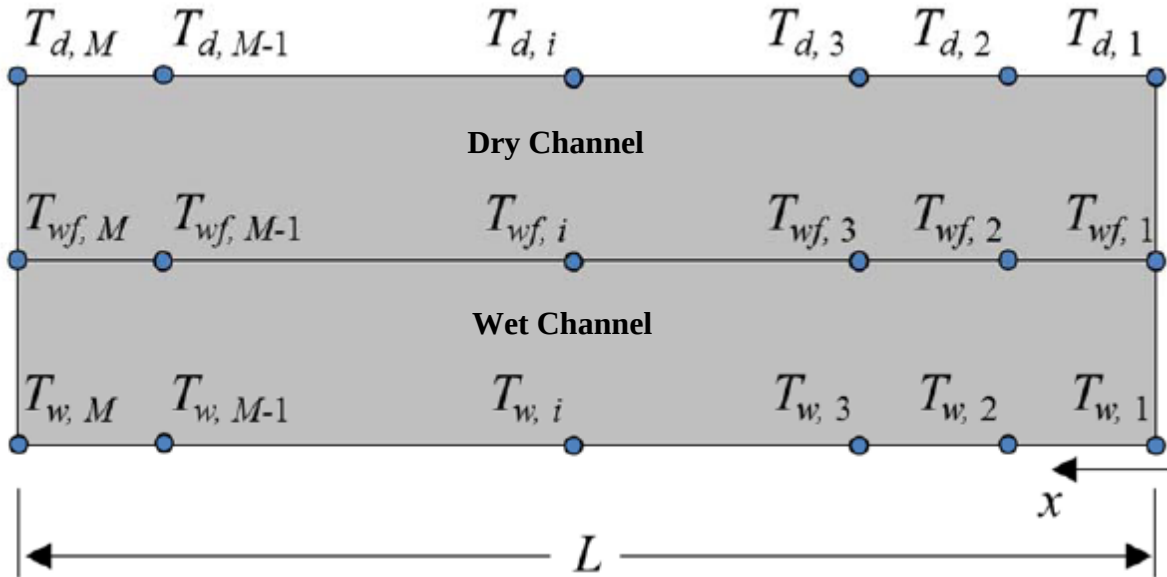


Figure 3-3: The evenly spaced nodes and the computational domain of a pair of dry and wet channels.

Using the above equations, the finite difference version of the governing equations, i.e., the numerical model, is obtained as follows:

Energy balance in the dry channel is defined by Equation (3-34).

$$-m_d c_d \frac{T_{d,i} - T_{d,i-1}}{\Delta x} = U W (T_{wf,i} - T_{d,i}) \quad (3 - 34)$$

Energy balance in the wet channel is defined by Equation (3-35).

$$\dot{m}_w c_w \frac{T_{w,i} - T_{w,i-1}}{\Delta x} = h_w W (T_{wf,i} - T_{w,i}) \quad (3 - 35)$$

Equation (3-36) describe the conservation of mass in the wet channel.

$$\dot{m}_w \frac{\omega_{w,i} - \omega_{w,i-1}}{\Delta x} = h_m W (\omega_{wf,i} - \omega_{w,i}) \quad (3 - 36)$$

Energy balance of the wall is defined by Equation (3-37).

$$\begin{aligned} h_w W (T_{w,i} - T_{wf,i}) + h_d W (T_{d,i} - T_{wf,i}) + H_{w,fg,i} h_m W (\omega_{w,i} - \omega_{wf,i}) \\ + k_{plate} W L \frac{T_{wf,i+1} - 2T_{wf,i} + T_{wf,i-1}}{\Delta x^2} = 0 \end{aligned} \quad (3 - 37)$$

3.2 Numerical Solution Method

The finite difference equations (Equations 34 to 37) derived from the differential equations of the mass and energy balances of the heat exchanger are implemented in the Engineering Equation Solver (EES) framework [85]. As it contains several built-in libraries comprising thermodynamic and thermo-physical properties, EES is a powerful tool for solving engineering problems, particularly thermodynamic and heat transfer problems. All the air, water, and water vapor properties are calculated using internal functions in the EES framework. Systems of equations are

solved using a variation of Newton's method by EES. The process begins in this method with a value guessed for each variable. These values are then iteratively refined until the set of equations are satisfied with a relative residual of 10^{-6} . An important aspect is to choose an appropriate number of nodes to ensure that the numerical solution has sufficient accuracy. The general approach for choosing the number of nodes [80] is to select a key aspect of the solution and examine its behavior as the number of nodes in the computational domain is increased. The selected aspect of the solution, in this case, is the supply temperature of the dry channel. A plot of the supply temperature as a function of the number of nodes is shown in Figure 3-4. The plot indicates that the solution has converged after about 100 nodes.

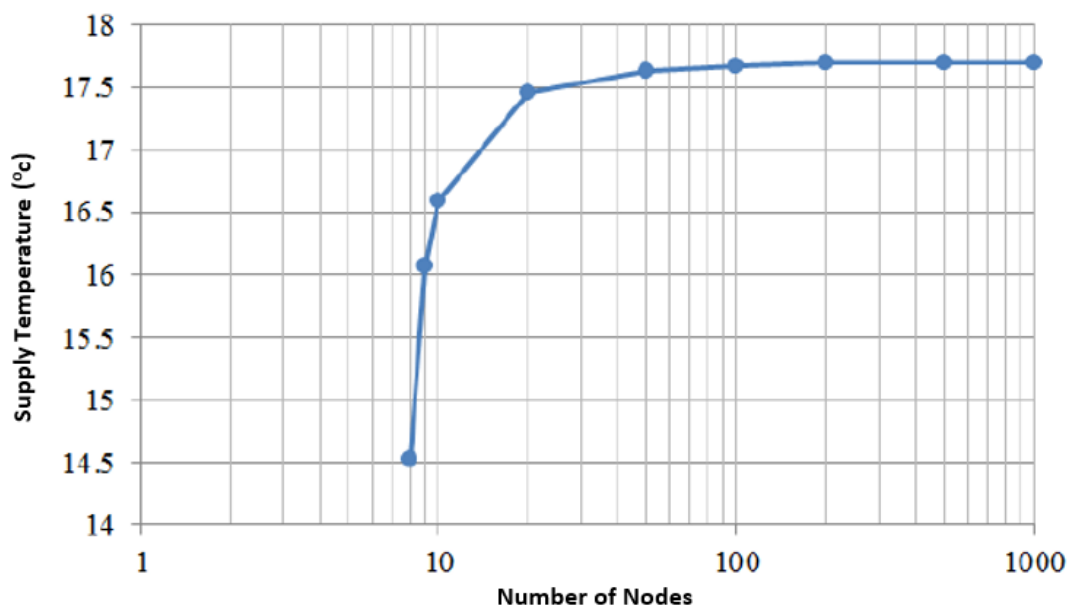


Figure 3-4: The supply temperature as a function of the number of the nodes.

The distribution of the temperature and humidity inside the dry and wet channels of the heat exchanger and the supply temperature and humidity are calculated by solving the systems

equations. The dimensions of the heat exchanger followed by the reference operating conditions are presented in Table 3-1 and 3-2.

Table 3-1: Heat exchanger specifications [51] [52].

Parameter	Symbol	Value (mm)
Dry channel spacing	y_{dry}	3
Wet channel spacing	y_{wet}	3
Channel length	L	1200
Channel width	W	80
Plate thickness	t_{plate}	0.2
Paper thickness	t_{paper}	0.15

Table 3-2: Reference operating conditions [51] [52].

Parameter	Symbol	Value
Inlet air temperature	$T_{inlet}(C)$	25
Inlet humidity ratio	$\omega_{inlet} (kg/kg)$	0.0069
Inlet air mass flow rate	$\dot{m}_d(kg/s)$	0.0006912
Working air to Supply air ratio	$\dot{m}_w / \dot{m}_d$	0.33

3.3 Code Validation

M-cycle based air cooling systems are evaluated using three main methods, namely, numerical simulation, statistical design, and analytical solution methods. However, the most common

evaluation method is numerical simulation. EES software is used to solve the developed mathematical code for the dew point cooling system. The results obtained by simulation using the numerical model were compared with experimental data from literature determined at the same operating conditions and heat exchanger dimensions obtained from the literature to assess the validity of the model. The difference between predicted and measured outlet temperatures is 0.04 to 0.77 °C, which represents a 3 to 5 % relative difference. Several computer programs have been developed to describe the mass and heat transfer phenomena in this novel cycle with an average error of 8%-10%. However, as illustrated in Figure 3-5 the overall error of predictions made using the model proposed in this study is 3.3%. Even though the results predicted by the model match the trend of the experimental results, the predicted supplied temperatures are lower than the experimental results published in the literature. The higher supplied temperatures obtained in experiments compared to the model predictions can be due to several factors, including but not limited to the non-uniform airflow and uneven water distribution in the wet channel.

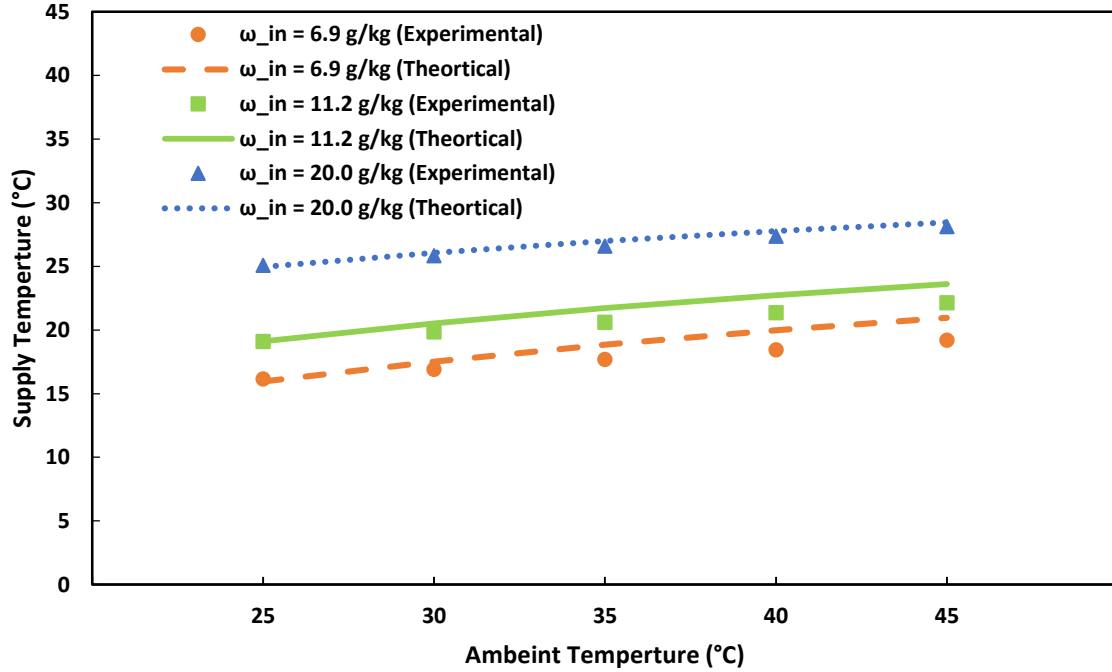


Figure 3-5: Comparison between the experimental results and the predictions of the present study.

3.4 Case Study

3.4.1 Description of the Base Case

Al-Hekmah mosque is modelled in this study. It is a typical large sized mosque with a rectangular single zone open space and a total area of 2840 m², 1:1.3 aspect ratio and 7 m ceiling height located in Riyadh, which is used to perform the daily as well as Friday prayers. The mosque operates intermittently five times a day all year around with assumed maximum occupancy of 100% in Friday prayers. The density of the occupants is 1.78 m²/Person. A design builder model for Al-Hekmah mosque was simulated to predict the patterns of electricity consumption of the mosque based on real data collected from a comprehensive site survey. The results of the model was validated using the actual electricity consumption by determining the accuracy of the model

with a calculated overall relative error of 5%. Table 3-3 describes the characteristics and specifications of the modeled mosque collected from a site visit.

Table 3-3: The specification of AL-Hekmah mosque

Characteristics	Description of the Mosque
Location	Riyadh (24.77 N Latitude , 46.74 E longitude
Orientation	Long axis facing the Holy Ka'bah (Qiblah) in the city of Makkah (oriented 25° due south of west)
Shape	Rectangular
Floor to Ceiling Height	7 m
Floor Area	(71x40) 2840 m ²
WWR	15%
Occupancy	1600 person
Occupant Density	1.78 m ² /person (1.78m x 1m)
Metabolic Rate	0.9 ma
Clothing Insulation	0.8 for summer and 1 for winter
Operating Schedule	Typically , five times daily intermittent occupancy with an average of one hour for each occupancy
Infiltration	0.5 ACH , it is the most difficult parameter to measure due to frequent opening of doors and the weak tightness construction

3.4.2 Details of the Envelope

The envelope of the mosque is composed of the external walls, windows, roof, and the flooring system. The average thickness of the walls is 273 mm and the overall U-value is 0.656. The outermost layer of the building is made of plaster (dense) with varying thickness depending on the location of the wall. The structure of the wall is constructed with two layers of medium-sized concrete blocks with thermal insulation of equal thickness sandwiched in between. The inside layer is made of plaster (lightweight). Windows with sliding panels made of fixed glass plates mounted on an aluminum frames without a thermal break are the most common in mosques. Typically, the windows are double glazed with an air gap and a total thickness of 24 mm and an overall U-value of 2.709 W/m²-K. Both glasses are lightly tinted with different thicknesses. The thickness of the

roof of the mosque is 348 mm and the overall U-value is 0.466 W/m²-K. The roof is constructed of sand/screed, cement mortar, asphalt roofing, thermal insulation, and dense reinforced concrete, with concrete roofing tiles as the outermost layer and lightweight plaster as the innermost layer. The flooring of the mosques consists a slab on grade with an overall U-value of 0.792 W/m²-K. The outermost layer is a fibrous pad carpet laid on top of glazed ceramic tiles. The tiles are laid on a base consisting of layer of sand at the bottom followed by high density polyethylene and dense reinforced concrete with cement mortar at the top. Details of the envelope of the base case mosque is summarized in Table 3-4.

Table 3-4: Collected data for the envelope of Al-Hekmah mosque

Envelope System Type	Layers (Outside to Inside)	Thickness (m)	U-Value (W/m ² .K)
Wall	Plaster , dense	0.013	0.656
	Concrete Block , medium	0.10	
	Extruded Polystyrene	0.05	
	Concrete Block , medium	0.10	
	Plaster , lightweight	0.01	
Roof	Concrete Tiles , roofing	0.025	0.460
	Cement Mortar	0.013	
	Extruded Polystyrene	0.06	
	Asphalt Roofing Roll	0.025	
	Reinforced Concrete	0.2	
	Plaster , lightweight	0.025	
Window	Glass , generic tinted	0.006	2.709
	Air Gap	0.012	
	Glass , generic tinted	0.006	
Floor	Carpet	0.013	0.792
	Ceramic Tiles , glazed	0.013	
	Cement Mortar	0.013	
	Reinforced Concrete , cast-dense	0.125	
	Polyethylene , high density	0.002	
	Earth , gravel	0.5	

3.4.3 Occupancy Profile

The occupancy depends on the prayer times and varies throughout the year, depending on the solar time. Table 3-5 tabulates the average prayer time in Riyadh region according to seasons to simplify simulating the variations of the loads and scheduling time for each season. Occupancy level, which is a very important parameter that affects the thermal events inside the mosque, affects the energy consumption of the mosque. Operating schedules of mosques are unique compared to other types of buildings. Those who pray normally start entering the mosque after Azan (the call for prayer) and they start leaving after the prayers. However, typically the maximum occupancy level is during the actual prayer, duration of which is normally 15 to 20 minutes. The Friday prayers as well as the Taraweeh prayers have special occupancy profiles as shown in Figure 3-6.

Table 3-5: The average prayer time in Riyadh region according to the season [82].

Season	Month	Fajr	Sunrise	Dhur	Asr	Maghrib	Isha
Spring	March	4:45 AM	6:01 AM	12:06 PM	4:21 PM	6:05 PM	7:18 PM
	April	4:13 AM	5:31 AM	11:58 AM	4:26 PM	6:18 PM	7:34 PM
	May	3:45 AM	5:09 AM	11:55 AM	4:29 PM	6:33 PM	7:54 PM
Summer	June	3:36 AM	5:04 AM	11:58 AM	4:36 PM	6:46 PM	8:11 PM
	July	3:47 AM	5:13 AM	12:04 PM	4:40 PM	6:47 PM	8:10 PM
	August	4:06 AM	5:27 AM	12:02 PM	4:33 PM	6:29 PM	7:47 PM
Autumn	September	4:22 AM	5:39 AM	11:53 AM	4:13 PM	6:00 PM	7:13 PM
	October	4:35 AM	5:51 AM	11:44 AM	3:48 PM	5:29 PM	6:42 PM
	November	4:50 AM	6:09 AM	11:43 AM	3:31 PM	5:10 PM	6:25 PM
Winter	December	5:08 AM	6:29 AM	11:53 AM	3:31 PM	5:10 PM	6:29 PM
	January	5:18 AM	6:38 AM	12:07 PM	3:50 PM	5:29 PM	6:46 PM
	February	5:10 AM	6:27 AM	12:11 PM	4:10 PM	5:49 PM	7:03 PM

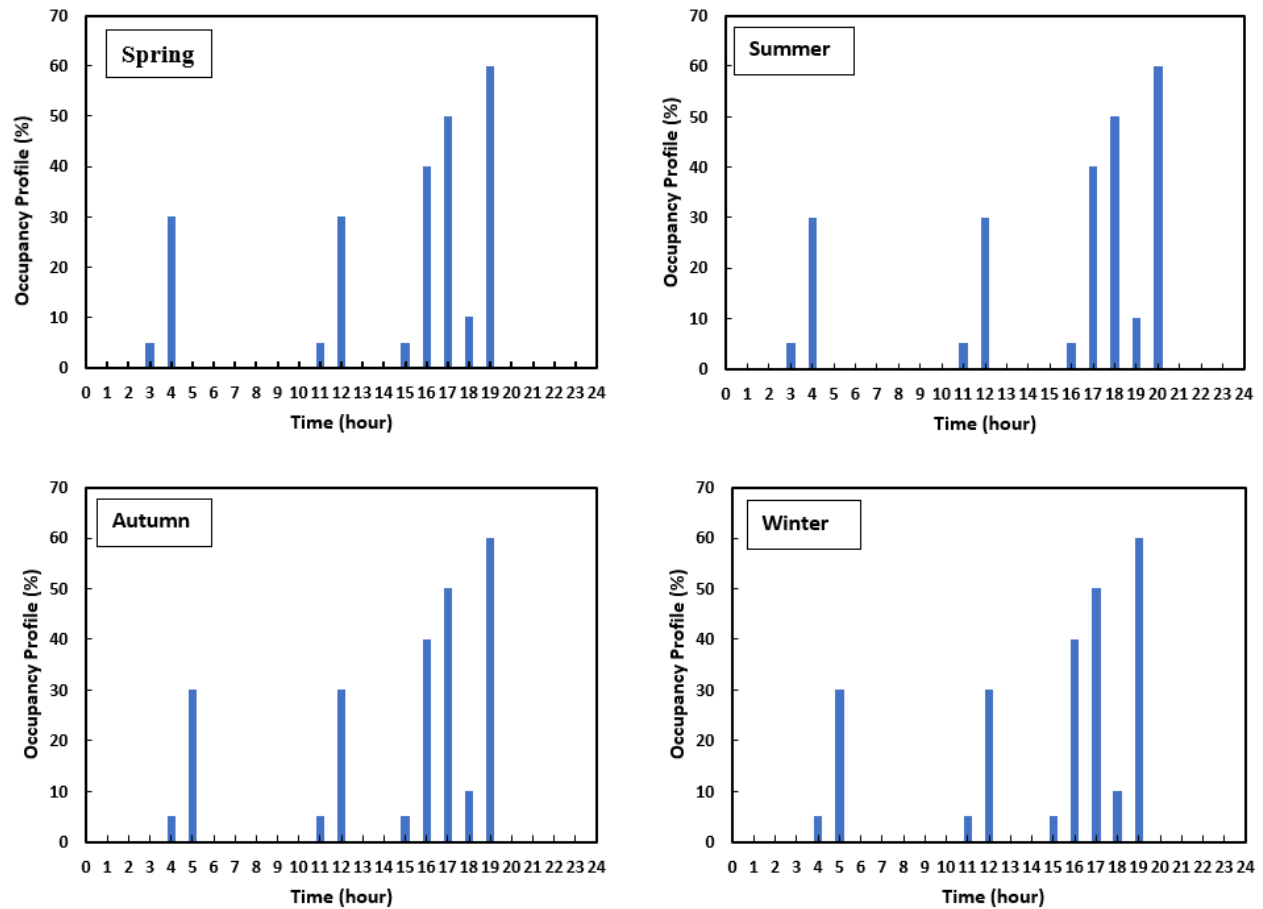


Figure 3-6: Occupancy profiles of mosques.

3.4.4 Cooling System

Unlike other buildings, mosques have a uniform cooling requirement because they have only a single zone. The cooling system in Al-Hekmah mosque consists of 17 central package units with

a total capacity of 139 Kw. According to ASHRAE Standard 62.1-2007 on religious places of worship, the recommended ventilation rates are 2.5 L/s/Person and 0.3 L/s/m² to provide comfortable indoor air quality. Minimum temperature of the air supplied by the system is expected to be 12 °C at a COP of 2.6. For all months requiring cooling the set-point temperature is 24 °C with continuous cooling. The characteristics of the cooling system of a base case mosque is summarized in Table 3-6.

Table 3-6: Characteristics of the cooling system of Al-Hekmah mosque

Characteristics	Description of the Mosque
HVAC System Type	Constant Volume System
Capacity (Kw)	139 Kw (20.5 Ton)
Ventilation	2.5 L/s/Person + 0.3 L/s/m ³
Operation	Typically five times daily intermittent occupancy with an average of one hour before each occupancy period
COP	2.6
Minimum Supply Temperature (°C)	12° C
Set Point Temperature (°C)	24° C

3.4.5 Lighting System

The amount of lighting required for a building, in watts per square meter, is specified by ASHRAE/IES 90.1. The required lighting power density (LPD) for the interior of a mosque is 16.5 W/m². The most common type of lights used in mosques in Riyadh is florescent tube lights. Hence, they were selected to fulfill the lighting requirement of the base case model. The characteristics of the lighting system of Al-Hekmah mosque is summarized in Table 3-7.

Table 3-7: Al-Hekmah mosque lighting system characteristics

Characteristic	Description of the system
Light Power Density	16.5 W/m ²
Type of Lighting	Florescent Lights
Operation	Typically during the night time prayers while in the day lighting can be greatly utilized during daytime prayers

3.4.6 Verification of the Base Model

Figure 3-7 depicts a 3D image of Al-Hekmah mosque. This model of the mosque was created using the data described in the previous sections, which was subsequently simulated based on the energy consumption pattern. The results indicate that the highest energy consumption is in August while the lowest is in February, which confirms an envelope behavioral pattern in accordance with the standards. In winter months, which are December, January, and February, cooling is not required. Hence, the electricity consumption drops sharply as illustrated in Figure 22. Verification of the model must be carried out using an actual mosque building before it can be used for analysis. Thus, the electricity consumption of Al-Hekmah that was obtained from a survey conducted in 2017 utilized to validate the accuracy of the model. The mosque is rectangular with the entrance facing a direction of 25 °C from the east. Dimensions of the mosque are 71 x 40 x 7 m. The building has excellent thermal insulation with a floor area about 2840 m². The maximum capacity of this mosque building is 1600 persons with a lighting power density is 16.5 W/m². A central air conditioning system, which is operated only during prayer periods, is used. Both the output of the simulation and the energy consumption from the survey were divided by the total area for scaling purposes. The results show good agreement between both electricity consumption patterns with an overall error of 6.5% as illustrated in Figure 3-8.

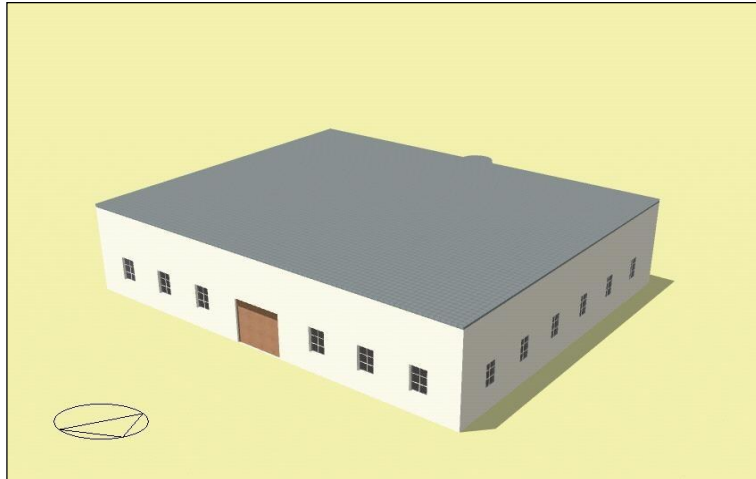


Figure 3-7: 3-D Rendered Image of the modeled mosque.

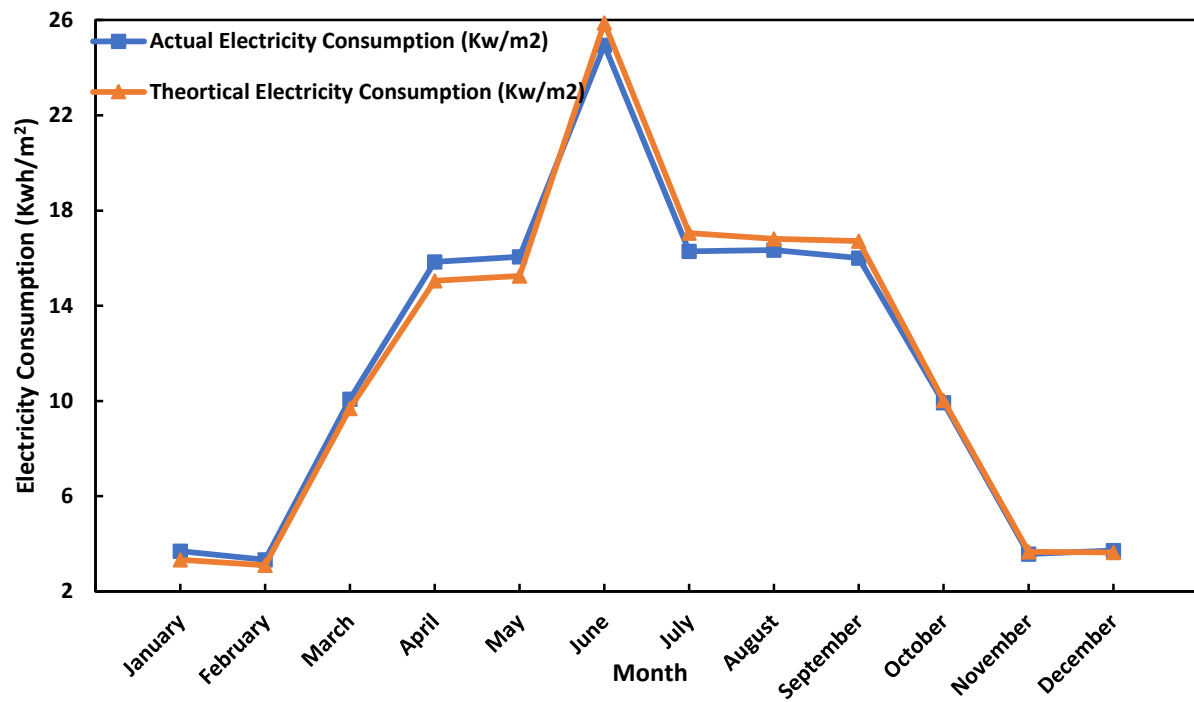


Figure 3-8: Measured and predicted electric energy consumption

CHAPTER 4

RESULTS AND DISCUSSION

The results of mathematical modeling are presented and discussed here in detail. The presentation of the results begins with a discussion of the weather data of Riyadh city, which has a significant effect on the performance of the proposed system, followed by a description of the study of the performance of the system for Riyadh climatic conditions. Finally, the performance of the heat and mass exchanger of the dew point evaporative cooling system under a range of climate conditions, channel geometries, and operational conditions are examined. Figure 4-1 illustrates how the temperature of the inlet air changes as it moves through the heat exchanger. Figure 23 depicts the temperature profiles of the air flowing in the dry and wet channels and the water film along the heat exchanger. As the temperature of the air in the dry channel (T_d) is higher than that of the water film (T_{wf}), heat is loss from the air in the dry channel to the water film. Also, the water film temperature is higher than the air temperature in the wet channel (T_w) for most of the length of the channel except for a small portion near the extraction point.

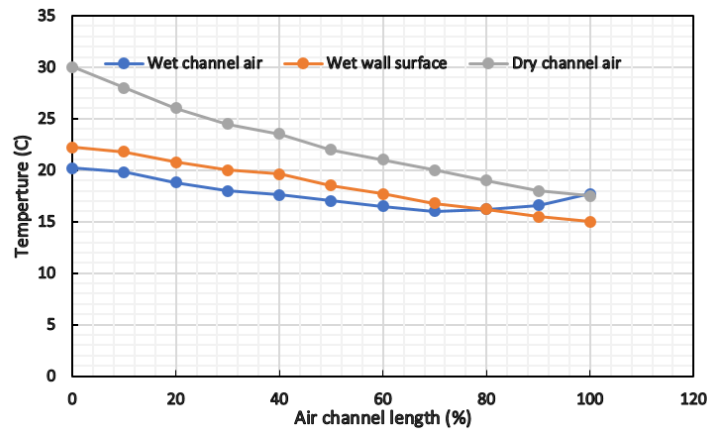


Figure 4-1: Numerical temperature profiles along dry and wet channels.

4.1 Weather Data for Riyadh

In this study, Riyadh was selected to examine the feasibility of using an M-cycle-based cooling system. Riyadh is the capital of the Kingdom of Saudi Arabia, which is characterized by a higher energy consumption due to its high population. The highest electricity demand is in the months of May, June, July, August, and September in Riyadh. Overall performance of the systems based on evaporative cooling is dependent upon the weather conditions, mainly the air temperature and the relative humidity. Hence, Riyadh weather for the whole year is studied thoroughly to gauge the potential of using such a system for cooling purposes. The weather data were obtained using the EnergyPlus Program of the Department of Energy. EnergyPlus Program provides the hourly weather data, including the barometric pressure, dry bulb temperature, and the dew point temperature. The hourly data are used to calculate average monthly data. Figure 4-2 and 4-3 illustrate the average monthly relative humidity and dry bulb temperature for Riyadh. In the summer season the temperature in Riyadh, except sometimes in September, is well above the comfortable level. However, the climate in Riyadh is characterized by low relative humidity. Diurnal variations indicate that the temperature increases from 6:00 a.m., reaching a maximum at 12:00 noon, and subsequently decreasing unlike the variations of relative humidity. Maximum, average, and minimum values of the temperature and relative humidity in Riyadh are listed in Table 16. The data in Table 16 indicate that the minimum average recorded temperature is 9 °C and the maximum is 43 °C. Riangvilaikul and Kumar [51] [52] have evaluated the M-cycle in the relative humidity range of 35–84%. Zhao et al. [60] have concluded that the M-cycle-based cooling system can be used in areas with a relative humidity below 70%. As shown in Table 4-1, the maximum relative humidity of 68% in Riyadh meets this criterion required for the successful

operation of an M-cycle-based cooling system. Furthermore, during the summer season the relative humidity is very low and well below the critical value.

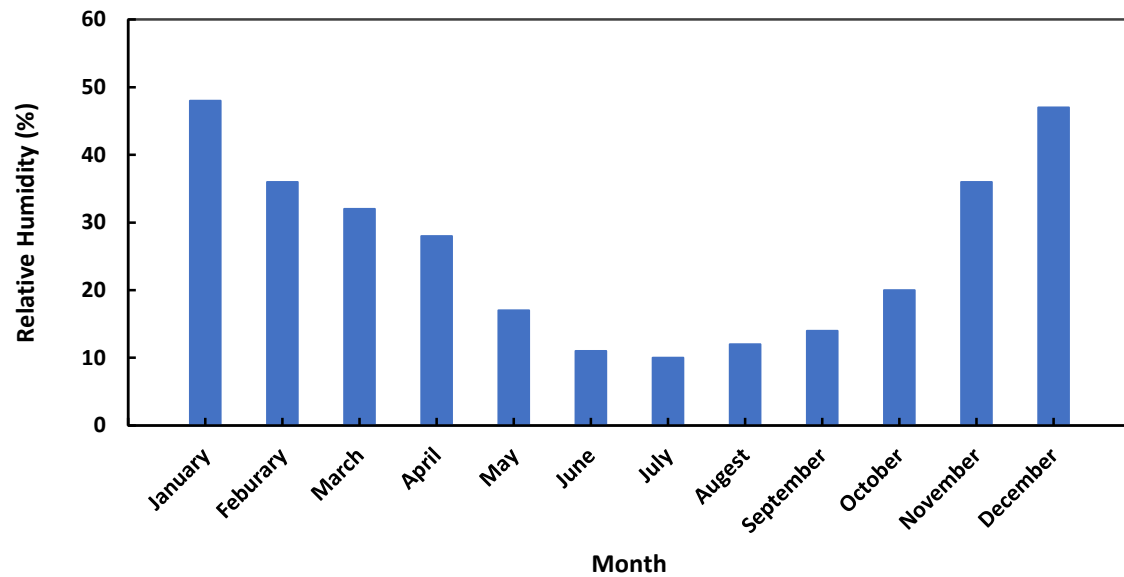


Figure 4-2: Average monthly temperature in Riyadh city.

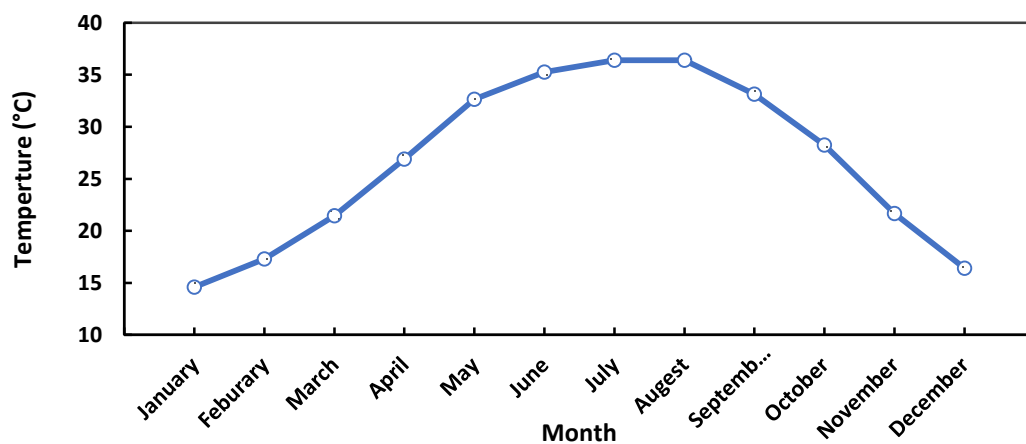


Figure 4-3: Average monthly humidity in Riyadh city.

Table 4-1: Dry bulb temperature and relative humidity for Riyadh city.

Month	Temperature (°C)			Relative Humidity (%)		
	Minimum	Maximum	Average	Minimum	Maximum	Average
January	9	20.2	14.6	28	66	47
February	11.2	23.4	17.3	7	65	36
March	15.2	27.7	21.45	7	57	32
April	20.4	33.4	26.9	7	49	28
May	25.9	39.4	32.65	6	28	17
June	28	42.5	35.25	5	17	11
July	29.3	43.5	36.4	5	15	10
August	29.2	43.6	36.4	6	18	12
September	25.9	40.4	33.15	4	24	14
October	21.2	35.3	28.25	9	31	20
November	15.5	27.8	21.65	14	58	36
December	10.6	22.2	16.4	26	68	47

4.2 Performance Assessment of M-cycle based Cooling System for the Climatic Conditions of Riyadh

The performance of M-cycle-based cooling system for the climatic conditions existing in Riyadh city is assessed in this section. The geometric dimensions of the proposed system are a channel length of 120 cm, channel width of 8.0 cm, and a gap of 3 mm, while the operating conditions are an air velocity of 2.4 m/s and an exhausted-air-to-intake-air ratio of 33%. The results are expressed in terms of the average monthly variation of the temperature of the supplied air, relative humidity of the supplied air, cooling effectiveness, cooling capacity, power, COP, and water consumption.

4.2.1 Characteristics of the Supplied Air

The monthly variation of the relative humidity and temperature of the supplied air are depicted in Figure 4-4 and 4-5, respectively. The results show that the monthly average temperature of the supplied air ranges from 13.8 to 19.89 °C and the average relative humidity ranges from 51% to 72%. In Riyadh the maximum temperature of 43.86 °C occurs in July. The temperature and relative humidity of the supplied air corresponding to July are 19.81 °C and 51, respectively. As shown in Figure 27, for most of the year the supplied air properties in Riyadh fulfill the comfortable conditions except in the period between November and April, during which the humidity of the supplied air does not meet the comfortable condition, i.e., 50–60% relative humidity. However, most of these months represent the winter season, during which the maximum outdoor temperature is 27 °C. Hence, the dependency on the AC system is at a minimum. Figure 4-6 illustrates the variation of dew point effectiveness and wet-bulb effectiveness. Cooling effectiveness and the ambient relative humidity are inversely related. The cooling effectiveness reaches its maximum in the summer months where the relative humidity is at its minimum. The value of the wet bulb effectiveness exceeds 100% because it is estimated by dividing the temperature difference along the dry side by the difference in the wet bulb temperature of the inlet dry bulb and the inlet.

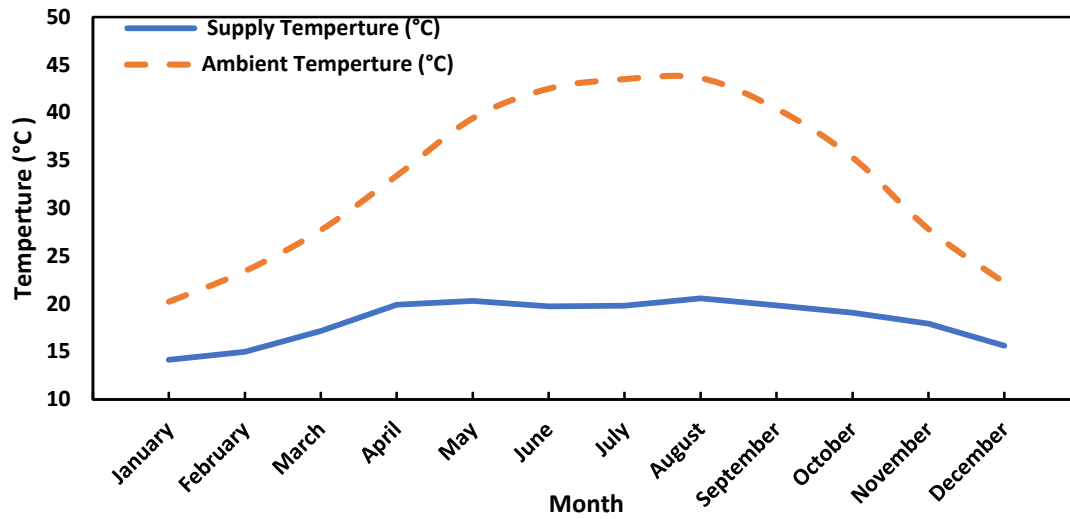


Figure 4-4: Monthly average temperature of the supplied air.

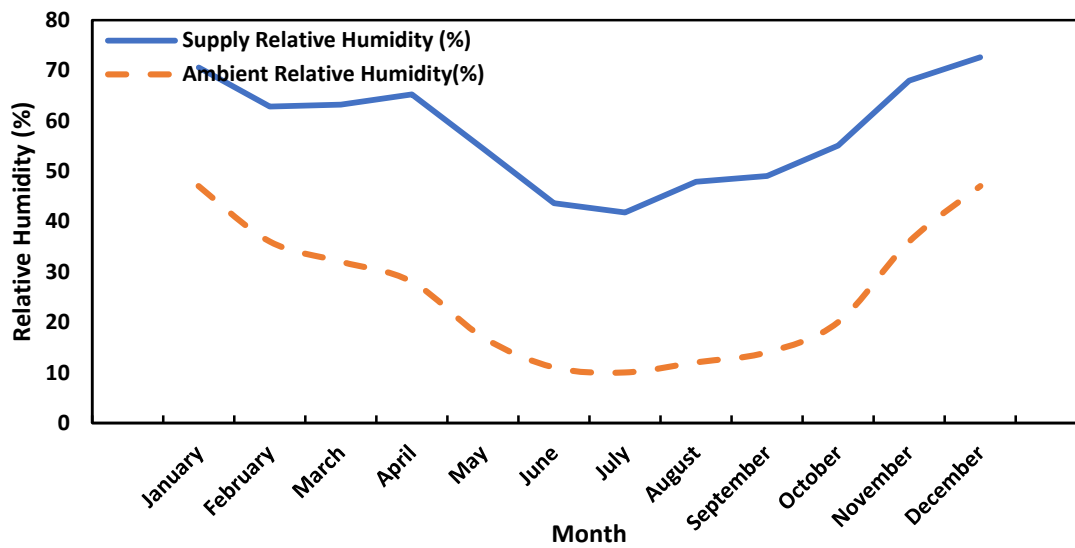


Figure 4-5: Monthly average relative humidity of the supplied air.

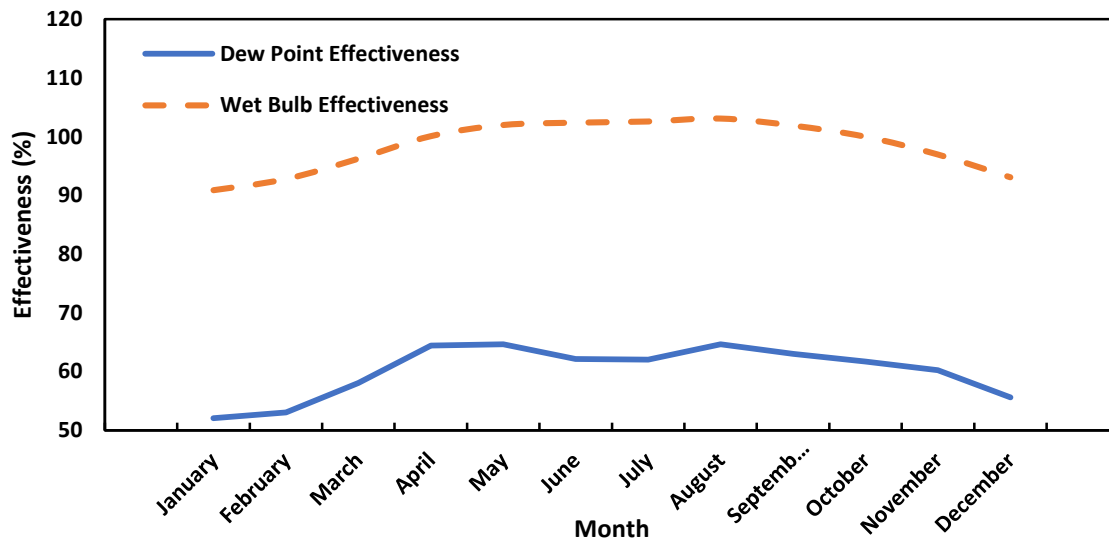


Figure 4-6: Cooling effectiveness for Riyadh.

4.2.2 Coefficient of Performance (COP)

The most important parameters for evaluating the performance of any cooling system are the cooling capacity, consumed electrical energy, and coefficient of performance (COP). Hence, the monthly variation of the cooling capacity of the proposed cooling system is depicted in Figure 4-7. Riyadh has a high cooling capacity due to its low relative humidity. The monthly variation of COP of the proposed cooling system is shown in Figure 4-8. The trend of the monthly variation of COP is comparable to the trend of the cooling capacity depicted in Figure 4-7. Table 4-2 lists the work expended, the average monthly cooling capacity, and the COP of the system, with the cooling capacity ranging between 0.71 and 1.075 kW. The maximum cooling capacity is 1.07520 kW and the corresponding COP is 17, as shown in Table 4-2. These values are for July when the average ambient temperature is 43.6 °C. The work consumption by the system is a function of both the

humidity ratio and ambient temperature. Therefore, the results indicate that electrical energy decreases as the ambient temperature increases. The consumed electrical energy ranges between 0.0041 and 0.06325 kW as shown in Table 4-2. Hence, the results indicate that a cooling system based on the M-cycle yields a high COP compared to the conventional air conditioning systems. Moreover, M-cycle based cooling system requires 0.066325 kW of electrical energy to produce 1.075 kW of cooling capacity. This is equivalent to 88% of the electrical energy consumption of a vapor-compression refrigeration system with a similar cooling capacity. Thus, the performance of M-cycle based cooling system is higher considering that it uses energy only to drive the fan and pump, unlike a conventional system that use electricity intensively to drive the compressor and the fan.

Table 4-2: COP, electrical energy, and cooling capacity of a cooling system based on the M-cycle in Riyadh.

Month	COP	Work Consumption (Kw)	Cooling Capacity (Kw)
January	16.72	0.00450	0.71080
February	16.48	0.00703	0.70730
March	16.24	0.02133	0.72115
April	16.24	0.04266	0.78125
May	16.00	0.05859	0.93750
June	16.75	0.06209	1.04000
July	17.00	0.06325	1.07520
August	16.50	0.06242	1.03000
September	16.00	0.06035	0.96563
October	16.72	0.04875	0.86806
November	16.24	0.01969	0.78125
December	16.72	0.00413	0.73451

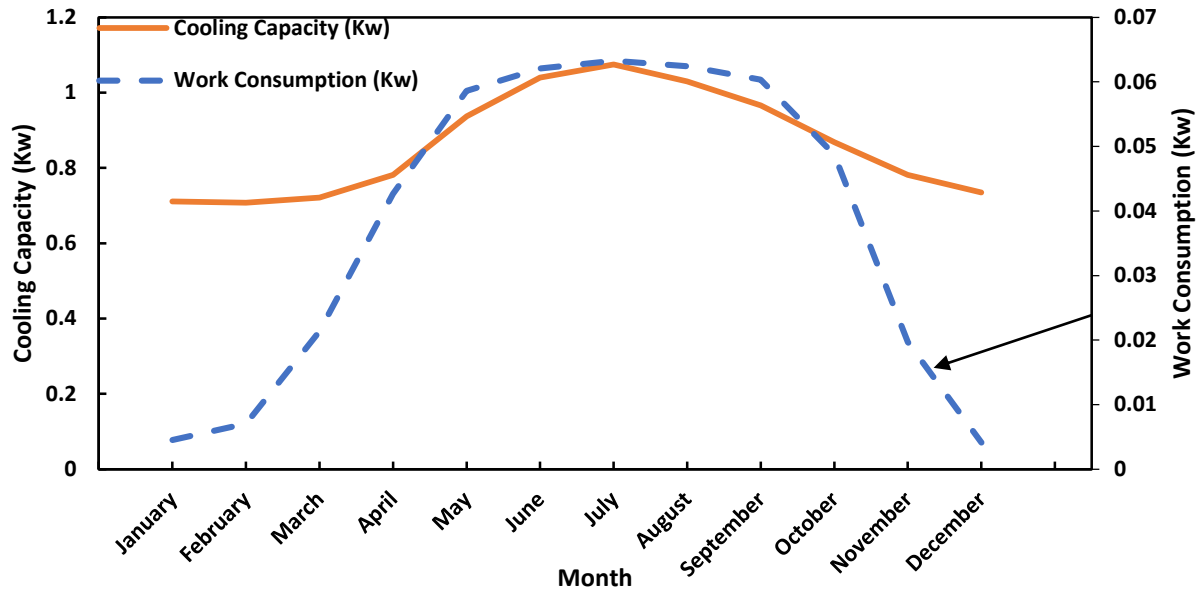


Figure 4-7: Cooling capacity and electrical energy used by a cooling system based on the M-cycle in Riyadh.

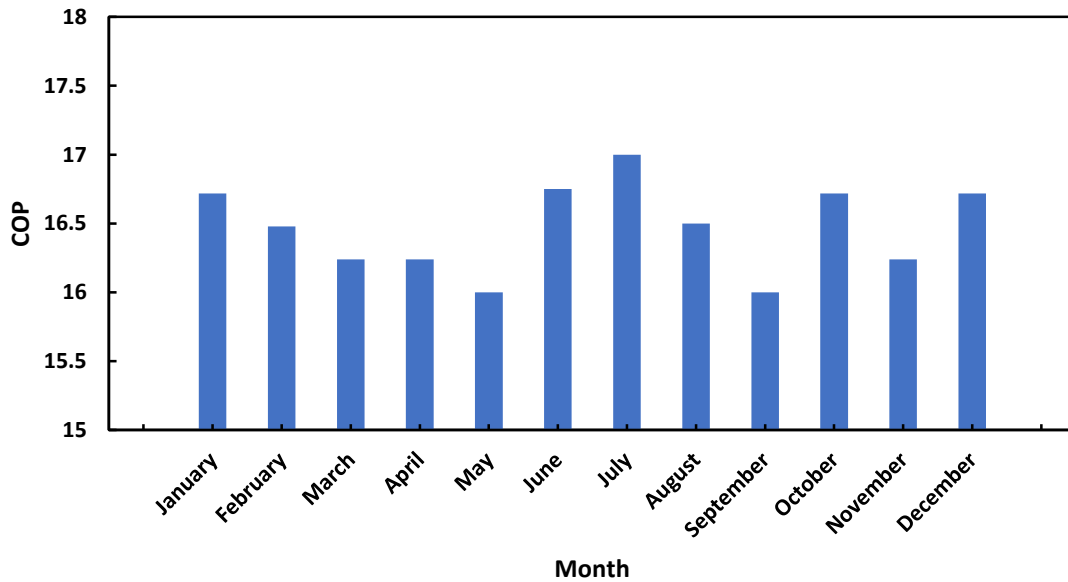


Figure 4-8: Variation of the coefficient of performance for the proposed cooling system in Riyadh.

4.2.3 Water Consumption

Theoretically, the evaporation rate of water is equal to the volume of moisture increase of the working air while passing the wet channel as described by Equation 18. Figure 4-9 and 4-10 displays the water consumption of the cooling system based on the M-cycle. The water consumption by the proposed cooling system in Riyadh city varies between 0.29 and 0.71 mL/s, and the maximum consumption occurs in August. The results demonstrate that a cooling system based on the M-cycle requires about 0.787 mL/s of water to reduce the air temperature in Riyadh from 43.9 to 19.8 °C equivalent to 1.05 kW of cooling (i.e., 2.3 L/kW).

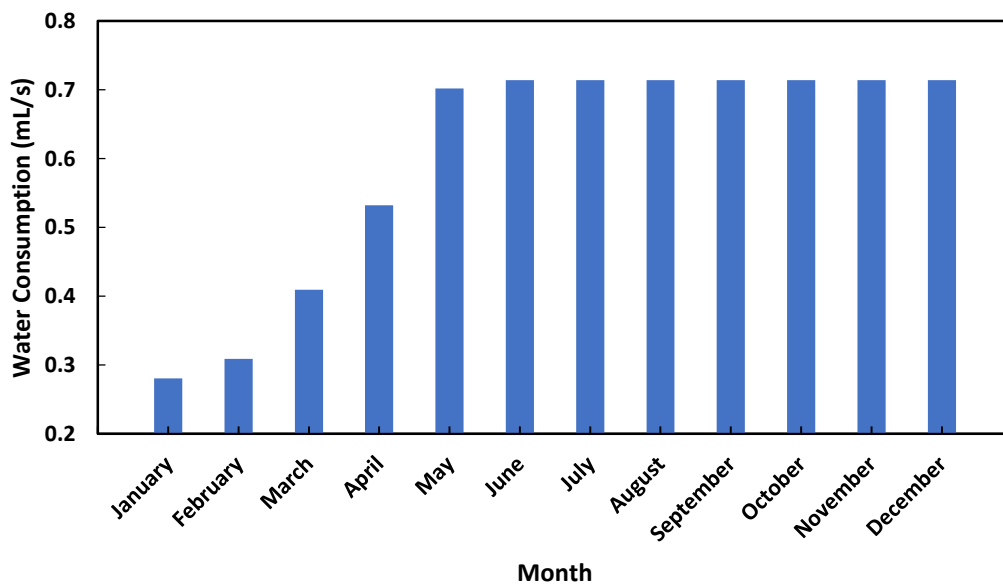


Figure 4-9: Variation of the water consumption by the air conditioning system based on the M-Cycle in Riyadh.

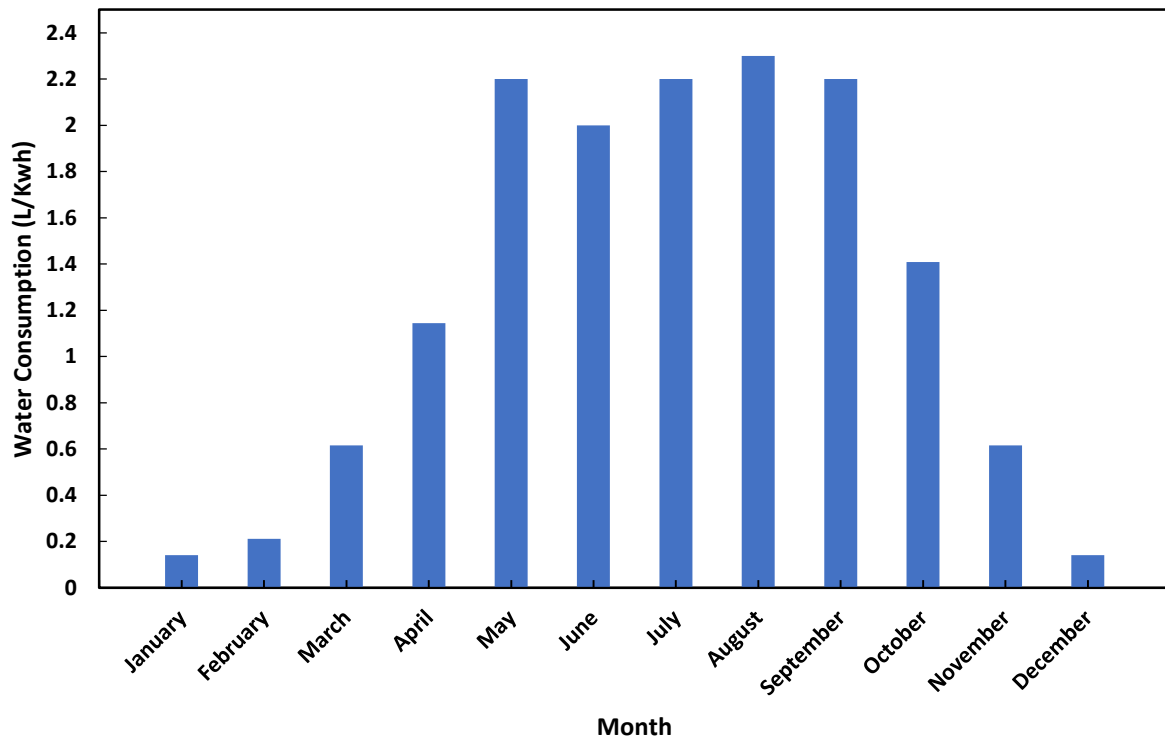


Figure 4-10: Water consumption per kWh by a cooling system based on the M-cycle in Riyadh.

4.3 Effect of Key Parameters on the System Performance

Using the developed model, the effect of different factors, including the channel spacing, length, inlet air velocity, temperature, humidity ratio, and extraction ratio, on the performance of the cooling system was investigated. The cooling performance of an evaporative cooling system is determined using the wet-bulb effectiveness computed based on the inlet and outlet air temperatures. The geometric aspects and flow conditions used for the parametric study are shown in Table 3-1 and 3-2.

4.3.1 Effect of the Inlet Condition of the Air on the Performance of the System

4.3.1.1 Effect of Inlet Air Temperature

Figure 4-11 depicts the effect of the temperature and humidity of the air at the inlet on the supplied air temperature. The results show that the supplied air temperature decreases as the inlet air temperature and humidity levels of entering air decreases. This is due to the enhanced evaporative heat transfer owing to the lower inlet air humidity ratio providing a greater driving force for mass transfer. Moreover, the temperature of the supplied air increases as the humidity of inlet air increases at a given temperature of the ambient air. Furthermore, the temperature of the inlet air increases almost linearly with the temperature of the supplied air. Considering that the slope of the curve of supplied air temperature versus the ambient temperature ranges between 0.1 and 0.3 °C/deg, the supplied air temperature increases by 1– 3 °C when the ambient air temperature increases by 10 °C. Also, at the same temperature of the intake air, the supplied air temperature decreases by almost 3.6 °C per 4.3 g/kg reduction of absolute humidity. Finally, the results indicate that if absolute humidity is lower than 12.9 g/kg, as in the case of Riyadh city, the resulting supplied air temperature will be under 25 °C.

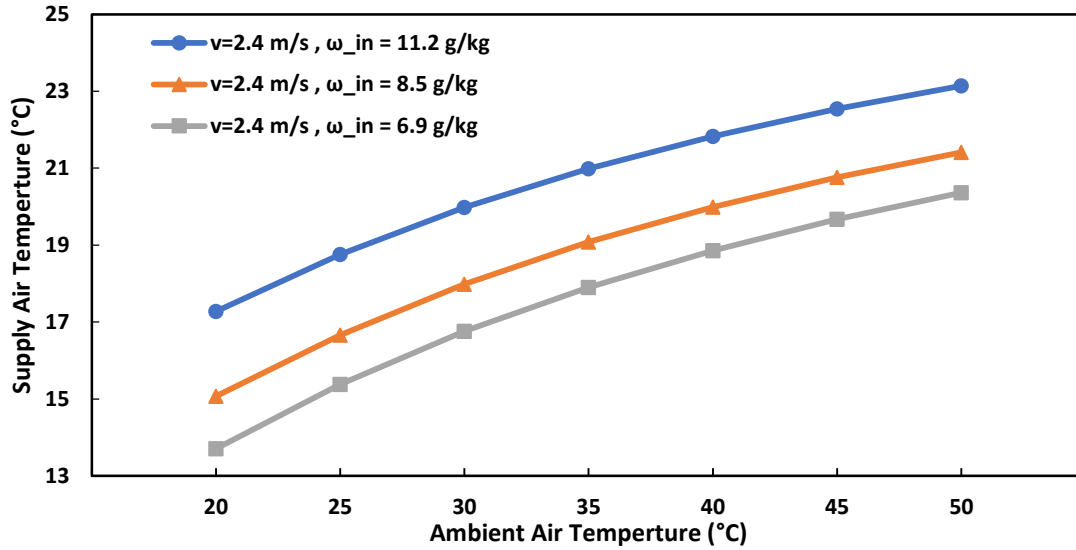


Figure 4-11: Supplied air temperature for different intake air conditions.

Figure 4-12 shows the wet-bulb effectiveness of the proposed system under different inlet air temperature and humidity conditions. When the ambient air temperature increases from 25 to 45 °C the wet-bulb effectiveness ranges between 95 and 110 %. The driving force for mass transfer is increased due to the larger vapor pressure difference between the air and water interface owing to the low humidity ratio of the inlet air. A larger amount of sensible heat is transferred from the dry channel to the wet channel and the moisture absorbing capacity of working air increases due to water evaporation when the driving force for mass transfer increases. Hence, when the humidity ratio of the inlet air is low, the wet-bulb effectiveness increases. Also, the wet-bulb effectiveness proportionately increases with the inlet air temperature. The results in Figure 4-13, on the other hand, indicates that the dew point effectiveness decreases as the humidity ratio of the inlet air decreases due to the temperature difference between the inlet dry bulb and dew point temperature, which increases rapidly at low values of the humidity ratio.

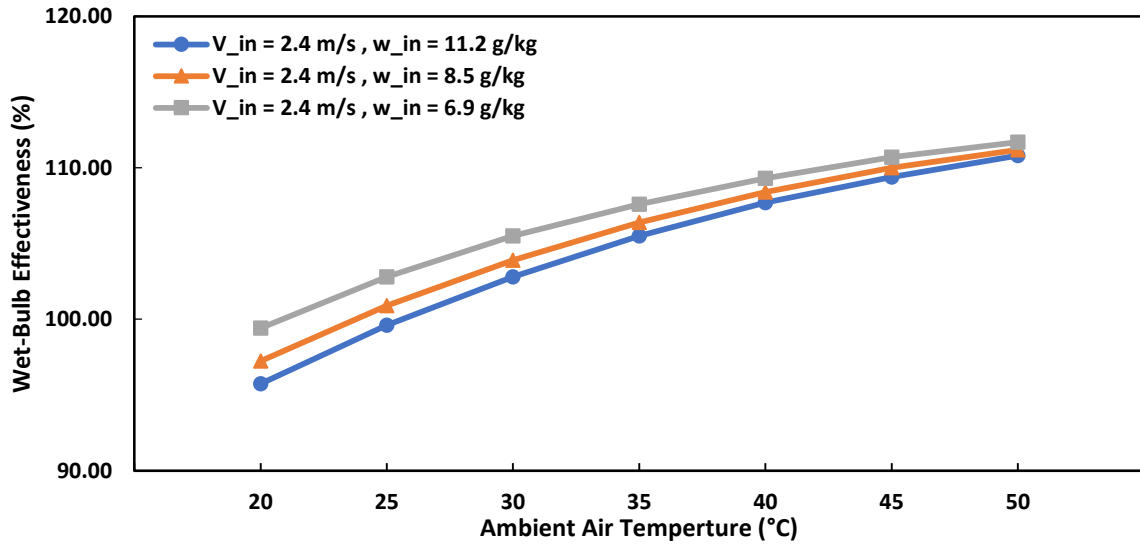


Figure 4-12: Wet-bulb effectiveness for different intake air conditions.

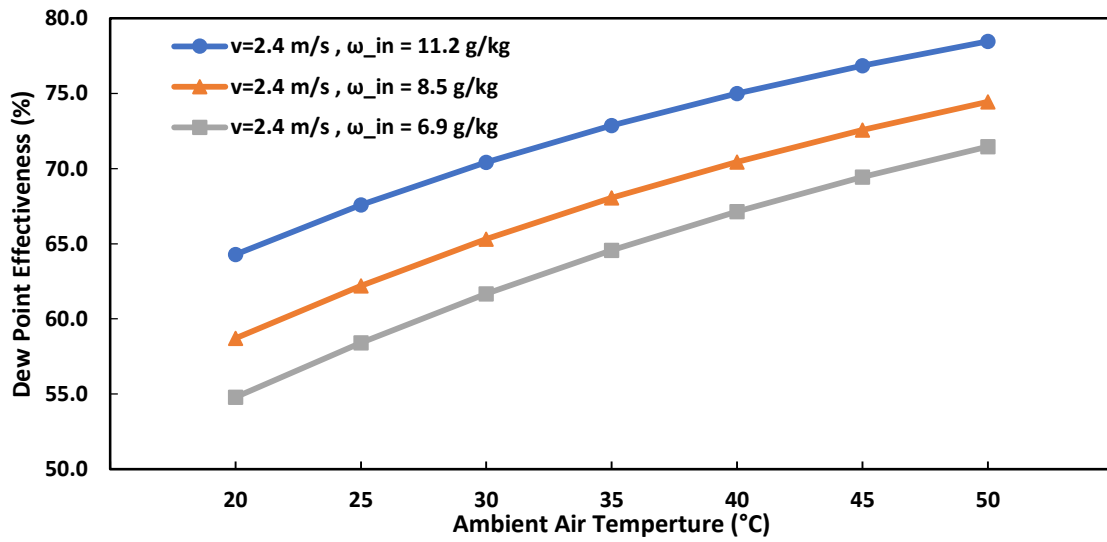


Figure 4-13: Dew point effectiveness for different intake air conditions.

4.3.1.2 Effect of the Intake Air Velocity

Figure 4-14 depicts the relationship between the inlet air velocity and the supplied air temperature. Cooling is higher for the same air temperature and humidity level at the inlet at lower velocities. Moreover, as illustrated in Figure 4-15 and 4-16, increased inlet velocity significantly decreases both wet-bulb effectiveness and dew point effectiveness. A shorter contact time with the wet surface at higher velocities decreases the evaporation of the water film, resulting in the increase of the temperature of the supplied air and the decrease of both dew point effectiveness and wet-bulb effectiveness.

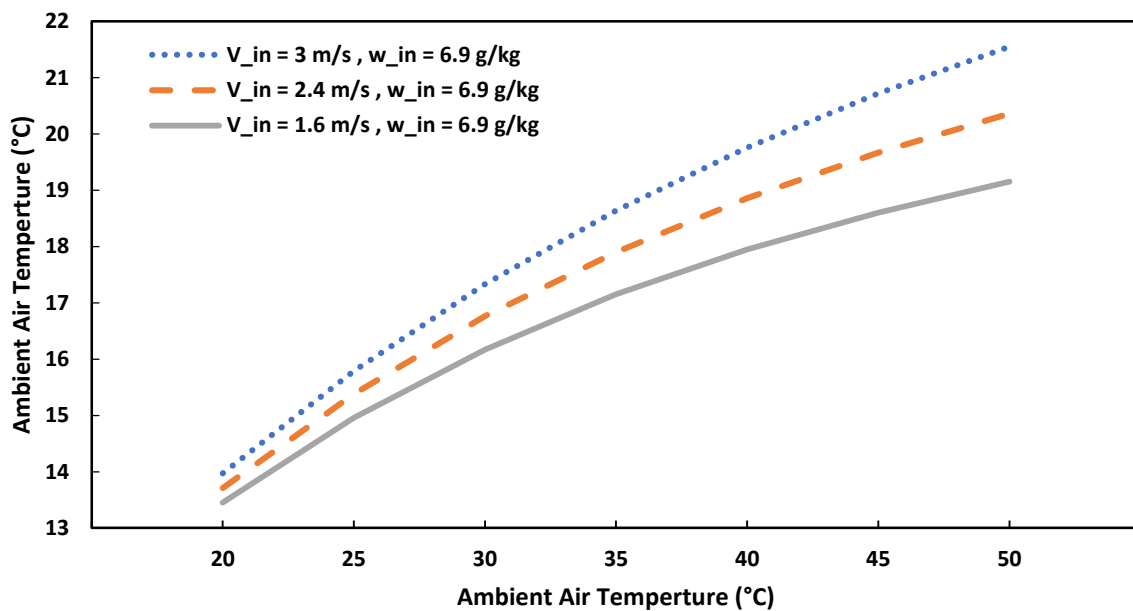


Figure 4-14: Effect of the variation of the intake air velocity on the supplied air temperature.

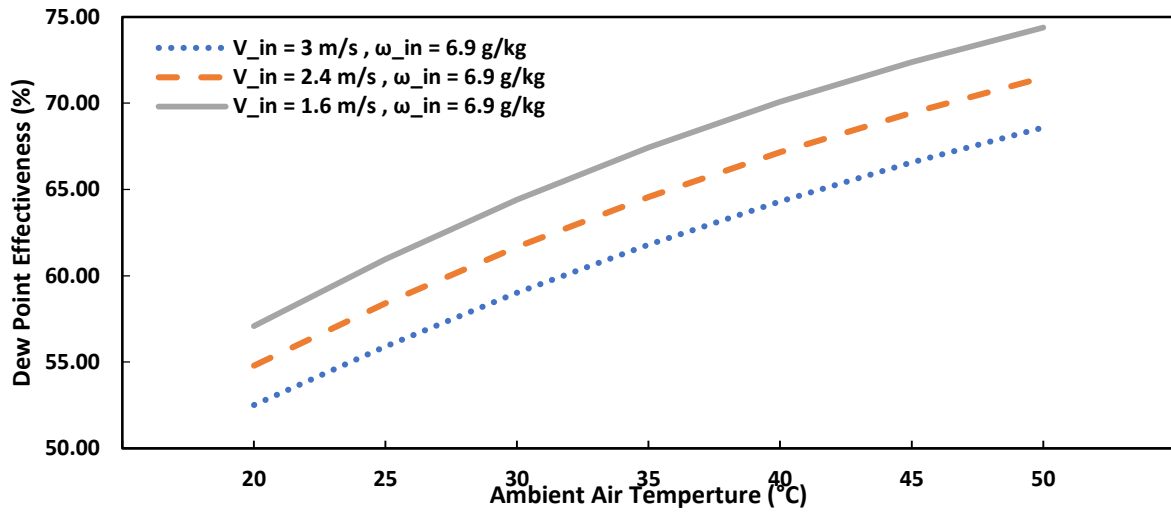


Figure 4-15: Effect of the variation of the intake air velocity on the dew point effectiveness.

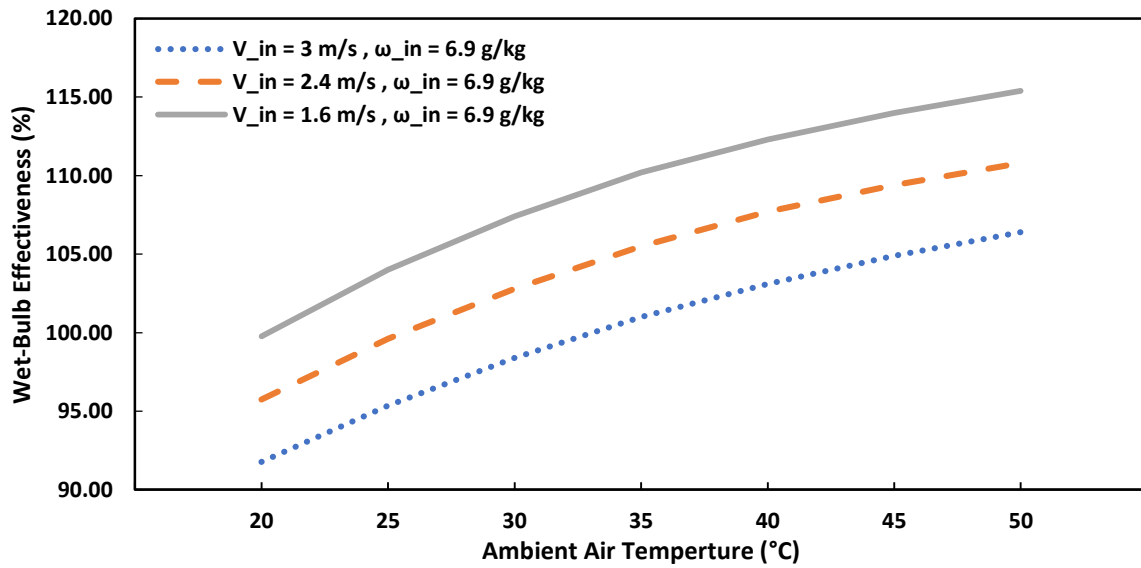


Figure 4-16: Effect of the variation of the intake air velocity on the wet-bulb effectiveness.

4.3.1.3 Effect of Humidity on the Cooling Capacity

The results depicted in Figure 4-17 indicate that, the cooling of the supplied air decreases as the humidity level of the inlet ambient air increases, and hence for the same temperature and velocity the cooling performance of the system decreases. Sensible heat is transferred from the dry side to the wet side of the channel wall due to the higher heat capacity of the working air compared to that of the water film owing to the lower humidity ratio of the inlet air.

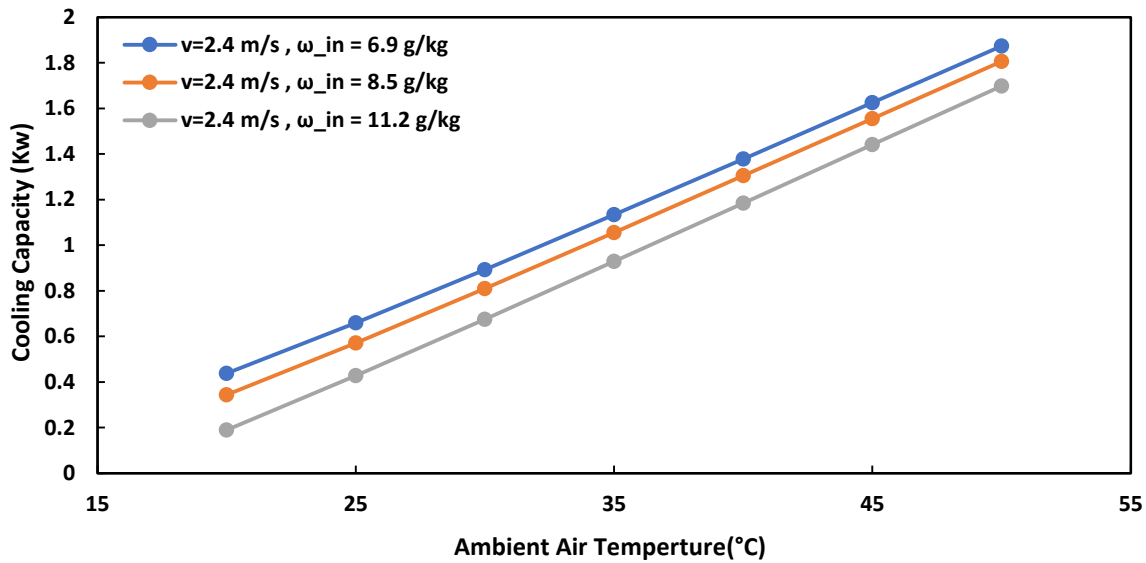


Figure 4-17: Effect of the humidity and temperature of the intake air on the specific cooling capacity.

4.3.2 Performance under Varying Operating Conditions

In this section, the results of a parametric analysis conducted to determine the optimum values of key parameters to design a highly efficient system are described. The parameters evaluated are the channel length, wet wall inlet temperature, and working air ratio (airflow mass ratio in the wet channel to that of the dry channel).

4.3.2.1 Effect of Working Air to Supply Air Ratio

The dependence of the dew point effectiveness and wet-bulb effectiveness on the working air to supplied air ratio is depicted in Figure 4-18. Even though a higher working air to supply air ratio improves the cooling effectiveness, it decreases the supply air flow rate and thus can reduce the cooling capacity of the system. As the heat removed from the dry channel is dependent on the amount of air available in the wet channel, the cooling capacity reduction is due to the increase of the water evaporation in the wet channel owing to the increasing ratio of working air. Moreover, the results show that the working air to supplied air ratio should be between 0.30 and 0.63. Higher values of this ratio adversely affects the temperature of the supplied air due to its effect on the cooling effectiveness of the dew point cooling system and the efficiency.

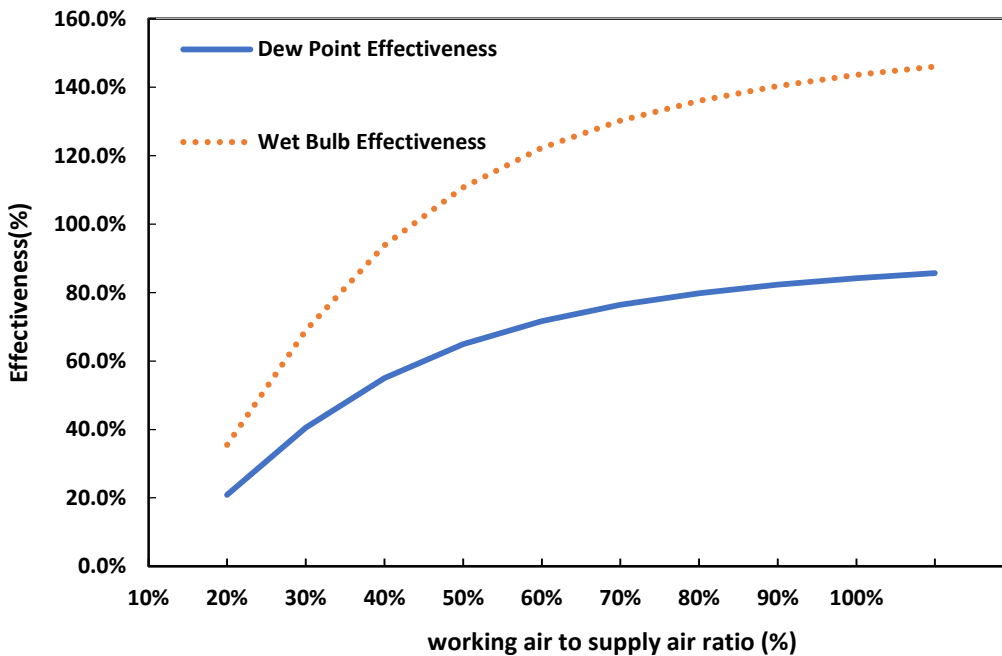


Figure 4-18: Dependence of the dew-point effectiveness and the wet-bulb effectiveness on the working air to supply air ratio.

4.3.2.2 Effect of the feed water temperature

The results of the study of the effect of the feed water temperature on the cooling effectiveness is presented in this section. Figure 4-19 shows the effect of the feed water temperature on the dew-point effectiveness and wet-bulb effectiveness of the system. The results indicate that when the temperature of water is raised from 19 to 23 °C, which is the temperature range of normal tap water, the dew-point effectiveness and wet-bulb effectiveness decrease less than 10%. Thus, the water temperature has a negligible effect on the dew-point effectiveness and wet-bulb effectiveness, and hence on the performance of the M-cycle based cooling system.

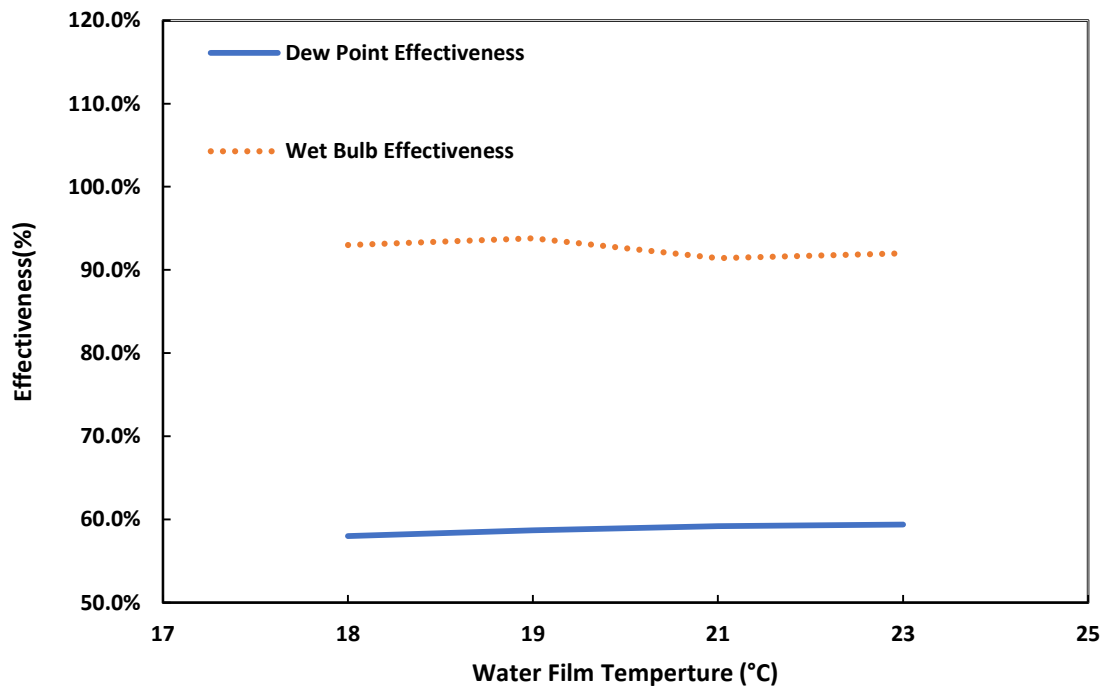


Figure 4-19: The effect of the water film temperature on the cooling effectiveness.

4.3.3 Effect of the Channel Dimensions on the Performance

4.3.3.1 Effect of Channel Spacing

Figure 4-20 shows how the supply temperature, the wet-bulb effectiveness, and dew point effectiveness vary with the channel gap under constant inlet air conditions. The results show that both wet-bulb effectiveness and dew point effectiveness vary inversely with the channel gap. Moreover, the results in Figure 4-20 show that the temperature of the supplied air increases as the channel spacing is increased. The observed effects of the channel spacing is due to the increase of the heat and mass transfer coefficients as the channel spacing is decreased (from equations 21-23). When the channel spacing is smaller than about 4 mm the wet-bulb effectiveness is higher than unity. In order to provide a system with a high cooling effectiveness and a wet-bulb effectiveness greater than 100%, the channel gap should be maintained below 4 mm. However, higher values of dew point effectiveness do not produce the lowest outlet air temperature owing to the dew point temperature of humid air being higher than that of dry air. Thus, the dew point effectiveness imposes a theoretical limit on the extent of the cooling of supplied air.

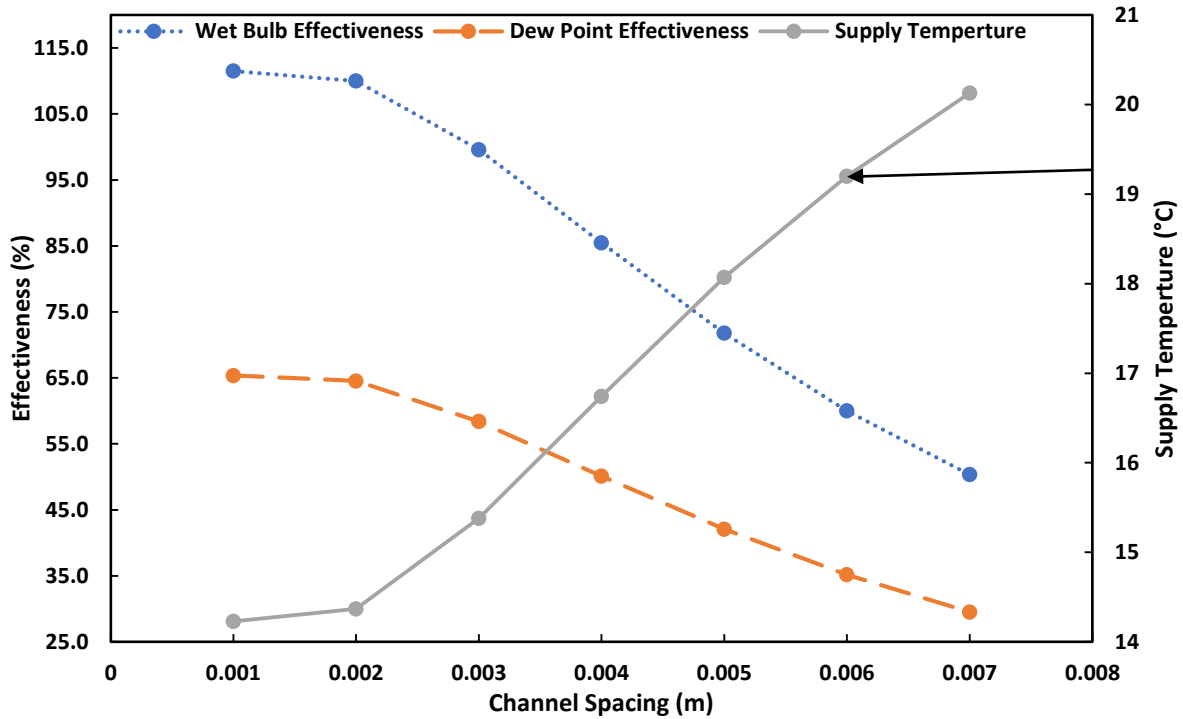


Figure 4-20: Influence of the channel gap on the cooling effectiveness and supplied air temperature.

4.3.3.2 Effect of Channel Length

Figure 4-21 shows the effect of the channel length on the dew point effectiveness, wet-bulb effectiveness, and the supplied air temperature. As the channel length is increased, both dew point effectiveness and wet-bulb effectiveness increase and supply air temperature decreases from 18.72 °C to 14 °C. This is due to the increased time and contact area between the air and water in the wet channel owing to the higher length of the channel enhancing the heat and mass transfer process. However, the increased cost of a larger unit and the operation cost due to higher friction losses should be considered in the optimization of the system. A channel length higher than about 1 m can provide a wet-bulb effectiveness greater than unity. However, the extra power required

by the fan to overcome additional resistance due to the longer channel must also be taken into consideration.

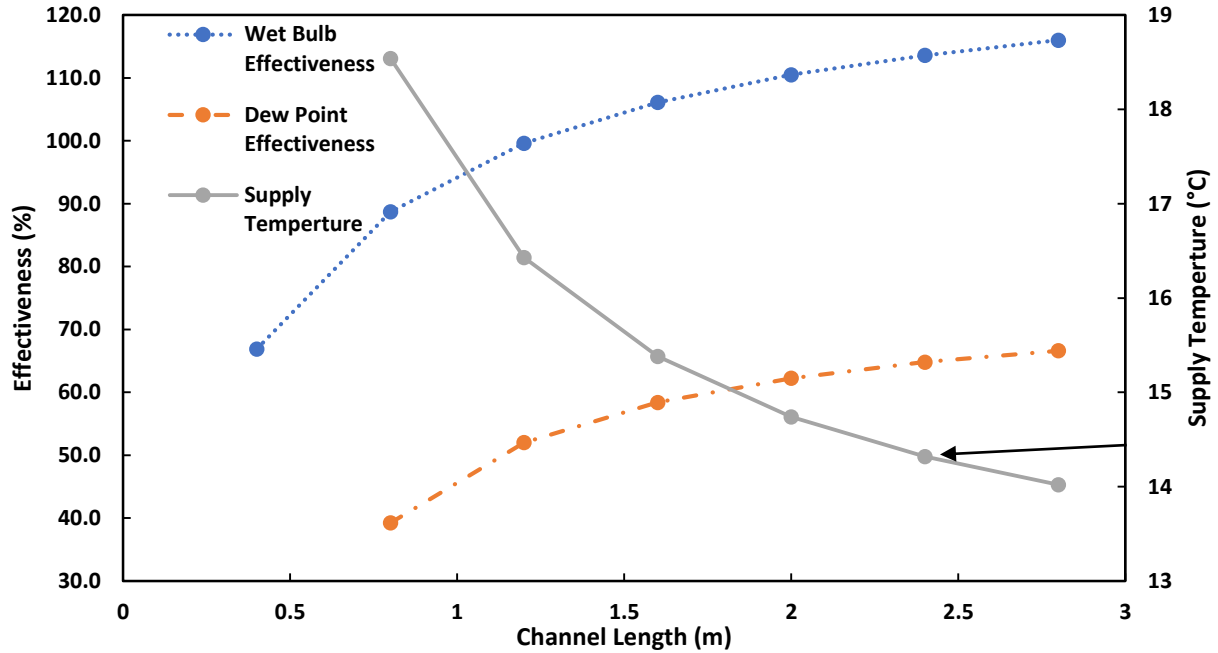


Figure 4-21: Influence of the channel length on the cooling effectiveness and supplied air temperature.

4.4 Cost Analysis of the System

4.4.1 Cooling System

The cost of a commercial cooler based on the M-cycle is compared to a conventional system of the same cooling capacity in this section and the results are provided in Table 4-3. The cost of the two options can be calculated as a function of the operating hours based on an electricity tariff of 0.053 \$/kWh and cost of water of 0.0019 \$/L, using equation (3-38) [87]:

$$Cost = IC + EC \times 0.053 \times \text{hr} + WC \times 0.000245 \times \text{hr} \quad (3 - 38)$$

Ignoring the rates of return, the increased cost of installation of an M-cycle based cooling system can be offset by the increased cost of operation of a conventional cooling system in about 14,500 hours of operation as shown in Figure 4-22. Thus, the payback period of an M-cycle based cooling system, compared to a conventional cooling system, is about 18 months of operation.

Table 4-3: Comparison of an M-cycle based cooling system with a conventional cooling system.

Parameter	Type of Cooling System	
	M-cycle based Cooling System	Conventional Cooling System
Cooling Capacity (CC)	360.5 Kw (102.5 Tons)	
Electricity Consumption (EC)	15.375 Kw	139 Kw
Water Consumption (WC)	1230 L/hr	0
Initial Cost (IC)	169894.80 \$	77666.33 \$

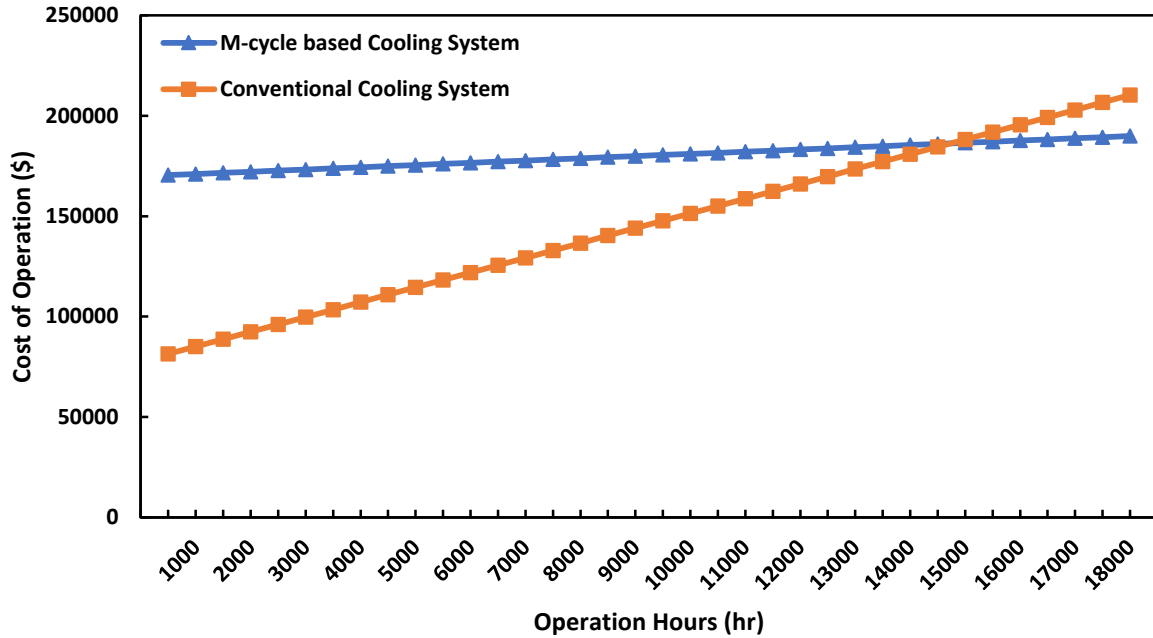


Figure 3-22: Operational cost of an M-cycle based cooling system and a conventional cooling system.

4.4.2 Ablution Grey Water Treatment System

The cost of a wastewater treatment system typically includes the capital cost as well as maintenance and operation costs. The capital costs consist land cost, construction costs, machinery and equipment costs, etc. The operation and maintenance costs comprise of the staff wages, energy costs, cost of chemicals, and sludge disposal costs. The total annualized cost was estimated using Equation (3-39).

$$C_{\text{total}} = C_i + C_F + C_O + C_C \quad (3-39)$$

Where, annualized capital cost (C_i , US\$/year), filter cost (C_F , US\$/year), operating costs (C_O , \$/year) and chemical consumption cost (C_C , \$/year) were estimated using a method given by Nandi et al. as indicated in Table 4-4 [72]. Based on the technical and economic evaluation of five

AGW treatment processes, FA treatment option was selected for integration with the M-cycle based cooling system. The results show that the FA option is the most feasible option to produce treated water for unrestricted irrigation, toilet flushing, cooling systems, street and pedestrian area cleaning, and firefighting provided that a disinfection stage is included. Moreover, the water production cost from the desalination plants is higher than the cost of production utilizing FA based greywater treatment system by 0.11 $\$/\text{m}^3$ as illustrated in Figure 4-11.

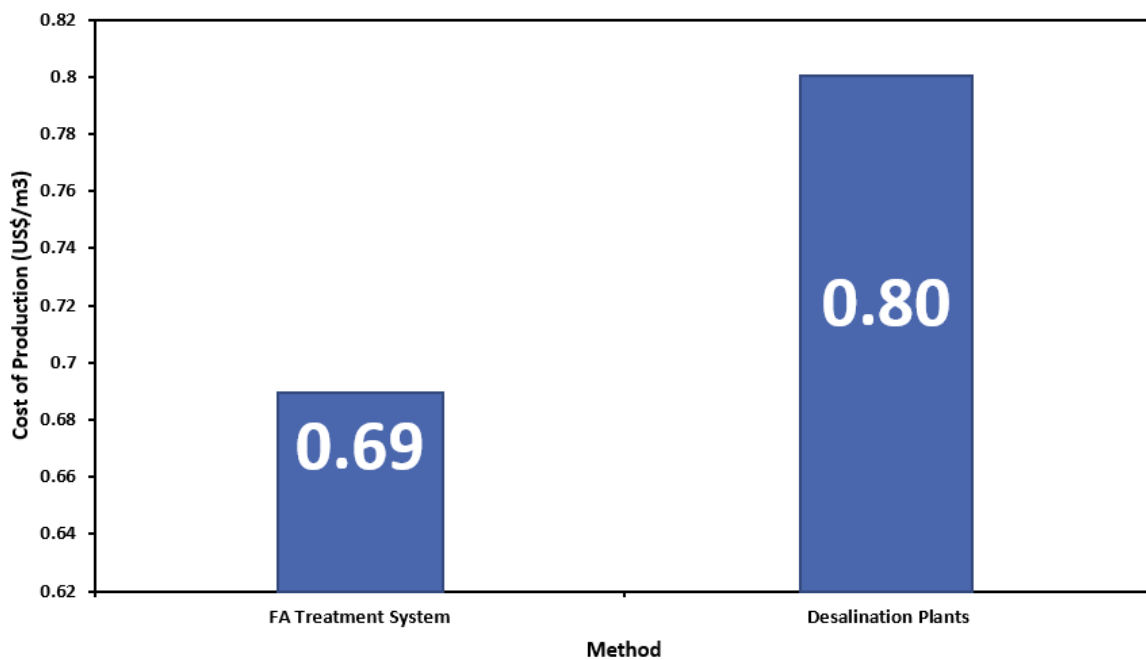


Figure 4-23: Comparison between water production cost utilizing desalination plants vs. FA based greywater treatment plant

4.5 Summary

In this chapter, the performance of the proposed system was investigated. The results show that the cooling system performs well with the wet-bulb effectiveness reaching a value as high as 100%. Furthermore, the inlet air velocity and the extraction ratio have a significant effect on the performance of the cooling system. Also, the heat and mass transfer performance of the proposed cooling system strongly depend on the channel dimensions (channel spacing and length). Moreover, based on the cost analysis, the replacement of conventional cooling systems by an M-cycle-based cooling system integrated with an ablution greywater treatment system would reduce the electricity consumption by 88%, even though the capital cost is almost double.

CHAPTER 5

CONCLUSIONS AND RECOMMENDATIONS

5.1 Conclusions

In this study, M-cycle-based cooling system integrated with an ablution greywater treatment system for mosques located in hot and dry climatic zones was proposed. In addition, a mathematical model has been developed to assess the performance of the cooling system using a variation of Newton's method. The mathematical model addressed the heat and mass transfer aspects and was solved through Engineering Equation Solver (EES). The main results are summarized as follows:

- The base model was validated according to the method described by Riangvilaikul et al. [52] and was found to be accurate with a maximum error percentage of 4%.
- The results show that the monthly average temperature of the supplied air ranges from 13.8 to 19.89 °C and the average relative humidity ranges from 51% to 72%.
- The increase of the channel length enhances the dew point effectiveness and wet-bulb effectiveness while reducing the temperature of the supply air. This is because the time and contact area between the air and water in the wet channel increase as the length of the channel is increased, which will enhance the heat and mass transfer process. However, the extra cost of a larger unit and the operation cost due to higher friction losses should be taken into consideration in the optimization of the system.
- The results show that a channel length higher than about 1.2 m would provide a wet-bulb effectiveness greater than unity. However, the extra power required by the fan to overcome

additional resistance due to the longer channel should be taken into consideration in the optimization stage.

- The results show that both wet-bulb effectiveness and dew point effectiveness vary inversely with the channel gap, whereas the temperature of the supplied air increases as the channel spacing increases. This is due to the increase of the heat and mass transfer coefficients owing to the decrease in channel spacing.
- The results indicate a less than 10% decrease of the wet-bulb effectiveness and the dew point effectiveness of the system with the increase of the temperature of water from 19 to 23 °C, the temperature range of normal tap water. Thus, the water temperature has a negligible effect on the performance of an air conditioning system based on the M-cycle.
- Even though a higher working air to supply air ratio improves the cooling effectiveness, it also reduces the supply air flow rate and thus reduces the cooling capacity of the system. Increasing the working air to supply air ratio increases the water evaporation in the wet channel because the heat removed from the dry channel is dependent on air available in the wet channel.
- Moreover, the results show that working air to supply air ratio must be within the range of 0.30 to 0.63, as the efficiency and cooling effectiveness of the system will be affected by higher values of this ratio and consequently affect the temperature of the supplied air.
- The results also reveal that the cooling of the supplied air decreases with increasing humidity levels of ambient air at the inlet, which reduces the cooling performance of the system at a given set of operating parameters.

- The results show that the inlet air velocity has a significant effect on the effectiveness of the system and the temperature of the supplied air. Both the wet-bulb effectiveness and dew point effectiveness decrease significantly as the inlet velocity increases while the supplied temperature increases.
- Furthermore, the results show that the effectiveness of the system and the supplied air temperature is influenced by the humidity ratio. As the humidity ratio increases, both the supplied air temperature and the dew point effectiveness increase.
- Alternatively, the results show that the wet-bulb effectiveness decreases as the humidity ratio of the inlet air increases due to the temperature difference between the inlet dry-bulb and dew point temperature, which increases rapidly at low humidity ratios.
- Moreover, the results show that the supplied air temperature decreases as the inlet air temperature and humidity levels of the entering air decrease. This is due to the greater driving force for mass transfer provided by the lower inlet air humidity ratio enhancing the evaporative heat transfer.
- Furthermore, the results show that as the ambient air temperature increases by 10 °C, the temperature of the supplied air increases in the range of 1– 3 °C. Also, every 4.3 g/kg decrease of absolute humidity decreases the supplied air temperature by about 3.6 °C at a given intake air temperature.
- The results also show that if the absolute humidity is less than 12.9 g/kg, which is the case in hot and dry climatic zones such as Riyadh city, the temperature of the supplied air will be less than 25 °C.
- Furthermore, the results show that, the water consumption by the system varies from 0.29 to 0.71 mL/s, where the maximum consumption rate is expected to occur during August.

- The results show that a wet-bulb effectiveness of about 120% can be achieved with an inlet air velocity of 1.6 m/s and working air to supply air ratio of 33%.
- The results also show that, the COP of the proposed cooling system has an average value of 16.2. However, its performance is highly dependent on the climate and may not be suitable for all weather conditions.
- Furthermore, the financial results show that the capital cost of the conventional cooling system is 54.2% less than the capital cost of an M-cycle-based cooling system. On the other hand, an M-cycle-based cooling system consumes 88% less electricity than that consumed by a conventional cooling system.
- The result shows that, the proposed ablution greywater treatment system can provide 20%-50% of the water required by the cooling system from a free source of lightly polluted water.
- The economic results show that, the FA treatment option is the most feasible option to be used in the proposed integrated system with a total annualized estimated cost of 0.00085 \$/L.

5.2 Recommendations

- Conduct an experimental investigation this study to increase the confidence in the results.
An experimental study can reveal all factors that affect the overall system performance by empirically verifying different aspects of the present study.
- Study the feasibility of running the system using solar energy as the primary source of heat.
This addition will provide a platform for an M-cycle based cooling system integrated with an ablution greywater treatment system powered by renewable energy.
- Investigate the feasibility of using the proposed system under different scenarios, such as using it in combination with the existing conventional cooling system and studying the overall economic benefits and performance enhancement under each scenario. The main intention of this recommendation is to evaluate many potential scenarios select the optimum system.

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