UNIVERSITY OF TURKISH AERONAUTICAL ASSOCIATION INSTITUE
OF SCIENCE AND TECHNOLOGY

REDUCTION OF ENERGY CONSUMPTION IN HVAC SYSTEMS OF PRISON HALLS IN IRAQ

MASTER THESIS

FADHIL ASAAD. MOHAMMED KADHIM AL-MALAKI

Institute of Science and Technology

Mechanical and Aeronautical Engineering Department

July 2017

UNIVERSITY OF TURKISH AERONAUTICAL ASSOCIATION

INSTITUE OF SCIENCE AND TECHNOLOGY

REDUCTION OF ENERGY CONSUMPTION IN HVAC SYSTEMS OF PRISON HALLS IN IRAQ

MASTER THESIS

Fadhil Asaad Mohammed Kadhim AL-Malaki

1403730034

IN PARTIAL FULFILLMENT OF THE REQUIREMENT FOR THE DEGREE OF MASTER OF SCIENCE IN MECHANICAL AND AERONAUTICAL ENGINEERING

Thesis Supervisor: Asst. Prof. Dr. Youde Han

Türk Hava Kurumu Üniversitesi Fen Bilimleri Enstitüsü'nün **1403730034** numarlı Yüksek Lisans öğrencisi **"Fadhil Asaad Mohammed Kadhim AL-Malaki" ilgili** yönetmeliklerin belirlediği gerekli tüm şartları yerine getirdikten sonar hazırladığı **"REDUCTION OF ENERGY CONSUMPTION IN HVAC SYSTEMS OF PRISONS HALLS IN IRAQ"** başlıklı tezini a şağıda imzaları bulunan jüri önünde başarı ile sunmuştur.

Tez Danışmanı : Yrd. Doç. Dr. Youde Han.
Çankaya Üniversitesi ....

Jüri Üyeleri : Yrd. Doç .Dr. Youde Han.
Çankaya Üniversitesi ....

: Yrd. Doç. Dr. Mohamed Salem Elmnefi.
Türk hava kurumu üniversitesi ....

: Yrd. Doç. Dr. Sudantha Balage.
Türk hava kurumu üniversitesi ....

Tez Savunma Tarihi: 10.07.2017.

STATEMENT OF NON-PLAGIARISM PAGE

I hereby declare that all information in this document has been obtained and presented in accordance with academic rules and ethical conduct. I also declare that, as required by these rules and conduct, I have fully cited and referenced all material and results that are not original to this work.

Fadhil Asaad AL -Malaki

10 / 07 / 2017

ACKNOWLEDGEMENTS

Thanks to the most compassionate, gracious and merciful. May Allah's blessings and peace be upon our prophet Mohammed, who protects us from the depths of darkness and leads us forth into light, and his household family.

It is a pleasure to express my special thanks to my wife L-Alaragi who helped me, Ammar abed Allah, his eminence Yacoubi who inspired me, my sons and other friends for their valuable support.

I would like to send my excellence and efforts to all Iraqi Martyrs, Orphans, my parents spirits, dear daughter spirit Rafal Zaid.

I would like to express my sincere gratitude to Asst. Prof. Dr. Youde Han for his supervision, special guidance, suggestions, and encouragement through the development of this thesis.

Fadhil Asaad Mohammed Kadhim AL Malaki

July- 2017

TABLE OF CONTENTS

Chapter Two: Reviw of Literaturs

Chapter Three:Methodolgy Scope

Chapter Four: EXPERIMENTS SIMULATIONS & RESULTS

Chapter Five: Conclusions and Suggestion for Future Works

LIST OF TABLES

LIST OF FIGURES

LIST OF ABBREVIATION

Abbreviation	***Meaning***
A	Area of the transition surface in m^2
ACH	Air Changes per hour
AHU	Air handling Unit
ADoor.in1	Internal Door area which, non-conditioned space in m^2
ADoor.in2	Internal Door area with heat generation source in m^2
ADoor.out	External Doors area in m^2
AGlass.in2	Internal Glass area with heat generation source in m^2
AGlass.out	External Glasses area in m^2
ARoof.out	External ceiling area through non-conditioned space in m^2
ARoof.in1	Internal ceiling area which, non-conditioned space in m^2
ARoof.in2	Internal ceiling area, generation heat source in m^2
AWall.in	External Wall area in m^2
AWall.out	Area of the wall non-conditioned space in m^2
AWall.in1	Internal wall area, non-conditioned space in m^2
AWall.in2	Internal wall area with heat generation source in m^2
BF	By Pass Factor.
BRS	Building Related Sickness
BTU	British Thermal Unit
CAD	Computer Aided Design
CAL	Computer Aided Learning
CD	Correction Factor.
CFD	Computational Fluid Dynamics
CLF	Cooling Load Factor
CLFAH	Cooling Load Factor for Covering Equipment.
CLFG	Cooling Load Factor for Glass
CLFL	Cooling Load Factor for fluorescent lamps
CLFP	Cooling Load Factor for Peoples

Abbreviation	*Meaning*
CLTD	Correction Load Temp difference
CLTD.CR	Correction Load for Heat gain through the Ceilings
CLTD.G	Cooling Load Temp Difference for Glasses
CLTD.GC	Correction difference Temp for Glass
CLTD. R	Cooling Loads Temp Difference for Ceiling or Roof in Kw
CLTD.CW	Correction Load for Heat gain through the walls.
CLTDW	Cooling load Temp difference for Wall in Kw
C.O.P	Coefficient of Performance.
DDC	The number of cooling days over basis temp
DDH	The number of Heating days under basis temp
DR	Daily Temp change Rate
E	Thermal Energy KJ
EC	Energy Consumption in Kwh
E.F.L	Number of the total equivalent load hours
ERLH	Effected Room Latent Heat
ERTH	Effected Room Total Heat
ERSH	Effected Room Sensible Heat
ESHF	Effected Sensible Heat Factor
F	Ventilation Factor
GSHF	Grand Sensible Heat Factor
H	Heating load KJ/h
hi, ho	Thermal Convection Coefficient
H/W	Aspect Ratio
K, K1, K2	Thermal Conductivity coefficient, for each materials
K-ε	Turbulence Model (Kinetic energy, dissipation rate)
KI	Internal air temp correction factor
KR	Roof Correction Factor
KW	Wall Correction Factor
L	Max Cooling Load

Abbreviation	***Meaning***
LHG	Latent Heat Gain from one service equipment, without covers.
LMR	Correction Factor For Monthly, latitude and longitude for Roof
MCW	Mass of water in condenser
MEW	Mass of water in evaporator
NOF	Number of fluorescent lamps
NOP	Number of peoples
NOT	Number of tungsten lamps
Q	Heat Transfer.
QALH	Latent Heat Gain from Ventilation in Kw
QAPL	Total Latent Heat Gain from service equipment, without covers in Kw
QAPS	Total Sensible Heat Gain from service equipment, without covers in Kw
QAPSH	Sensible Heat Gain from cover equipment in Kw
QASH	Sensible Heat Gain from Ventilation in Kw
QDoor.cin1	Heat gain from internal doors, non- conditioned with generation source in Kw
QDoor.cin2	Heat gain from internal doors with heat generation source in Kw
QDoor.cout	Heat Gain from external doors in Kw
QFloor.c	Heat Gain from the floor or earth in Kw
QGlass	Heat Gain through glasses in Kw
QGlass.cin2	Heat Gain for glasses with heat generation source in Kw
QGlass.cout	Heat Gain from external glasses by condition in Kw
QGlass.solar	Heat Gain from the solar through glasses in Kw
QInf.L	Latent Heat Load from air leakage in Kw
QInf.S	Sensible Heat Loads from air leakage in Kw
QLight.F	Heat gain from fluorescent lamps and led in Kw
QLight.T	Heat gain from Tungsten lamps in Kw
QPeoples.S	Total Sensible Heat Gain from peoples in Kw
QPeoples.L	Latent Heat gain from peoples in Kw
QRoof.cin1	Heat gain through the non- conditioned space for the ceilings in Kw
QRoof.cin2	Heat gain from ceiling with heat generation source in Kw

Abbreviation	***Meaning***
QRoof.cout	Heat Gain from external ceiling in Kw
QT	Total Heat Transfer in Kw
QWall.cout	Heat Gain from external Walls in Kw
QWall.cin1	Heat Gain from internal walls, non- conditioned space in Kw
QWall.cin2	Heat Gain from the non- conditioned space for the walls with generation heat source in Kw
R	Utilization factor
RLH	Room Latent heat in Kw
RLHL	Room Latent heat Load from HVAC system in Kw.
RSH	Room Sensible heat in Kw
RSHF	Room Sensible heat Factor.
RSHS	Room Sensible heat Load from HVAC system in Kw
RTH	Room Total heat in Kw
SAF	Safety Factor for Lamps
SBS	Sick Building Syndrome
SC	Shading Coefficient
SHFG	Max Heat Gain for Glasses in Kw
SH.G.H	Sensible Heat Gain for covering equipment in Kw
SHGP	Sensible Heat Gain from peoples in Kw
Ta	Outside Temperature rate at every hour °C
Tadp	Dew-Point °C
Tb	Variable basic Temp°C
Tc	Condenser Temperature °C
TDC	Slot Length for doors in m
Te	Evaporator Temperature °C
TID	Temperature in indoor(Internal) °C
TLH	Total Latent Heat in Kw
TOD	Temperature in Outdoor(External) °C
TOG	Glasses Correction Factor for External dry bulb temp.

Abbreviation	***Meaning***
TOR	Ceiling Correction Factor for External dry bulb temp.
TOW	Walls Correction Factor for External dry bulb temp.
TSH	Total Sensible Heat in Kw
TWC	Slot Length for glasses in m^2
U	Total Heat Transfer Coefficient in W/m^2K
UA	Overall Heat Transfer Coefficient in W/m^2K
UDoor.cout	Total Heat Transfer Coefficient for external doors in summer W/m^2K
UDoor.in1	Total Heat Transfer Coefficient for internal doors W/m^2K
UGlass.cout	Total Heat Transfer Coefficient for glasses in summer W/m^2K
UGlass.in1	Total Heat Transfer Coefficient for internal glasses W/m^2K
URoof.cout	Total Heat Transfer Coefficient for external ceiling in summer W/m^2K
URoof.in1	Total Heat Transfer Coefficient for internal ceiling in summer W/m^2K
UWall.cout	Total Heat Transfer Coefficient for external walls in summer W/m^2K
UWall.in1	Total Heat Transfer Coefficient for internal walls W/m^2K
VAV	Variable Air Volume
W	Elbow Width in m^2
Watt.F	Fluorescent lamp power Watt
Wout	Specific humidity for external condition in summer gr water/gr of dry air
Win	Specific humidity for internal condition in summer
ηm	Motor Mechanical efficiency.
ηf	Fan efficiency
ΔPTS	Pressure Difference for Fan psi
Δt	Temperature Difference in between the exterior and interior of day bulb.
ΔX	Thickness mm
2D	Two Dimensions
3D	Three Dimensions

ABSTRACT

REDUCTION OF ENERGY CONSUMPTION IN HVAC SYSTEMS OF PRISON HALLS IN IRAQ

AL-Malaki, Fadhil A. Mohammed

M.Sc., Department of Mechanical Engineering
Supervisor: Asst. Prof. Dr. Youde Han
July 2017, 99 pages

Corrections facilities (Prisons) are like small towns. They have all of the energy demands of housing (dormitories, lavatories, laundry, kitchen, etc.) as well as specialty functions (administration, security, medical and dental clinics, vocational training, storage, etc.) it also needs very good ventilation to provide safe environment and prevent transferring diseases, keeping in mind that it works for 24 hours 7 days a week. So to save energy in this diverse environment, every operational aspect must be closely examined for potential savings. Located in a high desert environment gives more challenges to engineers and designers. At the present days, energy is playing a very important role and this role is increasing in Iraq since the energy cost has been increased rapidly at the last few years.

Knowing that air conditioning system for such facilities were designed in a poor manner and energy consumption was nearly neglected in Iraq, this raise the need of reconsidering the air conditioning systems of these facilities under the present conditions and find the best suitable system that may satisfy the international recommendations and at the same time reduce the energy cost.

In-order to find the best system/accessories that gives reliability and efficiency, we had considered the following:

1. Different types of facilities such as normal prison and TB (Tuberculosis) personal Prison, Large facilities, small facilities etc…

2. These facilities have been solved for different systems like traditional split AC, Central cooling, and direct/indirect evaporative cooling in order to find which system performs more efficient.

3. Applying some efficient considerations for the conditions and control system that may be used with the system like economizers, Variable Speed Drives for the AC motors, Variable Air Volume for the supplied air, Ventilation control.

For the two Stage cooler system, Direct Indirect Evaporative coolers IDEC with using the variable air volume energy, are contributed to decrease the required energy consumption dramatically at a yearly total amount as much as 40% compared with any other systems. The cost on monthly energy consumption can be reduced at about 25 US$ in low temperature seasons and about 43US$ in high temperature seasons for every prisoner. Air provision will be sometimes as large as three times compared with other systems. This feature gives a big advantage to achieve the correct ventilation in prison halls and low maintenance.

Keywords: HVAC Systems- HVAC Requirements– System approach – Optimization – Degree, Day Method - Heat gain loads - HAP program.

ÖZET

Irak'ta Hapishane Salonların Havalandırma Sistemindeki Enerji Tüketiminin Azaltılması

AL-Malaki, Fadhil A. Mohammed

Yüksek Lisans, Makine Mühendisliği Anabilim Dalı

Tez Tanışmanı: Yrd. Doç. Dr. Youde Han

Temmuz 2017, 99 Sayfa

Düzeltme tesisleri (Hapishaneler) küçük kasaba gibidir. Konutun (Yurtlar, lavabolar, çamaşırhaneler, mutfaklar, vs.) ve özel fonksiyonların (İdare, güvenlik, tıbbi ve diş klinikleri, mesleki eğitim, depolama vb.) tüm enerji taleplerine sahiptirler. Aynı zamanda güvenli bir ortam sağlamak ve haftanın 7 günü 24 saat çalışabildiğini göz önünde bulundurarak hastalıkların aktarılmasını önlemek için de çok iyi havalandırmaya ihtiyaç duyar. Bu nedenle, bu farklı çevrede enerji tasarrufu yapmak için, ve potansiyel tasarruflar için her operasyonel açıdan yakından incelenmelidir. Yüksek bir çöl ortamında bulunan nedeniyle mühendisleri ve tasarımcıları da daha fazla zorlukları veriliyor. Şu günlerde, enerji çok önemli bir rol oynamaktadır ve özelikle Irak'ta son yıllarda enerji maliyeti hızla artığından nedeniyle bu rol daha önemli görünüyor.

Gerçeğini biliyor ki, bunun gibi tesisleri için klima sistemleri sıklıkla kötü bir şekilde tasarlanmış ve Irak'ta enerji tüketimi neredeyse ihmal edildi. Bu sebebiyle, mevcut koşullar altında bu tesislerin klima sistemlerini yeniden gözden geçirme ihtiyacını doğurmaktadır. Uluslararası standartların tatmin edebilecek ve enerji maliyetini de düşürür edebilecek en uygun sistemleri bulmak da gerekmektedir.

Güvenilirlik ve verimlilik sağlayan en iyi sistemleri / aksesuarları bulmak için, bu tez çalışmasında aşağıdaki konuları inceliyoruz**:**

1. Farklı tesis türleri, normal cezaevleri, TB kişisel cezaevleri, büyük tesisler, küçük tesisler vs. dikkate almaktadır.
2. Bu tesisler farklı sistemler için çözülmelidir. Geleneksel split AC, Merkezi soğutma, Doğrudan evaporatif soğutma ve doğrudan / dolaylı evaporatif soğutma gibi sistemleri, bunlarda hangi sistemin daha verimli çalıştığını bulmaktadır.

3. Farklı koşullar için bazı etkin hususlar uygulanır. Ekonomizörler, AC motorlar için Değişken Hız Sürücüleri, verilen hava için Değişken Hava Hacmi ve Havalandırma kontrolü gibi sistemlerde bazı kontrol sistemleri de kullanılmaktadır.

İki kademeli soğutucu sistemi için, değişken hava hacmi enerjisini kullanan Doğrudan Dolaylı Evaporatif soğutucular (DDES), diğer enerji sistemlerle karşılaştırıldığında, gerekli enerji tüketimini yılda toplamda yaklaşık 40% oranında dramatik bir şekilde azaltmaya katkıda bulunmaktadır. Aylık enerji tüketimindeki maliyeti, her mahpus için düşük sıcaklıklardaki mevsimlerde yaklaşık 25 ABD doları ve yüksek sıcaklık mevsimlerde yaklaşık 43 ABD doları olarak düşürülebilir. Hava sağlanması bazen diğer sistemlere göre üç kat daha fazla olacaktır. Bu özellik, hapishane salonlarında doğru havalandırma sağlamak için büyük bir avantaj sağlar,dusuk bakim.

Anahtar Kelimeler: HVAC Sistemleri- HVAC Gereksinimleri- Sistem yaklaşımı- Optimizasyon- Derece, Gün Metodu- Isı Kazanı Yükleri- HAP Programı.

CHAPTER ONE

Introduction

1.1 Meteorological & Energy Depletion

The Industrial renaissance and modern technology that followed the second world war was followed by technology revolutionary in many fields horizontally and vertically, especially in modern education, which has led to designing of powerful and effective air conditioning systems that access all fields of human lives [1]. These developments have also suit with the progresses in other fields such as industrial, agricultural and households. Air conditioners have become an essential part in manufacturing and architecture areas whose functions cannot be neglected, and they are extremely useful where the climate is often very dry and hot. For example, many countries in Arab regions often have such weathers. In order to make people feel more comfortable, air conditioners are something that must be considered and used in almost all government institutions and private homes in Arab countries [2]. As a result, energy consumption becomes high and energy shortage problems occur rapidly. One of the representative examples is the energy crisis problem that occurred during the October war between Arabs and Israel in 1973, when large energy shortage problem appeared, this problem had affected people's lives profoundly. Since then scientists in middle east counties have begun making various researches on finding ways of predicting future energy consumptions as well as reducing environmental pollution. Among these, research works on reducing energy consumption of air conditioning in buildings (especially in prison halls) was extensively studied. Air conditioning represents a high proportion of the overall total energy consumption of prisons of Iraq. Since the energy which is wasted and lost in the transition of prison hall cannot be estimated, the inappropriate design for air conditioning systems and ill-operation of these devices lead to the increasing of the amount in energy consumption as well as rising disbursed costs. Therefore, it is necessary to find ways to reduce power percentage by choosing the appropriate design for HVAC systems which is characterized by insurance thermal requirements that do not affect the comfortable conditions for prisoners [3]. This thesis is considered as one of these efforts to deal with the energy consumption problems in prison halls of Iraq. The thesis is organized as follows: ***Chapter one*** presents an introduction to this thesis. It consists of Introduction, Thesis Organization, Research Problem, Brief History of Air Conditioning, The Meaning of (HVAC), HVAC Requirements, A clean Room Definition, Optimization, Computer Simulation, System approach and the Aim of Research. ***Chapter two*** shows the literature review of HVAC systems, where a number of researches, including Arabic and foreign studies for simulation purposes of energy rationalizations for HVAC systems are explained and compared. ***Chapter three*** explains in detail of the methodologies and the mathematical representation of HVAC systems. ***Chapter four*** is concentrated on the simulation studies and experimental work. ***Chapter five*** concludes this research work and will give guidance for the future studies.

1.2 Research Problems

This thesis is trying to provide some investigation methods for two research problems, engineering problems and technological problems. Below is a brief explanation of these two problems:

1.2.1 Engineering Research Problem

Many air conditioning systems can be used for building facilities especially in prison hall applications. However, the lack of appropriate choices which should assist HVAC engineers to determine optimum systems in terms of providing the thermal requirements with low power consumption may lead to a selection of bad systems (e.g. The system that may consume greater energy than its actual need). These inefficient systems will result some waste in power consumption, rising disbursed costs unless special measurement is taken into consideration. Therefore, choosing appropriate styles to determine the optimal system is essential.

1.2.2 Technological Research Problem

The training of engineers on how to develop analytical skills to make the decision for choosing the optimization system of HVAC, is considered a difficult task with least energy, this problem in any academic study.

In this study, we will focus on mechanical parameters which serve HVAC in prison halls. The process of energy consumption in HVAC systems is affected by several factors; like (prison type, the nature of the thermal act of prison towards the external conditions, type of design and type of air-conditioning system), accordingly we must offer training system by simulating the energy consumption. Subsequently, train the engineers on the skill of making a suitable decision to choose optimum HVAC.

1.3 Brief History of Air Conditioning

The way of the air conditioning developed is only progressively from the predecessor art of cooling, cleaning, heating and ventilation. Leonardo Da Vinci had make a painting, it was relating a ventilating fan, at the 15th century. The first version on heating and ventilating was written by Robertson Buchanan, a Glasgow civil engineer, in 1815 [4]. Towards the second half of the 19th century, the development in the art of humidifying air went along with the progress of cloth industry, so in England the devices for measuring pressure, temperature, humidity and flow air were fulfilling during this period [5].

It is imposing to mention the name of A. R. Wolff who designed an air-conditioning system, he could build 100 devices of his invention. W. H. Carrier (1876-1950) who is also known as the “father of Air Conditioning” while was working with Buffalo Forge Co., developed a model for optimizing the application of forced-draft fan and developed ratings of pipe coil heaters and set up a research laboratory. He patterned and installed the first year-round Air Conditioning system, providing for four major functions of heating, cooling, humidifying and dehumidifying [5].

In 1902, Carrier discovered the link among temperature and moisture and how to control them. In 1904 he developed the air washer, a chamber installed with several banks of water sprays for air humidification. His method of temperature and humidity regulation, achieved by controlling the dew point of supply air, this style is used in many industrial applications, such as lithographic printing plants and textile factory. Perhaps the first air-conditioned office was the Larkin Administration Building, designed by Frank L. in 1906 he completed ducts handled air at the roof. Wright specified a refrigeration mill which distributed 10°C cooling water to air cooling coils in air-handling systems [6]. The U.S. Capitol was air conditioned in 1929 and supplied it from overhead diffusers to keep up a temperature of 75°F (23.9°C) and a relative moisture of 40 percent amid the summer, what's more, 80°F (26.7°C) and 50 percent amid winter. The volume of suppled air was controlled by the supply air; it was controlled by a pressure regulator to deny cold air in the occupied zone [6]. Already the first perfectly cooled office building was the Milan Building in San Antonio, Texas, which was composed by George Willis in 1928. This air conditioning system consisted of one centralized plant to serve the lower floors and many small units to serve the top office floors. In 1937 Carrier developed the channel induction system for multi- room buildings, in which recirculation of space air is invited through a warming/cooling curl by a high-speed releasing airstream. This system supplies only a limited amount of outdoor air for the occupants [6].

1.4 The Definition of HVAC

HVAC implies, heating ventilation and air conditioning that precisely means (the automatic control environment for two things the first one is for the comfort of organisms, principally human, animals or plants. The second aims to correct performance of some manufacturing materials or scientific process). There are several important parameters regarding HVAC like purity, movement, relative humidity, pressure, dew point, velocity of ventilation and temperature. The air pressure must be controlled according to limitations imposed, by the design specification universal standards, so if you want to make HVAC device, you must know these standards like ASHRAE, ASTM, AISI, iklimlendirme, HVAC Licensed and certified and ISO for HVAC manufacturing devices [7]. The standards logos which are seen in Figure 1.1

Figure 1.1: Standards Logos.

1.5 HVAC Requirements

Most people spend time inside the door; approximately 90% and a large fraction of this time is being spent in a residential or a commercial environment. Air conditioning and ventilation occupy the main position in the halls design process as occupants. The anticipation standards, study seriously care about indoor for air quality and comfortable by heating or locales [8]. Everyone has become more aware of the effect of the indoor environment on weather purity as a consequence of media publicity surrounding the Building Related Sickness (BRS). Sick Building Syndrome (SBS), is the principle a complaint about the indoor air quality of the hall which indicates more diffusing in air, air conditioned halls are less pure than naturalistic ventilated halls may be because of increase of carbon dioxide, surely the low in oxygen. (BRS) unite with the feeling of stuffy or throttling, rotten and inadmissibility indoor air, occurs annoying on the snotty nose, headache, apathy. This means that the proportion of fresh air is few, polluted air ratio has become clear and sharp [8]. As for mechanism, the General HVAC Cycle is explained the system's work: There are four important items in every HVAC cycle, compressor, evaporator, condenser and expansion valve as shown in Figure 1.2,1.3[9]

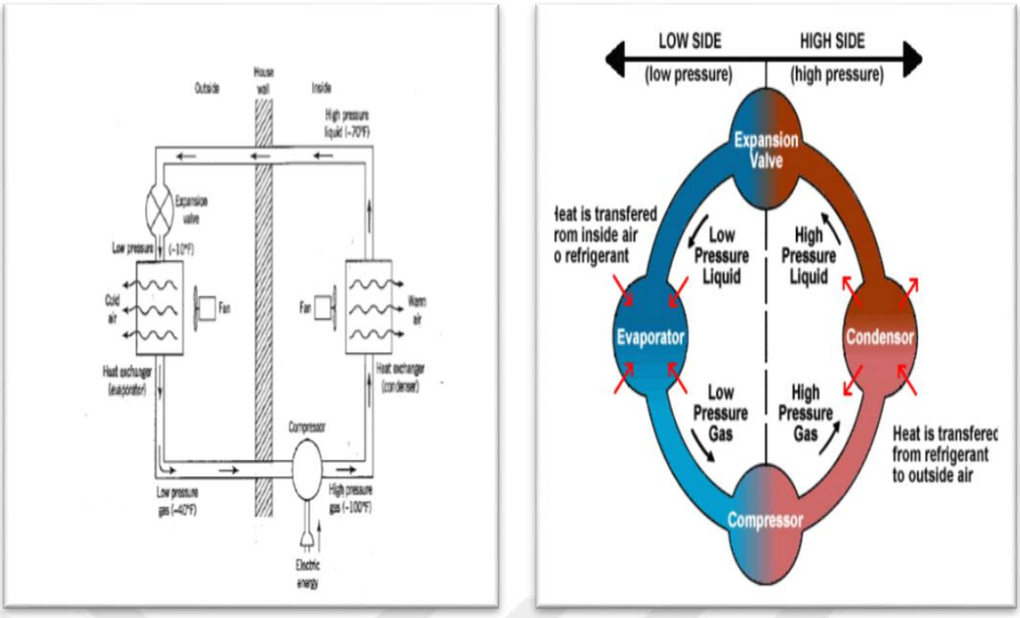

Figure 1.2,1.3: Schematics of General HVAC Cycle [9].

1.6 System Selection

The engineers are heard the expression "plan to budget plan" and, regardless of their endeavors, see great aims and good practices in design diminished in the bidding and award phase because the project is over spending plan. This is absolutely not any more pervasive as far as I can tell than outlining correction facilities. As with many health-care projects, beds are typically the main focus of most correctional facilities. The quantity of informal lodging undertaking's ability to give the required number, sometimes supersedes many other building choices. While selecting a framework, it is certainly vital to consider the primary expenses and settle on sensible choices, in my experience the administrator is basic to this condition also. Who will really be working and keeping up the hardware? Frequently, remedial proprietors don't keep up a staff of exceedingly talented specialists what's more, pros. As the architect, you would prefer not to plan a framework that augments path past the proprietor's capacities or the abilities of their support work force. No measure of care amid outline and development can defeat the administrator's failure to keep the framework kept up what's more, practical long after the venture's consummation. It is additionally essential to comprehend the requirements of the project. In many states, correctional facilities are not conditioned by any means, they are just ventilated and heated. For instance, in Texas, just 19 out of 112 jails have frameworks that give warming and cooling. A considerable lot of the 19 air-condition facilities are for sick and mentally ill prisoners. So we want to correct the design path for HVAC system in prison halls in Iraq to keep pace with global development, prison model in Texas for patients are shown in Figure 1.4 [10]

Figure 1.4: Prisons Models in Texas for Patients [10].

1.7 A Clean Room Definition

The term clean room is generally understood to mean, that any room or area in which an attempt is made to limit and control the indoor contamination [11].

The general control options fall into the following categories:

1. Source elimination.
2. Use of outdoor air.
3. Space air distribution.
4. Air cleaning.

Now a wide variety of industries and applications which require a clean environment. These include classic applications such as:

1. Prison halls
2. Pharmaceutical, biotechnology, product preparation, and genetic engineering.
3. Electronics, semiconductor, microelectronics and disk drive manufacturing.
4. Aerospace.
5. Laboratories for chemical reaction [12].

The courtyard breaks of prisoners which is seen in Figure 1.5 [10]

Figure 1.5: Courtyard Breaks of Prisoners [10].

1.8 Optimization

It is a mathematical way to find the optimum value for a mathematical function that is called an objective function. This function depends on a number of variables which a subjected to a number of limitations and restrictions in the form of equations or ranges, all of them is called constraints. The best choice for HVAC system, intended in this thesis, it is to identify the least energy-consuming system [8]. I took ABO-GREAB prison model in Iraq, which is shown in Figure 1.6, as a mathematical model and simulation pattern in chapter 4.

Figure 1.6: ABO-GREAB Realistic Prison Model.

1.9 Computer Simulation

This term is used to describe the software programs which are dealing with math or logical models on engineering operations. This simulator allows user to supervise and watch the computation with testing the results values and compares them with changes in other systems [13], [14].

1.10 System Approach

A design represents an integrated system and plans to work all the elements and components in operations to achieve a particular goal. This system consists of four elements; input, processes, output, feedback [15].

1.11 Purpose of the Research

This research seeks to achieve the following objectives: -

1. Study the possibility of rationalization of energy consumption in HVAC systems by preparing of computer software to simulate the Energy Consumption (EC) for different HVAC systems in prison halls in Iraq.

2. Determine the best system which is least energy consumption by choosing the optimum system for prison halls in Iraq.
3. Employ the principles and foundations of engineering education technology especially with regard to the technological features of computer as a part of the training of engineers based on simulation of (EC) for HVAC through previous program as an assistant within the computer simulation software, then raise the skill level of HVAC designers.
4. Evaluate the efficiency of the performance of computer program which has been prepared in this from the point of view of academic experts in air conditioning and educational technology department as well as beneficiaries of guests “prisoners”.
5. This study offers another way in reduction of HVAC energy by taking advantage from determination of the period of time that can stop the work of cooling equipment then operating units push the air and allow to outside air intakes to admit it to the hall for cooling, operating the system in the least energy consumption rate.
6. Possibility of using thermal loads in prisons by simulating the purpose of designing.
7. The research contributes the engineers in gaining some knowledge skills in clarifying the impact of differences each of (HVAC system type, design type, type of used equipment, prison type, building materials that were used) all of them and relationship with (EC), therefore subsequently determine the best choice for HVAC system for any prison in Iraq.
8. The possibility of benefiting and study utilization in: -
   A- Engineering consultancy center.
   B- Maintenance engineers in prisons.
   C- Continuing education centers which dealing with skills and improve the mental abilities.
   D- Colleges, mechanical engineering department

CHAPTER TWO

REVIEW OF LITERATURES

2.1 Introduction

This section introduces a concise depiction of the distributed writing and recorded trials brought to the field of HVAC parameters. In spite of the fact that the rundown ought not to be viewed as thorough, the survey does adequately speak to the advances, discoveries, and commitments which are of specific importance to this thesis. This review is subdivided into three main topics, namely Experimental Works, Numerical Simulations and Experimental with Numerical Studies.

2.2 Experimental Work

The main objective of **Surrander Naganathan, PE** [16] was to reduce the Energy Cost at Deer Ridge Correctional Institution in Madras, USA. Which is seen in Figure 2.1

Figure 2.1: Correctional Institution in Madras, USA [16].

Energy conservation measures at Deer Ridge Correctional Institution resulted in savings in each category, which is seen in Figure 2.2, with the most significant savings gained in heating and cooling.

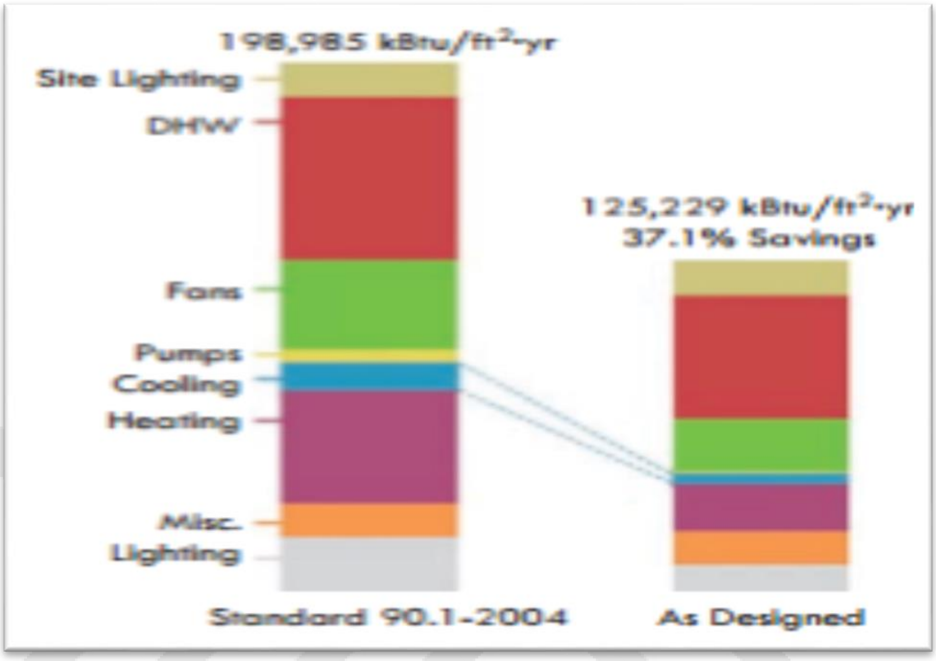

Figure 2.2: Energy Use Intensity Comparison [16].

The following summarizes the major energy conservation measures. In a review examined in an effective electric lighting framework, the Oregon Energy Code (OEC) permits 1.27 W/ft^2 (13.66 W/m^2) for pattern displays. Through the introduction of an ultra-efficient electric lighting system using high performance T8 lamps (3,200 initial lumens and 24,000-hour lamp life) and carefully matching these lamps with their optimal electronic ballasts, the design achieves an actual lighting level of 0.91 W/ ft^2 (9.79 W/m^2), a reduction of 28% from the baseline criteria. High output electronic ballasts overdrive the lamp current by approximately 25% as compared to standard electronic ballasts, reducing the number of lamps needed to achieve the required baseline foot-candle levels. These measuring data about a yearly electric saving are reserving funds of 330,829 kWh every year. With regard to innovative cooling, we understand that while evaporative systems are not innovative, they are (to the team's knowledge) the first time that this type of system had been used in a prison and to a degree that resulted in nearly every spaces being cooled. No refrigerated HVAC cooling is used on campus with the exceptions of the telecom, security electronics and control rooms. To increase the system's efficiency and have it perform at targets levels, the team modified the housing unit design to increase cooling efficiencies and created a system based on multistage evaporative cooling with heat recovery. As Figure 2.3 details, the system still delivers 100% outside air into the building, but uses exhaust air combined with indirect evaporative cooling in a heat recovery section to pre-temperature supply air entering from outside. Outside air passes during evaporative cooling by entering the ducts. Additionally, the prison design presented the team the scope to further increase energy efficiency by modulating the water tank and pump system by employing the enter cooling unit. While a model unit design as seen in Figure 2.4, includes a tank and pump situated within each unit, the team instead devised a tank/pump system that was exterior to the unit. The design very necessarily implements a domestic environmental quality to prevent a Legionella Bacteria, Alga growth. The researcher team physically examined the units themselves, which was a massive task given the sizes of the units. The researchers resolved the experimental parameters and set to have the units sent to the only testing lab in the U.S. capable of handling a test of such size. The lab simulated the site's environmental stipulation (88°F [31°C] at 20% relative humidity) in which the units were tested to check if they could eject 59°F (15°C) air. An experiment of this measure, to the team's knowledge, had not previously been undertaken, and it was a critical that executions be enclosed in this system so that it would meet the needs of the prison [16].

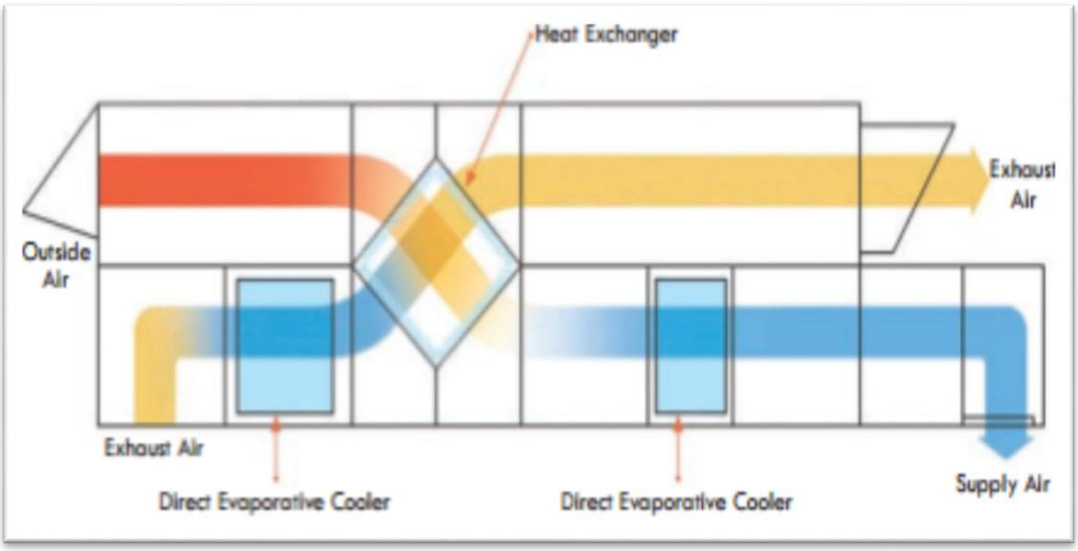

Figure 2.3: Multistate Evaporative Cooling Unit with Heat Recovery [16].

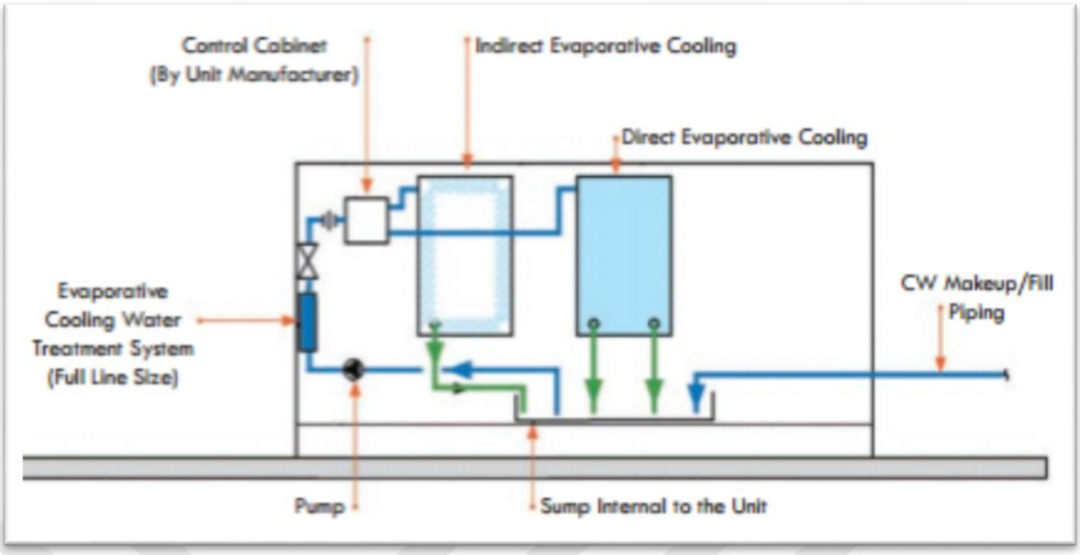

Figure 2.4: Typical Evaporative Cooling Unit [16].

2.3 Numerical Simulations

Crall, et al., [17] simulated the results which present the thermal domains and HVAC device for the usually modeled room under the cooling. An air flow analysis was carried out based on the Cooling Load Temperature Difference Method (CLTD) with a building type boundary condition. The temperatures of a topic, such as walls were obtained by resolving the heat balance equation, in which the convective and radiative heat transfer were integrated. The numerical simulations were conducted for room space under the cooling condition in which 250 m^3/h of conditioned air at 23 °C, and the same volume of Outside Dry-Bulb 31 °C. The gained results for the consumption range were provided at the position is nearly to the building. So, the power of the outside air is not drastic in the operation area. It is possible to maintain a comfortable environment in the operation area. **A. D. Gosman**, **et al.,** [18] this study is conducted in mechanical engineering department- Royal College of Science and Technology in London. The objective of the research is served by using the computer for cooperating in education, so called Computer Aided Learning (CAL). The main goal of this / is to express a more realistic exploratory design for jails. The researcher studied the

Computational Fluid Dynamics modeling (CFD) / and made comparison or results with experimentally obtained data. The main aim is to find the optimum design of ventilation air conditioning system inlets, in order to increase the efficiency of HVAC devices and increasing the skills of trainers.

Zhu, et al., [19**]** simulated a two and three dimensional incompressible turbulence of indoor air in a turbulence model. To simplify the issues, the models assumed that the indoor air was in an incompressible, constant properties, steady-state flow and concentration with the basic assumption of Boussinesq equations. Air infiltration was not considered. The disturbance model, considering the effect of the buoyant force, selected a two-equation model with a High-Reynolds number with little air-supply quantity and depressed air velocity in the air-conditioned area. Therefore, the supply velocity was 0.3 m/sec. **Wei Cai, Xubo Yu.,** [20] have investigated the method of (CFD) for air distribution and comfort in a typical room with cold air diffusion. A mathematical model was established and air conditioning was simulated. The velocity and temperature area under different conditions were analyzed. The results demonstrated that the air plunge and the air attachment were easily generated. Therefore, the series of the above numerical simulation was achieved to advocate the numerical predictions in order to ensure that the simulations can be used with confidence to amend the designer's innovations in the HVAC computer simulation. **Hamilton, et al., [**21] studied the performance of a normal air conditioner (window type) by constructing a software program to simulate the influence of air conditioning. The researchers adopted the manufacture's performance schemes to prepare a mathematical algorithm about the compressor, condenser, evaporator, expansion valve and fans. The dry-bulb temperature ranged from 29.4°C to 40.5°C. The program adopted the diameters of the capillary tubes, from 0.08" to 0.058" The study explained that any increase in external dry-bulb temperature would lead to an increase in the compressor potential and Coefficient of Performance (C.O.P). Moreover, the increased diameter of the capillary tube would lead to an increase in the cooling capacity up to a diameter of 1.32mm, after which the cooling capacity starts to decrease. The dynamic characteristics of indoor data, such as air temperature, moisture, and contaminants, can be simulated.

Carbontrust Journal., [22] presented a focus on the adjusting the system. At times, it was logical to use the outdoor temperature to set the conditions inside the hall. However, when the HVAC system was on, it was likely to save up to 30% on energy costs by reducing the quantity of outside air entering a building. It is always better to modify the framework instead of opening an entry way or a window and allow warmed or cooled air to pass. When it is too hot, people resort to opening windows or doors to have their space become more comfortable. In order to ensure that staff understands the implications of opening windows and doors when air-conditioners are in use, personnel are encouraged to adjust the thermostat instead. This study took the following in to consideration:

-Replacing traditional boilers with condensing boilers.

-Replacing standard motors with high capacity motors.

-Opportunities for heat recovery and recirculation in the building.

-Installing a building energy management system (BMS or BEMS) which offers close control and monitoring of building services and performance, including HVAC.

-Investing in Variable Velocity Drive Systems (VSDS) for motors to match velocity with output demands.

-Exploiting in direct drive pumps and fans, which are more efficient than belt-driven drive pumps and fans.

The results are offered on a computer screen in real time and can be monitored. Settings can be changed quickly and distinctly. Furthermore, BEMS can reduce the energy price by up to 10%.

2.4 Experimental with Numerical Studies

Gluck and Pollack, et al., [23] investigated the cooling capacity and temperature division in a building hotel using a hot air insert at different levels. Numerical solutions for the energy equation and Naiver Stokes Equations were obtained through a Semi-Implicit Method for pressure with algorithms equations that have steady, laminar and incompressible flow. A selected cubic room or hotel 3m×3m×3m with two positions of the inlet and outlet was assumed. Solutions were presented for several locations including the inlets and outlets and for different values of internal loads. An increase in the Grashof Number increased the power of the recirculation and the product symmetrical temp division. **Silverman, et al.,** [24] studied the supplied air that was uniformly distributed in an air conditioned area under various conditions. An experimental study was first conducted to obtain more general information about the air conditioning machinery on indoor air quality problems in divided offices. The planned environment consisted of two interconnected rooms, the dimensions of each being 6 m×4 m×3 m height. The contaminant distribution was influenced by almost every type of HVAC parameters in the office with different diffusers. The outcomes, demonstrating that the contaminant escaped a mechanically acclimatized divided office, need to be studied individually according to different cases. CFD simulation is an efficient tool for such kinds of studies. **Ramiz, et al.,** [25] explained the results of an experimental system for measuring the performance of air conditioning units that are designed, constructed and numerically simulated in a three-dimensional distribution of testing rooms with

Figure 2.5: Code Tester Device, Experimental with Numerical Work [25].

the computational fluid dynamics technique. The code tester device is shown in Figure 2.5. Therefore, the researcher has selected for this thesis the telecommunications building in Baghdad. The following three substitute designs in this building were studied:

1-Using a centralized refrigeration plant inside the crypt.

2-Putting a centralized refrigeration plant on the fourth floor.

3-Using four substations distributing refrigeration plants on the ninth floor

The twelve measuring points were located isometrically at the intersection of three horizontal and three vertical lines used to measure the velocity and temperature. All the measuring points after the air environment were studied. The computational fluid dynamics software FLUENT was used in this study based on the K-ε turbulence model. The air flow and heat transfer problems were resolved by finite volume method, the experimental data and simulation results showed that the speed and temperature of the testing room disseminated homogenously, and the indoor air environment assembly tested the demands of the air conditioning unit. The simulation output proved that the supply air parameters and the location of the testing unit have an apparent effect on the air distribution. Then, the results showed that the four sub-refrigeration plants are the highest cost and energy consumption than the plant in the fourth floor.

Hamilton and Qingyan Chen., [26] investigated 3D air flow and contamination dispersal and compared the results of the relatively simple 3D numerical study, CFD with particle image velocimetry (PIV) experimental measurement of indoor temperature in a one-tenth sub- scale model room.

The side branches of the room were constructed with aluminum and had four glass windows, which provided a sufficient visual vent. Numerical turbulence models were used and evaluated with respect to their performance in the simulation. The flow in the model room and results of the numerical simulations and velocimetry measurement showed how obstructions can greatly influence the air flow and contaminate its transport in to the room. It is important, therefore, that obstructions be considered in air conditioning room design.

CHAPTER THREE

Methodology Scope

3.1 Introduction

For informational purposes, we noticed from Iraq the solar energy accounts that 21 June is the longest day in the year. The max temperature and humidity in July. The shortest day at 21 December. The coldest day in January or February. The max temperature occurs at 3 afternoons, throughout the day. In addition, the max amount of heat entering the room depends on the falling solar radiation about the third hour in the afternoon, this heat transfers by conduction in the walls and roofs it occurs at this time, because the default time. The default time can be varied defending on the difference time between the fall solar radiation and its transition into the room [27].

3.2 Calculation of the Needed Energy for Heating and Cooling

3.2.1 The methods used to calculate the necessary energy for heating and cooling purposes

3.2.1.1 Degree-Day Method

It is one of the simples and most common methods which is used to calculate the required energy of heating. It relies on two assumptions. The first assumption is a moderate temperature, outside air, it's called Base Temp, we are assuming at this degree, there is no needing for heating. This means the resulting load parameters came from solar radiation, generating heat inside space is equal to leaking heat from inside to outside space. As far, the second assumption, is based on existence of extrusive direct proportion between the amount of energy required for heating with the difference between the average temperature of the air and Base Temp. We can identify the amount of expended energy for heating from equation 3.1[28]

$$E = \frac{H \times DDH \times 24 \times CD}{\Delta t \times R} \tag{3.1}$$

Where

E: Thermal energy KJ.

H: Heating load KJ/h

DDH: The number of heating days under Base Temp day.

Δt: Temperature difference in between the exterior and interior of dry bulb °C.

R: Utilization factor 0.6-0.8, equal 1 when using electricity.

CD: Correction factor for heating days, it depends on Base Temp [28].

It can be using the previous method of estimating in the necessary energy consumption, for cooling purposes. Which we are calculating the temperatures, above Basic Temp for each day of the estimation period, then combines these temperature degrees, so it represents the value of (temperature × day). It can compute expended energy value from these equations 3.2, we can calculate the number of cooling days from equation 3.3 [29].

$$Q = UA \times 24 \times DD_C \tag{3.2}$$

$$DD_C = \sum_1^n \frac{1}{24} \sum_1^{24} (Tb - Ta) \tag{3.3}$$

We can write equation 3.3 by using the average temperature during the day

$$DD_C = \sum_1^n \quad Tb - \frac{Tmax + Tmin}{2} \tag{3.4}$$

UA: Overall heat transfer coefficient.

Tb: Basic Temperature (°C).

Ta: Outside temperature rate at every hour (°C).

Tmax: Maximum value rate of monthly outside temperature at (°C).

Tmin: Minimum value rate of monthly outside temperature at (°C).

It should be noticed that if we are using this method for heating status, it is better than for cooling status because this method depends on temperature difference only, but cooling loads are depended on other factors like, solar radiation and indoor loads for space [29].

3.2.1.2 Simplified Multiple Measure Method

The previous method based on an analysis of the cumulating values for single factor only outside temperature, this way was limited; it cannot be used in case of a large building, it causes a large error in guessing, because of the negligence of periodical changes per factor [29]. But Bin method, is ×one of the common ways, especially in this field. It depends on the heating or cooling load calculation, for building a number of bins for outside temperature and from analysis of changing the degree of the actual hours. Multiplying the number of hours by a thermal load for each bin to obtain the thermal load for completely group of the required energy for air conditioning [30].

3.2.1.3 Equivalent Full Load Hours Method

In this method, the necessary energy for cooling is guessed by depending on the number of operation hours of the air for maximum energy equivalent, for actual operating hours where the air conditioner compressor doesn't constantly work even during the actual operating hours, but it stops for some times then return to work, depending on the cooling space degree. The total number of equivalent working hours depends on the quality of the building, construction, combination, geographical location, and nature of use. Then we can evaluate the number equivalent hours after making statistics within different areas [30].

The electric power can be calculated from equation 3.5:

$$E = \frac{L \times E.F.L}{C.O.P} \tag{3.5}$$

Where:

L: The maximum cooling load.

E.F.L: Number of the total equivalent load hours (h).

C.O.P: Coefficient of Performance for cooling equipment including fans, and pumps [30].

3.2.1.4 Detailed Simulation Method

The previously applied methods were inaccurate because of the negligence of some important effecting factors which are effecting on disbursements energy level as, thermal performance of the building, external and internal influences, and thermal dynamic proceedings of HVAC system [31]. Because of the difficulty of putting a real model of the building for holding an account then is redacted mathematical model to describe the building, it includes representation of thermal performance, is named load model. As well as the external and internal influences, thermal dynamic proceeding for HVAC, the system is named, system model. In addition, the establishment of a mathematical equations were linking between energy and thermal load, a program is named, plant model. The program runs by entering the effective difference date, after analysis; we will know the needed energy to operate the heating, ventilation and air conditioning systems with the details [31].

3.2.2 Air Conditioning Load calculation

The heat that should be removed from the building is named “cooling load” but, the heat that should be provided to the building is named “heating load”. The purpose in both cases is to get the comfortable designing conditions in summer and winter. The calculation of the air conditioning loads is considered the first step to choose the HVAC equipment and limitation of the amount of energy consumed [5].

3.2.3 Sources of Air Conditioning Loads

The Conditioning loads are divided into two parts [5]

A- External source: -

- Heat gain through the wall, ceilings and floors.
- Heat gain through the doors.
- Heat gain through the windows.

- The resulting heat flows from the leakage and ventilation.

B- Internal sources especially for cooling load: -

- Peoples.
- Lighting.
- Service devices.
- Secondary sources

3.2.4 Cooling Loads Calculation Method

We are calculated the cooling loads, depending on cooling load temperature difference method (CLTD). We can find the tables available which explain the temperatures differences change for [5]

Various types of ceiling for 24 hours a day.

Various types of walls for 24 hours a day and for 8 different directions.

Various types of glasses for 24 hours a day [5].

These data are based upon standard conditions in Iraq, Baghdad.

External dry bulb temp 35°C.

Internal dry bulb temp 25.5°C.

Average external temp 29.4°C.

The range of change in daily temp 11.6°C.

The month at July.

Orbit degree, Iraq is located between °29-°37 north of the equator.

The color of the exterior walls, dark color.

The color of the exterior ceiling, dark color [32].

If we change above conditions, we would make corrections as follows:

A- We can calculate the correction factor for external dry bulb temp and the range in daily temp, from equations 3.6, 3.7, 3.8 [1]

$$TOR = TOD - 0.5\ DR \quad (3.6)$$

TOW = TOD – 0.5 DR. (3.7)

TOG = TOD – 0.5 DR . (3.8)

Where DR represents the range of daily temp change, from equation 3.9 [1].

DR = TMAX – TMIN. (3.9)

TOR: Ceiling correction factor for external dry bulb temp.

TOW: Walls correction factor for external dry bulb temp.

TOG: Glasses correction factor for external dry bulb temp.

A- The correction factor for external ceiling, Change Color Factor (KR),1 for the Dark-color ceiling [5].
0.5 for Light-color ceiling.

B- The correction factor for external wall, Change Color Factor (KW),
1 for external Dark-color wall.

0.83 for Medium-color wall.

0.65 for dark-color wall [5].

C- We can calculate the correction factor for internal air temperature (KI) we can calculate KI from equation 3.10[5].
KI = TID – 25.5. (3.10)
Where:
TID: Indoor Temperature.

3.2.5 Calculation of space Cooling Loads.

It includes calculation of external, internal load sources for space conditioning. For general calculation, is calculated from the Fourier equation. If we consider that the wall section is shown in Figure 3.1[33].

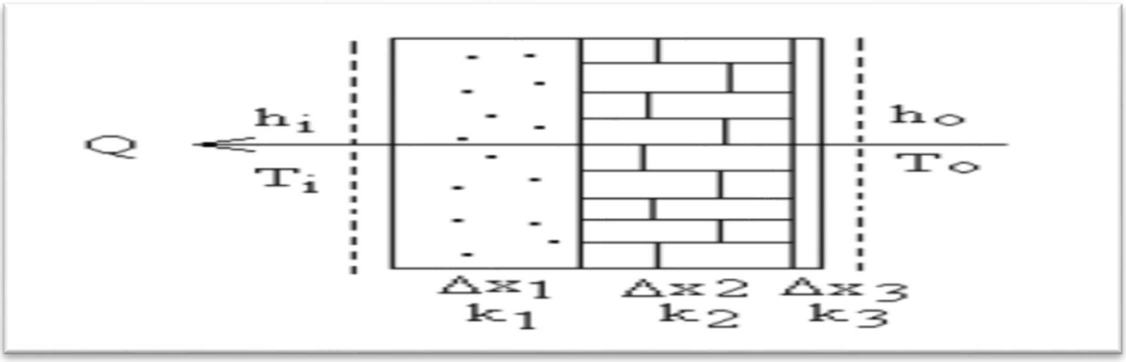

Figure 3.1: Wall Section Model [33].

It contains different materials, such as building bricks, solid gypsum, heat insulation and air layers. The heat transfer coefficient can be calculated which is seen in equation 3.11[33]

$$\frac{1}{U} = \frac{1}{h_i} + \frac{\Delta x_1}{k_1} + \frac{\Delta x_2}{k_2} + \frac{\Delta x_3}{k_3} + \frac{1}{h_0} \qquad (3.11)$$

Δx: Represents the thickness of the constituent materials of the wall or roofs.

k: Thermal Conductivity Coefficient for each material.

hi, ho: Thermal Convection Coefficient.

U: Total thermal transmitted coefficient.

A: Area of the transition surface.

QT: Total Heat Transfer.

We can write equation 3.11 after pithiness to calculate the inaccurate total heat transfer for wall section model from equation 3.12 [27]

$$Q_{wall\&\ Roofs} = U \times A \times \Delta T. \qquad (3.12)$$

Table 3.1 depicts the Heat Transfer Coefficient of the Surface [34].

Table 3.1: Heat Transfer Coefficient of the Surface W/m2×°K [34].

	Heat Direction	h W/m² × °K
Stable air		
Horizontal	To Up	10
Horizontal	To Down	6
Vertical	Horizontal	8
Moving air		
At velocity 6.7m/sec	In any direction	34
At velocity 3.4m/sec	In any direction	23

The thermal conductivity coefficient is shown in table 3.2[34]

Table 3.2: Thermal Conductivity Coefficient [34].

Material	K W/m× °K
Common brick	0.72
Face brick	1.30
Concrete	1.72
Tiles	1.10
Stone	1.80
Cement plaster	0.72
Gypsum plaster	0.80
Hard wood	0.16
Soft wood	0.12
Sand	1.72
Cork	0.036
Glass wool	0.036
Polystyrene	0.040
Polyurethane	0.023
Glass	0.79

3.2.5.1 Transmission of Heat Through the Ceiling

A- Heat gain through the ceilings exhibit to the outside

It can be defined as an eventual heat, from roof thermal gain which is seen in equation 3.13 [27]

$$CLTD.CR = [(CLTDR + LMR) \times KR + (TID - 25.5) + (TOR - 29.4)] \times F \qquad (3.13)$$

F: Ventilation factor between a secondary ceiling and a tilt surface (Attic), there is 1 when no ventilation fans or air duct between the roof and secondary ceiling.

F = 0.75 when there is no mechanical ventilation in space gap.

We can calculate the heat transfer for ceiling from equation 3.14 [27]

$$Q\ Roof.\ cout = U\ Roof.\ Cout \times A\ Roof.\ out \times CLTD.\ CR \qquad (3.14)$$

B- Heat gain through the non- conditioned space for the ceilings

Which is seen in equation 3.15 [35]

$$QRoof.Cin1 = U\ Roof.in1 \times A\ Roof.in1 \times (TOD - TID) \times 0.667 \qquad (3.15)$$

C- Heat gain through the non- conditioned space for the roofs with generation heat source

Which is seen in equation 3.16 [5]

$$QRoof.Cin2 = U\ Roof.in1 \times A\ Roof.in2 \times (TOD + 3.88 - TID). \qquad (3.16)$$

3.2.5.2 Transmission Heat transfer through the Walls

A- Heat gain through the walls exhibit to the outside

we can calculate this item from the equation 3.17 [5]

$$CLTD.CW = [(CLTDW + LMW) \times KW + (TID - 25.5) + (TOW - 29.4)]. \qquad (3.17)$$

We can calculate the heat transfer for wall from equation 3.18 [5]

$$Q\ Wall.\ Out = U\ Wall.\ Cout \times A\ Wall.\ Out \times CLTD.\ CW. \qquad (3.18)$$

B- Heat gain through the non- conditioned space for the walls

we can calculate this item from the equation 3.19 [35]

$$QWall.Cin1 = U\ Wall.in1 \times A\ Wall.in1 \times (TOD - TID) \times 0.667. \quad (3.19)$$

C- Heat gain through the non- conditioned space for the walls with generation heat source

we can calculate this item from the equation 3.20 [5]

$$QWall.cin2 = U\ Wall.in1 \times AWall.in2 \times (TOD + 3.88 - TID). \quad (3.20)$$

The total heat transfers for many kinds of walls, which is shown in table 3.3 [34].

Table 3.3: Total Heat Transfer [34].

Items	U $W/m^2 \times °K$.
Plain brick wall	1.35
Wall of stones	1.88
Concrete wall	2.26
Hollow Concrete wall	1.83
Insulate Roof	1.3
The roof and floor	1.75
Wooden floor	2.44

3.2.5.3 Transmission Heat Through the Doors

A- Heat gain through the doors exhibit to the outside

We can calculate this item from the equation 3.21 [5]

$$Q_{Door.cout} = U_{Door.cout} \times A_{Door.out} \times (TOD - TID). \tag{3.21}$$

B- Heat gain through the non- conditioned space for the doors

we can calculate this item from the equation 3.22 [35]

$$Q_{Door.cin1} = U_{Door.in1} \times A_{Door.in1} \times (TOD - TID) \times 0.667. \tag{3.22}$$

C- Heat gain through the non- conditioned space for the doors with generation heat source

we can calculate this item from the equation 3.23 [5]

$$Q_{Door.cin2} = U_{Door.in1} \times A_{Door.in2} \times (TOD + 3.88 - TID). \tag{3.23}$$

3.2.5.4 Transmission of Heat Through the Glasses

Firstly, we can calculate the total heat transfer coefficient from equation 3.24. Then we can calculate the conduction heat transmitted and collection with convection heat, we will get approximation the Total thermal transmitted. Heat gain through the windows can be calculated from equation 3.25 [33]

$$\frac{1}{U} = \frac{1}{h_i} + \frac{\Delta x}{k} + \frac{1}{h_o} \tag{3.24}$$

$$Q_{Glass} = U \times A \times \Delta T. \tag{3.25}$$

Where hi, ho in W/m²× °C.

Now we calculated by second way, in details and accuracy:

A- Solar heat gain: - we can find it by equation 3.26 [5]

$$Q_{GLASS.SOLAR} = A_{GLASS.OUT} \times SC \times SHFG \times CLFG. \tag{3.26}$$

B- Conduction heat gain: we can find it by the equations 3.27, 3.28 [5]

$$CLTD.GC = [CLTDG + (TID - 25.5) + (TOG - 29.4)]. \quad (3.27)$$

$$QGLASS.\ count = UGLASS.cout \times AGLASS.out \times CLTD.GC. \quad (3.28)$$

C- Heat gain through the non- conditioned space for the glasses

We can calculate it from equation 3.29 [35]

$$QGLASS.CIN1 = UGLASS.CIN1.AGLASS.IN1 \times (TOD - TID) \times 0.667. \quad (3.29)$$

D- Heat gain through the non- conditioned space for the windows with generation heat source

We can calculate it from equation 3.30 [5]

$$QGLASS.CIN2 = UGLASS.IN1 \times AGLASS.IN2 \times (TOD + 3.88 - TID). \quad (3.30)$$

The Heat Transfer Coefficient of the Glasses, which can be seen in Table 3.4[34]

Table 3.4: Heat Transfer Coefficient of the Glasses [34].

Glasses quantity	U $W/m^2 \times °K$
One	6.4
Two with 13mm gap	3.2
Three with 13mm gap or space	2.2

3.2.5.5 Transmission the Heat gain Through the Floor(earth)

It is being calculating from the equation 3.31[5].

$$QFloor.c = P.\ FLOOR \times K \times (TOD - TID). \quad (3.31)$$

Where:P. FLOOR: Ground Perimeter which exposed to the outside in (m).

3.2.5.6 Heat due to Infiltration

Sensible heat load: It is calculated from equation 3.32 [5]

$$QINF.S = 1.22\times (TWC\times X\times TDC\times Y) \times (TOD - TID). \tag{3.32}$$

Or it is calculated from equation 3.33 [35]

$$QINF.S = 1.22 \times VINF\times (TOD - TID). \tag{3.33}$$

Latent heat load: It is calculated from equation 3.34 [5]

$$QINF.L= 3012* (TWC\times X + TDC\times Y) \times (WOUT - WIN). \tag{3.34}$$

Where: 3012 humidity ratio difference.

Or it is calculated from equation 3.35 [35]

$$QINF.L= 3012\times VINF\times (WOUT-WIN). \tag{3.35}$$

3.2.5.7 Heat Gain Through the Lights Emissions

Lighting heat additions are a critical patron to a building's cooling load. A careful understanding of the lighting heat gains is necessary to accurately calculate building thermal loads and at last, decide the proper limit with respect to the HVAC frameworks. Convection and thermal radiation are two main heat transfer forms in converting electric energy to heat gains while any warmth exchange by conduction can be overlooked (ASHRAE 2013a). Convective heat exchange is an instantaneous thermal gain in the conditioned space and therefore an immediate cooling load. Emanate warmth is ingested after some time by the surfaces and parts (dividers, furniture, etc.) in the room therefore, any radiative heat gains have a time lag effect on the cooling load. It is essential to decide the split between convective warmth and radiative warmth in order to accurately calculate the space cooling load. Another key considers lighting heat pick up circulation is the part between the adapted space heat gain and ceiling plenum heat gain, since typical commercial building zone spaces utilize roof tiles to isolate adapted space and plenum space. Albeit 100% of electric energy going into a lighting luminary is converted to heat gains of its surroundings, only the warmth exchanged to the molded space is considered as space cooling load. For recessed and surface-mount luminaries, a portion of the lighting heat gains is transferred into the roof plenum. This bit of warmth is then exchanged back to the air-taking care of unit (AHU) via its return air, contributing to the overall AHU coil cooling load. An exception is the plenum air being recycled back to the adapted space when fan-controlled Variable Air Volume (VAV) terminal units are used [36], [37].

Heat Gain through Lights emissions is calculated:

A- Heat gain from Fluorescent lamps is calculated from equation 3.36 [36]

$$Q_{Light.F} = NOF \times SAF \times CLFL \times WATT.\ F. \qquad (3.36)$$

Where:

SAF: Safety permittivity factor for fluorescent, always equal 1.25.

The optimum value of illumination intensity is shown in table 3.5[36]

Table 3.5: The Optimum Value of Illumination Intensity [36].

Intensity W/m²	The quality of building use
60-120	Offices
500	Tiny works
1000	Very precise works
50	Factories or universities
15	Prisons, Museums

B- Heat gain from Glowing wire lamps. It is calculated from equation 3.37[5]

$$Q_{Light.T} = NOT \times WATT. \qquad (3.37)$$

C- Heat gain from Led lamps. The manufacturer is indicated the value of the heat gain for every type [37].

3.2.5.8 Heat Gain Through the People's Bodies

A- Sensible Heat gain from the bodies, which is given in equation 3.38[5]

$$QPEOPLE.S = NOP \times S.HGP \times CLFP. \tag{3.38}$$

B- Latent Heat gain from the bodies, which is shown in equation 3.39[5]

It is intended the resulting water stream of breathing, perspiring from space occupants. In addition, associated moisture with air filtration and ventilation.

$$QPEOPLE.L = NOP \times L.HGP. \tag{3.39}$$

Table 3.6 depicts the values of the sensible heat and latent heat for peoples [34]

Table 3.6: Peoples Heat [34].

People condition	Uses	Sensible Heat in Watt	Latent Heat in Watt	Total in Watt
Sitting and comfortable	Stage	66	31	97
Sitting and simple work	Office	72	45	117
Standing and simple work	supermarket	73	59	146
Walking slowly	supermarket	73	73	146
Sitting	Bank	81	81	162
Worker moves	Factory	110	183	292
Person playing sport	Play ground	170	255	425

As shown in table 3.7 depicts the optimum Ventilation Rates for every person [34]

Table 3.7: Ventilation Rates **L/S/Person** [34].

Uses	Smoking	Ventilation Rates		Times change air number at every hour
		Min	Favorite	
Flat	May be	7	9.5	1
Bank	Maybe	5	7.5	1.5
Salon	Maybe	5	7	1.5
Supermarket	Forbidden	2.5	3.5	2
Factories	Forbidden	3.5	5	3
Hospital	Forbidden	12	14	7
Meeting room	Strong	14	24	11
Theaters	Forbidden	2.5	5	9

3.2.5.9 Heat Gain from the Service Devices

A- Service device with vacuum discharge, equation 3.40 is obtained [5]

$$Q.\ APSH = CLFAH \times SH.G.H \times NO.\ APSH \quad (3.40)$$

B- Service device without vacuum discharge.

B-1 Sensible Heat gain, equation 3.41 is valid [5]

$$Q.APS = S.H.G \times CLFA \times No.APS. \quad (3.41)$$

B-2 Latent Heat gain, equation 3.42 is obtained [5]

$$Q.APL = L.H.G \times NO.APS. \quad (3.42)$$

Table 3.8 below shows the burner heat [34].

Table 3.8: Burner Heat [34].

Burner Type	Without Hood (W)		With Hood (W)
	Sensible Heat	Latent Heat	Sensible Heat
Coffee store	515	220	150
Fire place	930	525	290
Toaster	1050	700	350
Chicks grill	2190	2190	875

3.2.6 Total Cooling Load for Enclosed Space

The total Sensible heat gain, equation 3.43 is valid [5]

(RSH) = QRoof.C + QWALL.C + QDoor.C + QGlass.C + QLight + QFloor.c + QInfs + QPeople.S + Q.APS + QAPSH. (3.43)

The total Latent heat gain, which is given in equation 3.44[5]

(RLH) = QPeople.L + QInfs + Q.APL. (3.44)

The total Cooling Loads for enclosed space are included by summation the sensible and latent loads for enclosed space, which are given in equation 3.45 [5]

RTH = RSH + RLH (3.45)

3.3 Calculation of the Cooling Loads Coil

3.3.1 Total Loads for Cooling Coil

A- Heat gain which resulting from ventilation it divides into two types

1- Sensible Heat gain, equation 3.46 is obtained [5]

QASH = 1.232 × NOP × (L/S) person × (TOD – TID). (3.46)

2- Latent Heat gain, we can calculate it from equation 3.47. [5]

$$QALH = 3012 \times NOP \times (L/S)\ person \times (Wo.C - Wi.C). \quad (3.47)$$

B- Heat gain which resulting from additional loads, it divided into two types

1- Sensible Heat gain (RSHS*): heat gain from Air ducts and Air pushing fan.

These loads are calculated as a percentage from (RSH), it can be estimated about (5-15) % from (RSH) [38].

2- Latent Heat gain (RLHL*): heat gain from Air ducts and Air pushing fan. This load is calculated as a percentage from (RLH), it can be estimated about 5% from (RLH) [38]. Thus, we can calculate the Grand Total Heat (GTH) for the cooling coil by summation all the loads, sensible and latent [38]

$$TSH = RSH + QASH + RSHS^*. \quad (3.48)$$

$$TLH = RLH + QALH + RLHL^*. \quad (3.49)$$

$$GTH = [TSH + TLH]\ . \quad (3.50)$$

3.3.2 Calculate some design variables

A- Design variables which related with moisture map (psychometric chart) [38]:

1- Effected sensible heat for the space, which is given in equation 3.51

$$ERSH = RSH + (BF \times QASH). \quad (3.51)$$

Where:

BF: By Pass factor equal 1.

2- Effected latent heat for the space, which is shown in equation 3.52

$$ERLH = RLH + (BF \times QALH). \quad (3.52)$$

3- Effected total heat for the space, as in equation 3.53

$$ERTH = ERSH + ERLH. \quad (3.53)$$

4- Effected sensible heat factor, equation 3.54 is valid

ESHF = $\frac{ERSH}{ERTH}$. (3.54)

5- Room sensible heat factor, equation 3.55 is valid

RSHF = $\frac{RSH}{RTH}$. (3.55)

6- Grand sensible heat factor, equation 3.56 is valid

GSHF = $\frac{TSH}{GTH}$. (3.56)

We can take advantage from the previous equations to determine some parameters on the air map as, input air temp output air temp from cooling coil for pushing air units, so we can know the processed temperature to the room define the dew point for cooling load. At below is a presentation of how we can draw the procedures on a map:

RSHF can be represented by line up between the air condition design in space and air condition that is equipped to the room, this detail is seen in Figure 3.2 [39]

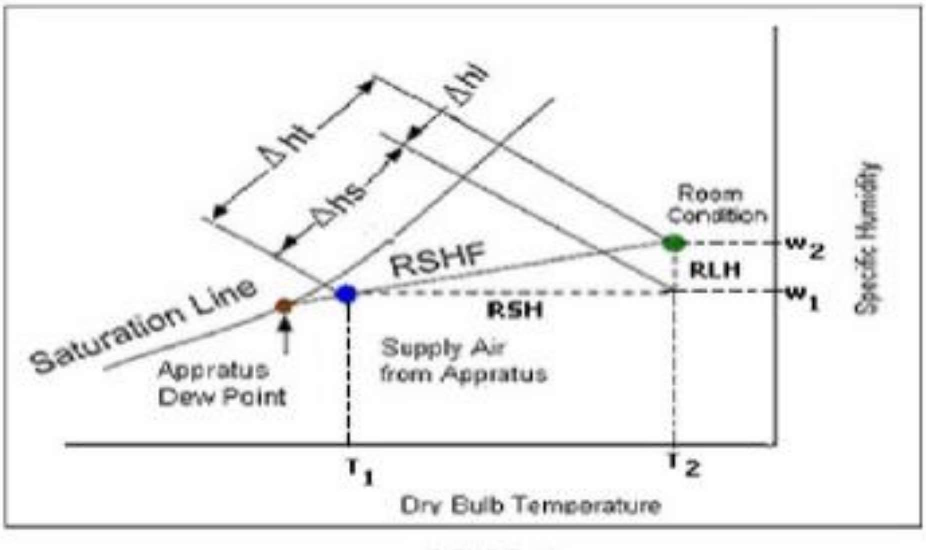

Figure 3.2: RSHF with Dry Bulb Temp-Specific Humidity [39].

The relationship between specific volume and wet bulb temperature with specific enthalpy, which is shown in Figure 3.3 [40]

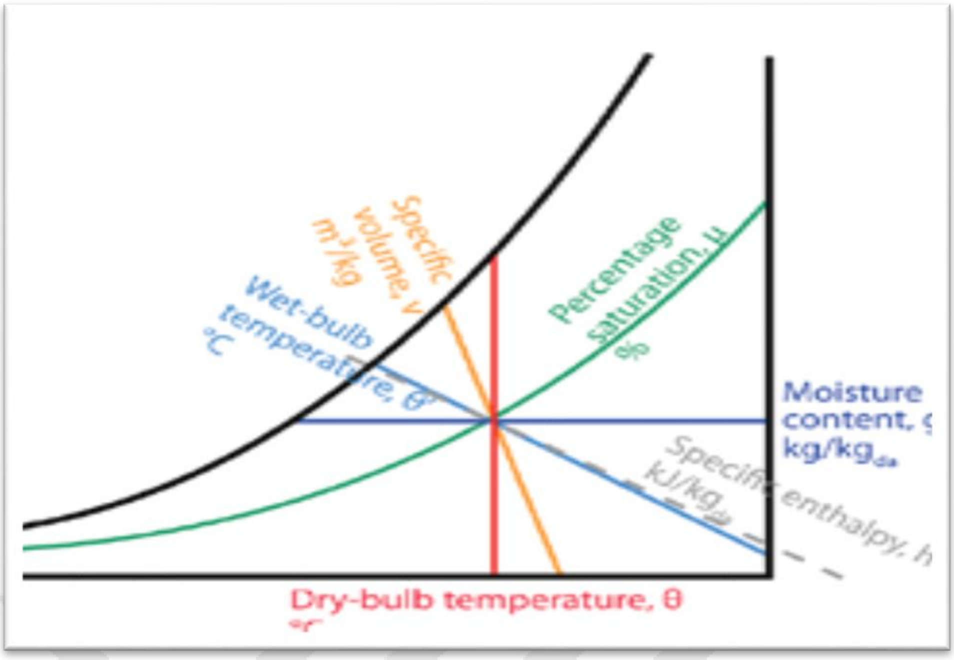

Figure 3.3: Wet Bulb – Specific Enthalpy-Specific Volume Chart [40].

B- (GSHF) can be represented by line up between the air state which enters to air equipment coil and air state which is equipped from pushing air units, so is shown in Figure 3-4 [40]

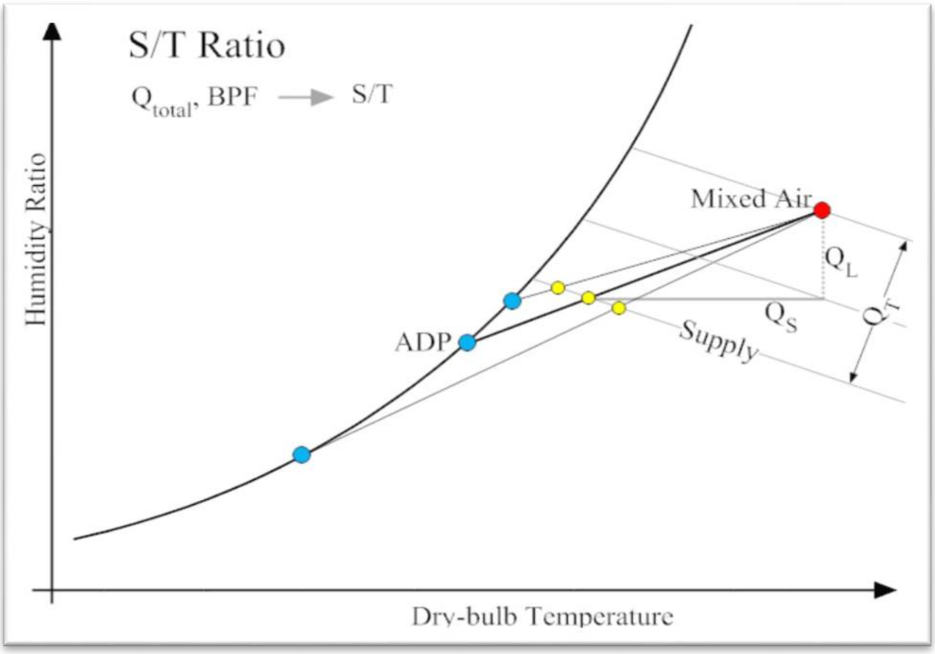

Figure 3.4: Apparatus Dew Point (ADP) with Dry Bulb-Humidity Ratio [40].

We are benefiting from above chart to determine the leaving air condition from apparatus.

C- RSHF and GSHF lines, these lines may be cross, the intersection point is represented the air condition that is equipped the room [40]. The lines may be doing not intersect, then in this case, we must draw the line from off point [air state which is equipped from pushing air units] to red point. The drawn line must intersect with RSHF line in point, saturation air, SA [Air condition that is equipped to the room]. About line “off-ADP” is represented the heat gain from additional cooling systems, Figure 3.5 shows the GSHF with Dry Bulb Temp-Specific Humidity [39].

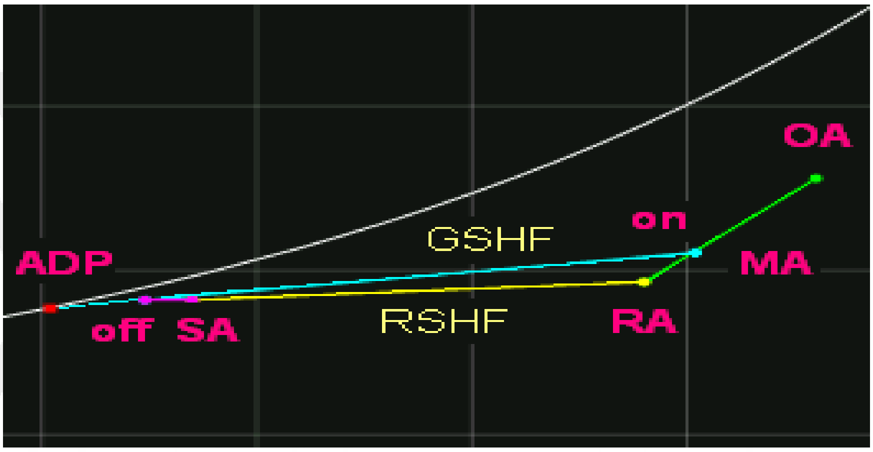

Figure 3.5: GSHF with Dry Bulb Temp-Specific Humidity [39].

The mechanism terms of system are shown in Figure 3.6 [39].

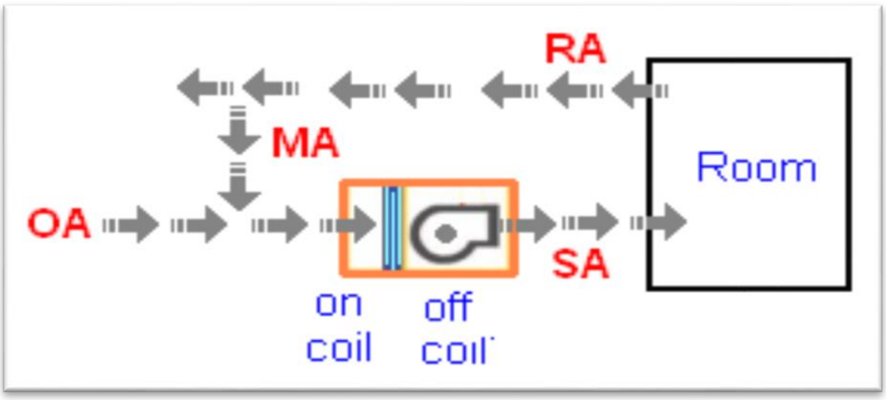

Figure 3.6: Mechanism Terms of System with SA (saturation air) [39].

D- Tadpc, it can be determined through drawing the line between ESHF and saturation line, which is seen in Figure 3.7 [40].

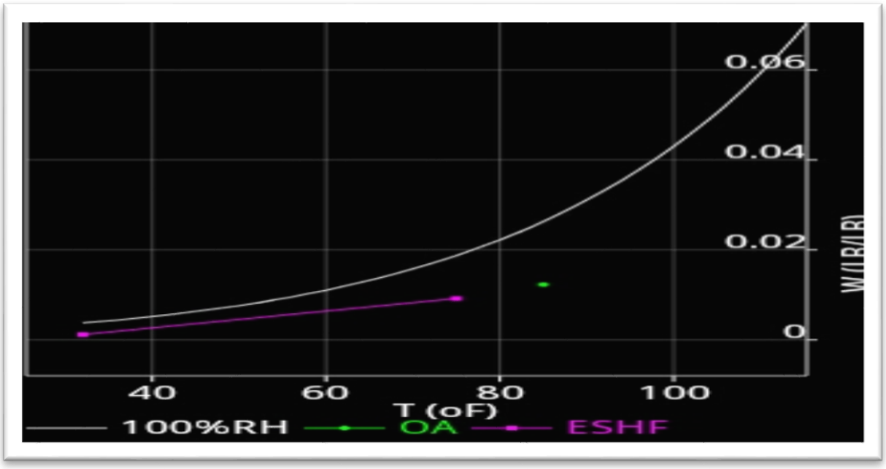

Figure 3.7: ESHF - Saturation Line Chart [40].

So we can find the Dew point for different Humidity Ratio from chart, which is shown in Figure 3.8 [40]

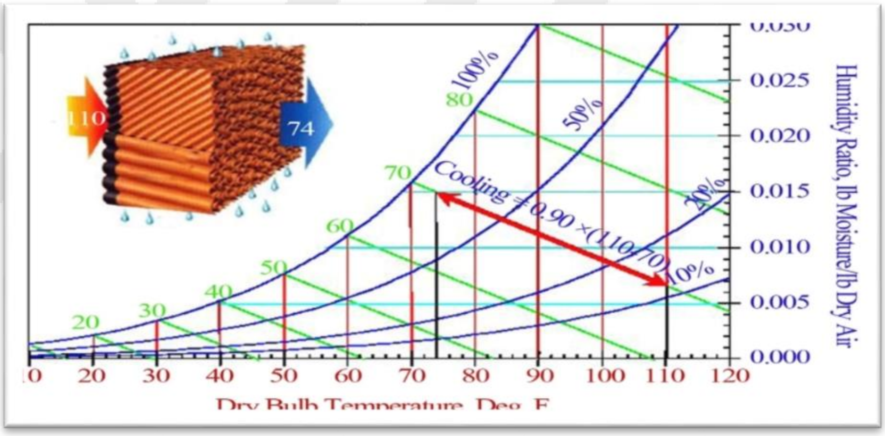

Figure 3.8: Dew Point- Humidity Ratio Chart [40].

AB- Calculating the flow rates. The air equipped volumetric flow rate

$$Vair = \frac{ERSH}{1.22(1-BF)(Troom-Tadpc)} \tag{3.57}$$

BPF is also used to express cooling coil efficiency as

BPF = (hs - hadp) / (hma - hadp). As we saw a high by pass factor indicates poor cooling performance.

We can determine the value of Tadpc from air map or from ESHF value, relative humidity interior design temperature. BF which is seen in Figure 3.9 [40]

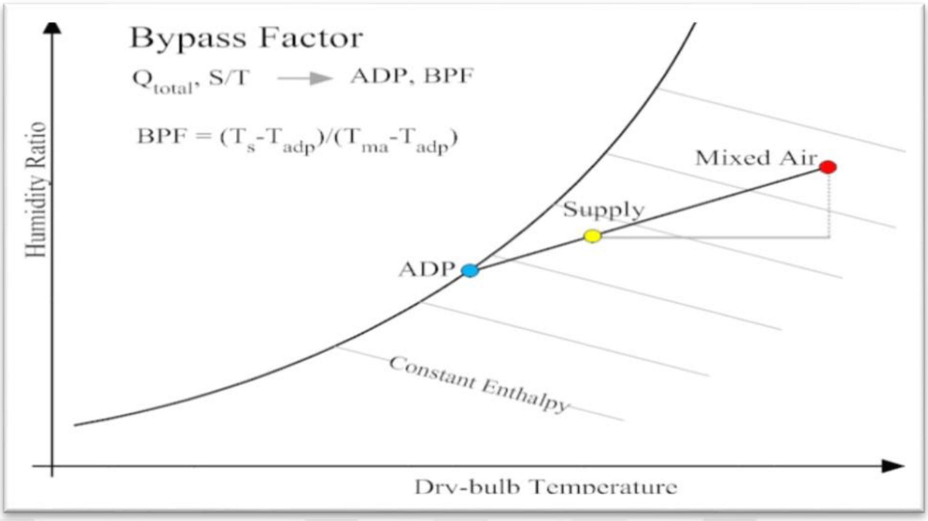

Figure 3.9: *BPF* Bypass Factor-ADP-Mixed Air [40].

2- The frosty water equipped volumetric flow rate

$$\text{Vwater} = \frac{GTH}{4190 \times \Delta Tw} \tag{3.58}$$

ΔTw: The difference between the equipped water temperature and returnable water from the air pushing units. It can be estimated at about 5.5°C [1]. The general psychometric chart is seen in Figure 3.10 [40]

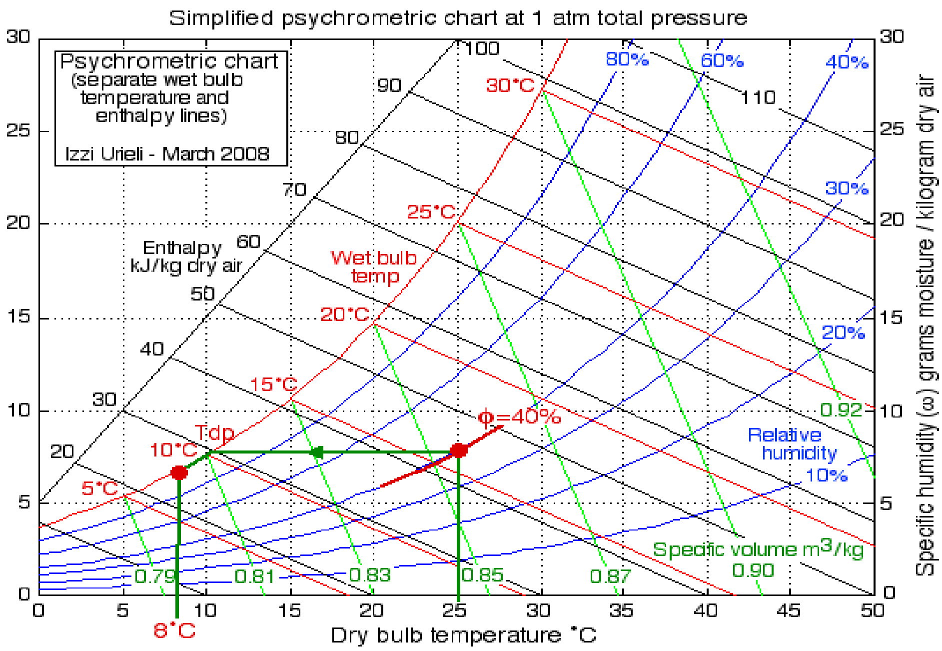

Figure 3.10: General Psychometric Chart [40].

3.4 Calculating of the air duct distribution system

There are four ways to determine the dimensions of the distribution duct:

1- Velocity Reduction Method: At this way we can chose speed at each section of the net sinkhole, speed is high, less at branches [40]. By using friction curved 3.11, we will get the size of the duct, pressure drop at each section. This method is used in modest air duct in rooms and small shops, because the problem of system balance, it requires putting a damper in air duct to provide a pressure drop. Therefore, the cost will be high, the relationship between Air Ducts and Friction Loss which is seen in Figure 3.11 [27]

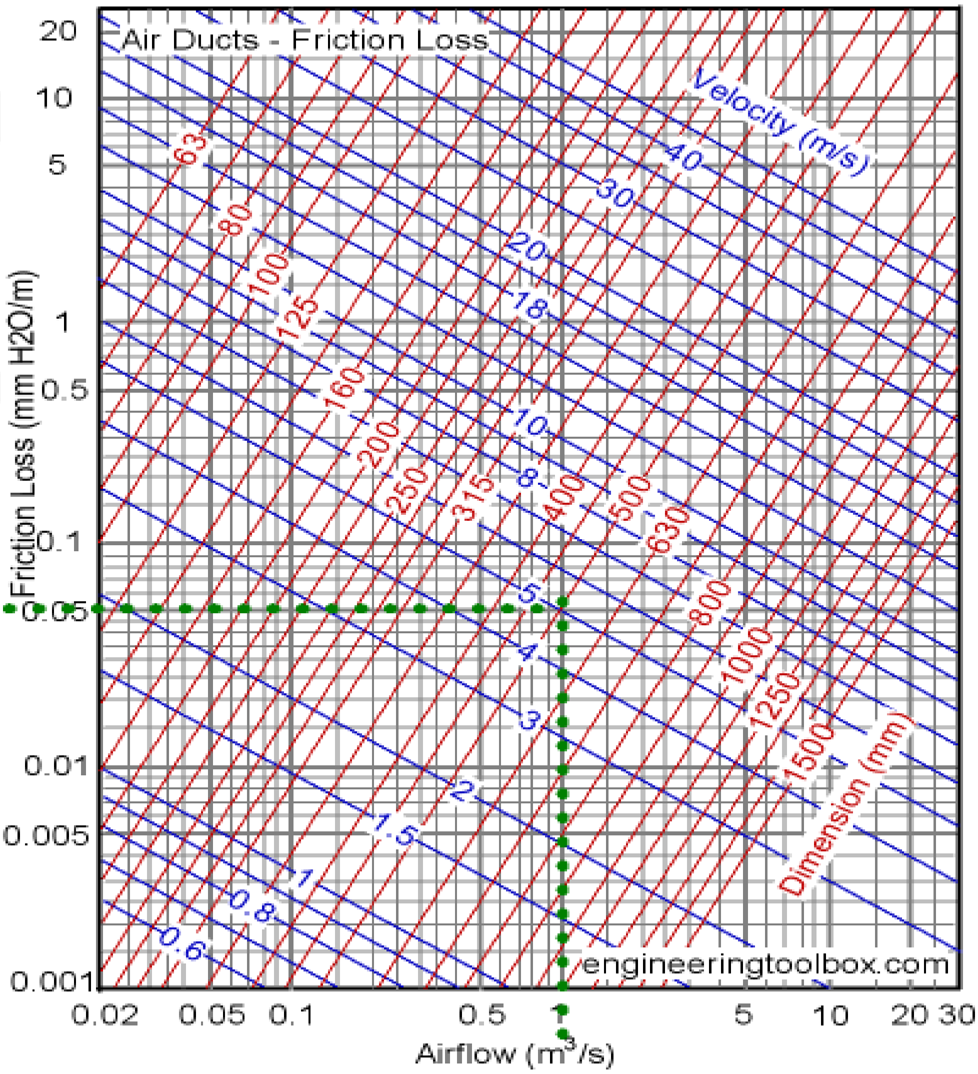

Figure 3.11: Air Ducts - Friction Loss Chart [27].

2- Equal Pressure Drop Method: This theory is based on pressure agent in which the pressure is fixed through the air duct. The total loss in pressure is calculated by multiplying the friction coefficient by air duct length, before that, the pressure drop is appointed [5]. Through the friction curved scheme 3.11, we can estimate the air speed in the sections of ducts and duct dimension [27].

3- Static Regain Method: The summary of this method is taking the advantage of the static pressure, increasing at every branch where the air speed is decreased, which means that the pressure turns up from speed pressure to static pressure. This called Re -Serenity. At this mode we must calculate the pressure drop from air volume scheme based on speed. This way is used in high speed HVAC. The design feature has a big air duct design, therefore, high cost but few balance problems [27].

4- Equal Balanced Pressure Drop Method: This method is used for the purpose of balance the non-symmetrical branches [5]. This is done by calculating the total pressure for branch duct, from the beginning system even the end of the distribution slot for branch Pmain. It is based on the new pressure value where re design the sub branch size, to make the pressure drop in the main stream, equations 3.59, 3.60 are valid [27]

$$\text{Pressure drop} = \text{Pmain} - \text{Pbranch} \tag{3.59}$$

$$\Delta\text{Pnew} = \frac{Pressure\ drop}{Length\ of\ branch} \tag{3.60}$$

On the basis of the new pressure drop value (ΔPnew), the redesign of the sub duct is done. So we concluded that the best ways to design was equal balance pressure drop method. The air duct dimensions are determined by the air duct volume scheme Fig 3.11 We can extract the air speed in the section and the duct diameter, which are given in equation 3.61[27]

$$\text{Air Flow} = \text{Area} \times \text{Velocity} \qquad = \frac{\pi}{4}\ (Dim)^2 \times \text{Ve.}$$

$$\text{Dim} = \sqrt{\frac{\pi}{4} \times \frac{Air\ Flow}{Ve}} \tag{3.61}$$

It can convert circle diameter to the rectangle equivalent by using diameter equivalent scheme, after determination the duct height, crossing with diameter to determine the width.

B- Calculating Friction loss in air ducts

1- Friction loss for Straight Duct. It is calculated from multiplying the duct length by pressure drop design as in equation 3.62 [2]

Flosses = Length of duct × ΔP design (3.62)

2- Bend, elbow Friction loss: It is calculated from multiplying the elbow equivalent length by pressure drop design. equation 3.63 is obtained [5]

P elbow or bend = Equivalent Length × ΔP design (3.63)

We can determine the equivalent length. equations 3.64,3.65 are valid [17]

$$\frac{Leq}{W} = [0.33\times (\frac{R}{W})]^{n} \quad (3.64)$$

$$n = -2.13\ (\frac{H}{W})^{0.126} \quad (3.65)$$

R/W is represented the ratio between the elbow radius and its width, usually the practical value equal 1.5 so, H/W is named Aspect Ratio.

3- Diffuser Friction loss: There are many dozens of diffuser kinds, they are depending on the quality of building and architectural requirements. Usually the manufactures informed us about these values [6].

4- Divergent, convergent, other structures, friction loss: At these items, we must know the pressure drop from friction “Dynamic loss”. The pressure drop is depending on the section area and design, so the manufactures are told us about these values of P fitting [5], [6].

5- Air Pushing Accessories, friction loss: The pressure drop is concentrated at air filters, cooling coil, heating coil, fan inlet and others, which is shown in equation 3.66 [6].

P(Air-Heading) = ρ air × g × H (3.66)

Where

g : gravity m/sec^2

H: height of item (m).

ρ: air density (kg/m^3).

C- Total pressure drop for air ducts. It is the summation of the static and dynamic pressures; Equations 3.67,3.68 are valid [5]

$$Pt = Ps + Pv. \tag{3.67}$$

Where:

$$Ps = Pflosses+Pbend+Pdiffuser+Pfitting+PAir\text{-}Heading. \tag{3.68}$$

As for Pv it is the total dynamic pressure drop, as in Equation 3.69 [5]

$$Pv = \frac{1}{2}\rho air\times (Vs^2 - Vr^2). \tag{3.69}$$

Where Vr, Vs are the air speeds at fan, outlet and inlet respectively.

3.5 Calculation of equipment energy consumption

1- Calculation of energy consumption for air pushing fans

A) Air Preparation Fan: The consumption depends on the amount of required flow per out fitted unit, as well depends on the total pressure drop about unit, which is given in equation 3.70 [41]

$$W_{fans} = \frac{Vair\ supply \times \Delta PTS}{\eta_m \times \eta_f} \tag{3.70}$$

Where*:*

η_m = motor efficiency.

η_f = mechanical fan efficiency.

Where the minimum value of (η_m, η_f) is 60% [41]

ΔPTS : pressure difference for fan.

Table 3.9 below shows the motor efficiency for difference power amplitude, [34].

Table 3.9: Motor Efficiency [34].

Motor Power	Efficiency %
1 → 200 watt	0.60
375→ 750 watt	0.70
1 → 4 kw	0.80
5.5 → 15 kw	0.85
> 15kw	0.88

B) Air vacuum fan: Which is shown in equation 3.71[41]

$$W_{\text{fanex}} = \frac{(percentag\ of\ exhaut\ air)\times Vreturn\ air\ \times \Delta \text{PTr}}{\eta_m \times \eta_{fanex}} \qquad (3.71)$$

Where the minimum value of (η_m, η_{fanex}) is 60% .

2- Calculation of energy consumption of compressor: The compressor is considered of the most important parts of HVAC systems, most consumption for its mechanism. We can estimate the consumption rates from manufactures performance maps for various operating conditions. The compressor consumption power is described as algebraic function for condensation temperature (Tc) and vaporization temperature (Te), vie second degree equation, as in equation 3.72 [42]

$$W_{incomp} = b_1 + b_2 t_e + b_3 t_e^2 + b_4 t_c + b_5 t_c^2 + b_6 t_e t_c + b_7 t_e^2 t_c + b_8 t_e t_c^2 + b_9 t_e^2 t_c^2 \qquad (3.72)$$

As for cooling capacity, equation 3.73 is obtained from this series

$$q_e = a_1 + a_2 t_e + a_3 t_e^2 + a_4 t_c + a_5 t_c^2 + a_6 t_e t_c + a_7 t_e^2 t_c + a_8 t_e t_c^2 + a_9 t_e^2 t_c^2 \qquad (3.73)$$

Where a_n , b_n are constants, it can be found by using the methods of mathematical analysis dependence on performance schemes [42].

3- Calculation of energy consumption of water pumps

A) Cooling Water Pump: The cooling water pump consumption is depending on the pressure drop in closed system and amount of water flow, equation 3.74 is valid

$$W_{pump\ evop} = \frac{Vwater\ evop \times \Delta \mathrm{PTevop}}{\eta_m \times \eta_{pump\ evop}} \tag{3.74}$$

Where (η_m , $\eta_{pump\ evop}$) are the total efficiency for water pump.

Where the minimum value which allowed to work is 60% [41].

B) Condensation Water Pump: The condensation water pump is depending on the pressure drop in open system and the amount of water flow. The flow rate is determined by the amount of removed heat in the condenser, equations 3.75, 3.76 are obtained [41]

$$q_c = q_e + W_{in\ comp} \tag{3.75}$$

$$V_{water\ cond} = \frac{q_c}{4190 * \Delta \mathrm{Tcond}} \tag{3.76}$$

Where:

ΔTcond : Difference temp between condenser inlet water coming from the cooling tower and condenser outlet.

The consumption power of condensation water pump, which is shown in equation 3.77[41]

$$W_{pump\ cond} = \frac{Vwater\ cond \times \Delta \mathrm{PTcond}}{\eta_m \times \eta_{pump\ cond}} \tag{3.77}$$

4- Calculation of energy consumption of fan coil unit: It is determined from manufactures data, after knowing the air and water flow rates, which are passing in these units, as in equation 3.78 [41]

$$W_{Tfan,coil} = \sum_{1}^{n} W_{fan,coil} \tag{3.78}$$

Where: n, the number of rooms which is used in the building.

5- Calculation of energy consumption at partial load: The operation of refrigeration unit at partial loads, it means the cooling load, that to be removed, it will be less than the design capacity. Therefore, the compressor consumption power is calculated from partial load curves and, which is given in equation 3.79[43]

$$\text{Capacity ratio} = \frac{Partial\ Load\ Value}{\text{Design Capacity}} \tag{3.79}$$

CHAPTER 4

EXPERIMENTS SIMULATIONS & RESULTS

4.1 INTRODUCTION

In this research, two prison halls were investigated and simulated using three types of HVAC systems the results were compared for these six cases to find the most energy efficient and energy saving system, which is very important on, sever summer conditions on Iraq.

The first hall represents the old prison building style where separate small buildings founded on the main area each represent different use all surrounded by walls guard’s facilities etc. The second type of building is more advanced two story large halls connected to each other. These two cell- hall are too much different from each other by size and structure so the difference can be noticed and the results will be more reliable and widely applicable. On the other hand, two common and widely used different HVAC systems Split Unit & Package Unit were suggested to investigate and study to show the best practice that can lead to the best energy saving and accepted comfort conditions.

A third type of HVAC system was suggested, that is the Indirect Direct Evaporative Cooler (IDEC) using Heat Mass Exchanger Media (HMX) the mechanism is seen in figure 4.1. This type is the most advanced evaporative cooler that can reach 120% of the wet bulb temperature, which is very efficient on the middle and north region of Iraq where high dry climate is the dominant on summer. On the same time, it gives 100% fresh air that is perfect for many situations and a mandatory for others. The dis advantage of this system is it depend on water and water TDS may affect its efficiency and its very expensive comparing with other type of HVAC system other than fresh air systems.

Not only the above three systems were applied for the two cell halls to compare the systems on both conditions, but also another application was used that is the

Variable Air Volume (VAV) which may also reduce the energy used for supplying the air to the system. Where (TDS: Total dissolved solids).

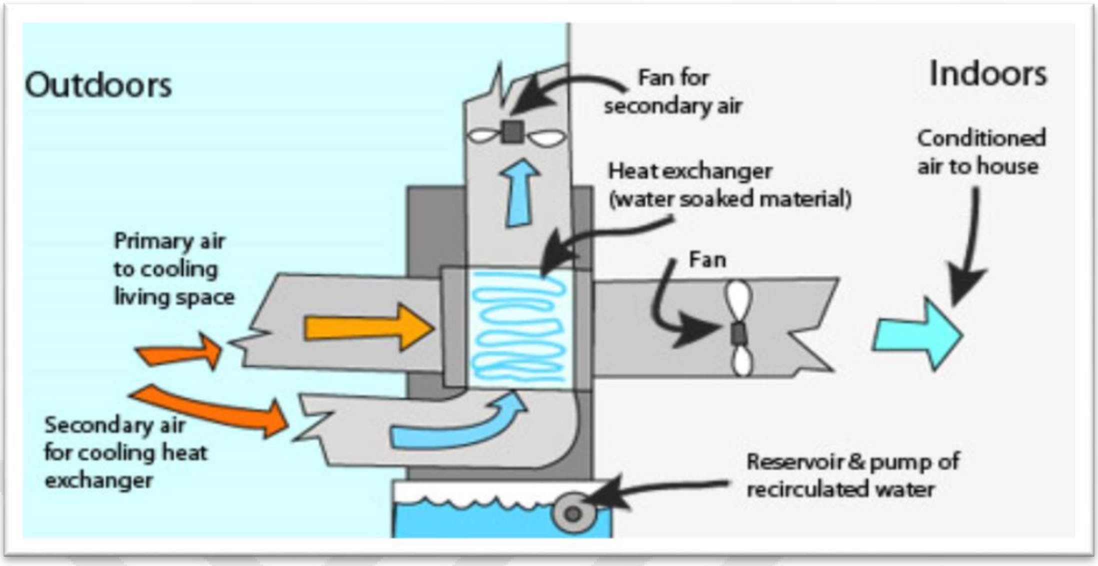

Figure 4.1: Two Stage or Indirect/Direct Evaporative Cooling.

4.2 ASSUMPTIONS

Fresh air supply was not used since it will increase the energy consumption dramatically for the first two systems while energy will stay the same for the DIEC this will make the results non-comparative.

All systems were assumed to supply the air using ducting system split units were assumed to use less ducts than other systems.

Baghdad climatic conditions used for the purpose of this simulation. For the small building and because of high occupation no heating is needed in winter.

A complete list of all assumptions and conditions used is listed below, so the prisons models layouts are shown in figures 4.2,4.3,4.4.

These are some of input &output data about HAP program:

4.3 Abu Ghraib- First Floor- DATA SHEET

09/06/2017 1:26 PM	**Space Input Data**	**Fadhil Asaad Al Malaki**

Abu Ghraib First Floor

1. General Details:

Floor Area .... **1050.0** m²
Avg. Ceiling Height .... **3.0** m
Building Weight .... **341.8** kg/m²

1.1. OA Ventilation Requirements:

Space Usage **CORRECTIONAL FACILITY: Cells**
OA Requirement 1 .... **10.0** L/s/person
OA Requirement 2 .... **0.00** L/(s-m²)
Space Usage Defaults **ASHRAE Standard 62-2001**
....

2. Internals:

2.1. Overhead Lighting:

Fixture Type **Recessed (Unvented)**....................
Wattage .... **10.00** W/m²
Ballast Multiplier **1.00**
Schedule ..... **24 hour**

2.2. Task Lighting:

Wattage .... **0.00** W/m²
Schedule .... **None**

2.3. Electrical Equipment:

Wattage .... **0.00** W/m²
Schedule .... **None**

2.4. People:

Occupancy .... **60.0** People
Activity Level **Sedentary Work**
Sensible .... **82.1** .... W/person
Latent .... **79.1** .... W/person
Schedule **off 1 hour Thurs.**

2.5. Miscellaneous Loads:

Sensible .... **5000** W
Schedule **off 1 hour Thurs.**
Latent .... **1000** W
Schedule **off 1 hour Thurs.**

3. Walls, Windows, Doors:

Exp.	Wall Gross Area (m²)	Window 1 Qty.	Window 2 Qty.	Door 1 Qty.
E	45.0	0	0	0
S	210.0	0	0	1
W	45.0	0	0	0
N	210.0	0	0	1

3.1. Construction Types for Exposure E

Wall Type **Stucco + 203mm common brick + 13mm gypsum plaster**

3.2. Construction Types for Exposure S

Wall Type **Stucco + 203mm common brick + 13mm gypsum plaster**

Door Type **door1**

3.3. Construction Types for Exposure W

Wall Type **Stucco + 203mm common brick + 13mm gypsum plaster**

3.4. Construction Types for Exposure N

Wall Type **Stucco + 203mm common brick + 13mm gypsum plaster**

Door Type **door1**

4. Roofs, Skylights:

(No Roof or Skylight data).

5. Infiltration:

Design Cooling .. **2.00** ACH

Design Heating . **2.00** ACH

Energy Analysis **2.00** ACH

Infiltration occurs only when the fan is off.

6. Floors:

Type **Slab Floor On Grade**

Floor Area ..... **1050.0** m^2

Total Floor U-Value **0.568** W/(m^2-°K)

Exposed Perimeter **170.0** m

Edge Insulation R-Value **0.00** (m^2-°K)/W

7. Partitions:

(No partition data).

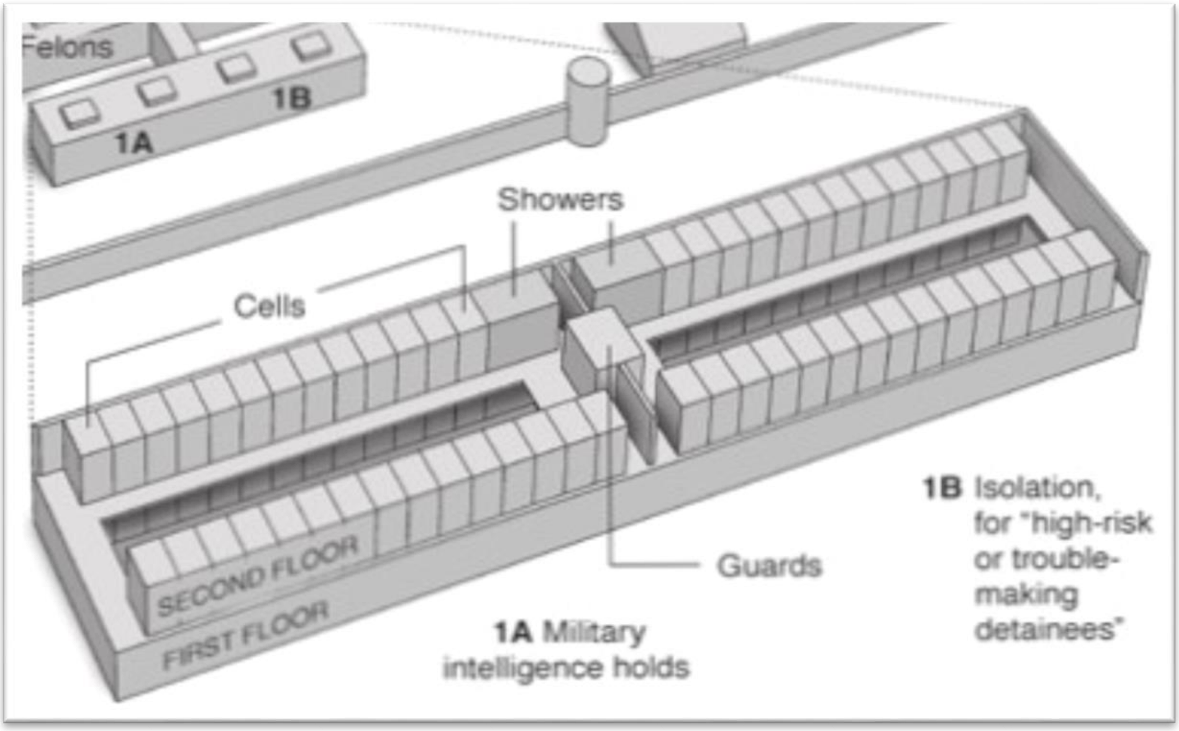

Figure 4.2: 3D Layout of Abu Ghraib, the Simulated Cells Halls.

4.4 - Abu Ghraib -Second Floor- DATA SHEET

09/06/2017 3:29 PM	Space Input Data	Fadhil Asaad Al Malaki

Abu Ghraib Second Floor

1. General Details:

Floor Area .... 1050.0 m²
Avg. Ceiling Height 3.0 m
Building Weight . 341.8 kg/m²

1.1. OA Ventilation Requirements:

Space Usage **CORRECTIONAL FACILITY: Cells**
OA Requirement 1 **10.0** L/s/person
OA Requirement 2 **0.00** L/(s-m²)
Space Usage Defaults **ASHRAE Standard 62-2001**

2. Internals:

2.1. Overhead Lighting:

Fixture Type **Recessed (Unvented)**......................
Wattage .... **10.00** W/m²
Ballast Multiplier .. **1.00**
Schedule .... **24 hour**

2.4. People:

Occupancy .... **60.0** People
Activity Level **Sedentary Work**
Sensible .... **82.1** .... W/person
Latent .... **79.1** .... W/person
Schedule **off 1 hour Thurs.**

2.2. Task Lighting:

Wattage .... **0.00** W/m²
Schedule .... **None**

2.5. Miscellaneous Loads:

Sensible .... **5000** W
Schedule **off 1 hour Thurs.**
Latent .... **1000** W
Schedule **off 1 hour Thurs.**

2.3. Electrical Equipment:

Wattage .... **0.00** W/m²
Schedule .... **None**

3. Walls, Windows, Doors:

Exp.	Wall Gross Area (m²)	Window 1 Qty.	Window 2 Qty.	Door 1 Qty.
E	45.0	0	0	0
S	210.0	0	0	0
W	45.0	0	0	0
N	210.0	0	0	0

3.1. Construction Types for Exposure E

Wall Type **Stucco + 203mm common brick + 13mm gypsum plaster**

3.2. Construction Types for Exposure S

Wall Type **Stucco + 203mm common brick + 13mm gypsum plaster**

3.3. Construction Types for Exposure W

Wall Type **Stucco + 203mm common brick + 13mm gypsum plaster**

3.4. Construction Types for Exposure N

Wall Type **Stucco + 203mm common brick + 13mm gypsum plaster**

4. Roofs, Skylights:

Exp.	Roof Gross Area (m^2)	Roof Slope (deg.)	Skylight Qty.
H	1050.0	0	0

4.1. Construction Types for Exposure H

Roof Type **Default Roof Assembly**

5. Infiltration:

Design Cooling .. **2.00** ACH
Design Heating . **2.00** ACH
Energy Analysis **2.00** ACH
Infiltration occurs only when the fan is off.

6. Floors:

Type **Floor Above Conditioned Space**
(No additional input required for this floor type).

7. Partitions:

(No partition data).

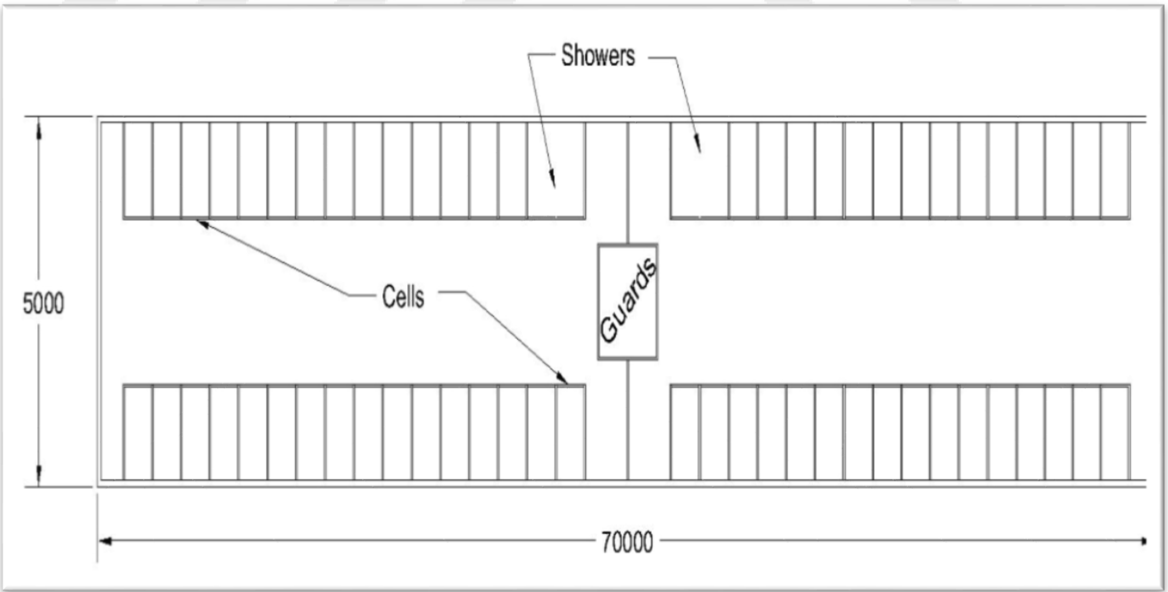

Figure 4.3: Layout of Abu Ghraib, the Simulated Cells Halls.

4.5 Rusafa-DATA SHEET

09/06/2017 3:30 pm	**Space Input Data**	Fadhil Asaad Al Malaki

Rusafa

---

1. General Details:

Floor Area .... **96.0** m²
Avg. Ceiling Height . **3.0** m
Building Weight .. **341.8** kg/m²

1.1. OA Ventilation Requirements:

Space Usage **CORRECTIONAL FACILITY: Cells**
OA Requirement 1 **10.0** L/s/person
OA Requirement 2 **0.00** L/(s-m²)
Space Usage Defaults **ASHRAE Standard 62-2001**

2. Internals:

2.1. Overhead Lighting:

Fixture Type **Recessed (Unvented)** ....
Wattage .... **10.00** W/m²
Ballast Multiplier ... **1.00**
Schedule .... **24 hour**

2.2. Task Lighting:

Wattage .... **0.00** W/m²
Schedule .... **None**

2.3. Electrical Equipment:

Wattage .... **0.00** W/m²
Schedule .... **None**
....

2.4. People:

Occupancy .... **36.0** People
Activity Level **Sedentary Work**
Sensible .... **82.1** .... W/person
Latent .... **79.1** .... W/person
Schedule **off 1 hour Thurs.**

2.5. Miscellaneous Loads:

Sensible .... **3000** W
Schedule **off 1 hour Thurs.**
Latent .... **1000** W
Schedule **off 1 hour Thurs.**

3. Walls, Windows, Doors:

Exp.	Wall Gross Area (m²)	Window 1 Qty.	Window 2 Qty.	Door 1 Qty.
E	36.0	3	0	0
S	24.0	0	0	0
W	24.0	0	0	0
W	12.0	0	0	0
N	24.0	0	0	1

3.1. Construction Types for Exposure E

Wall Type **Stucco + 203mm common brick + 13mm gypsum plaster**

1st Window Type **Sample Window Assembly**

3.2. Construction Types for Exposure S

Wall Type **Stucco + 203mm common brick + 13mm gypsum plaster**

3.3. Construction Types for Exposure W

Wall Type **Stucco + 203mm common brick + 13mm gypsum plaster**

3.4. Construction Types for Exposure W

Wall Type **Stucco + 203mm common brick + 13mm gypsum plaster**

3.5. Construction Types for Exposure N

Wall Type **Stucco + 203mm common brick + 13mm gypsum plaster**

Door Type **door1**

4. Roofs, Skylights:

Exp.	Roof Gross Area (m^2)	Roof Slope (deg.)	Skylight Qty.
H	96.0	0	0

4.1. Construction Types for Exposure H

Roof Type **Default Roof Assembly**

5. Infiltration:

Design Cooling .. **2.00** ACH

Design Heating . **2.00** ACH

Infiltration occurs only when the fan is off.

6. Floors:

Type Slab Floor On Grade

Floor Area .... 96.0 m^2

Total Floor U-Value 0.568 W/(m^2-°K)

Exposed Perimeter 60.0 m

Edge Insulation R-Value 0.00 (m^2-°K)/W

7. Partitions **(No partition data)**

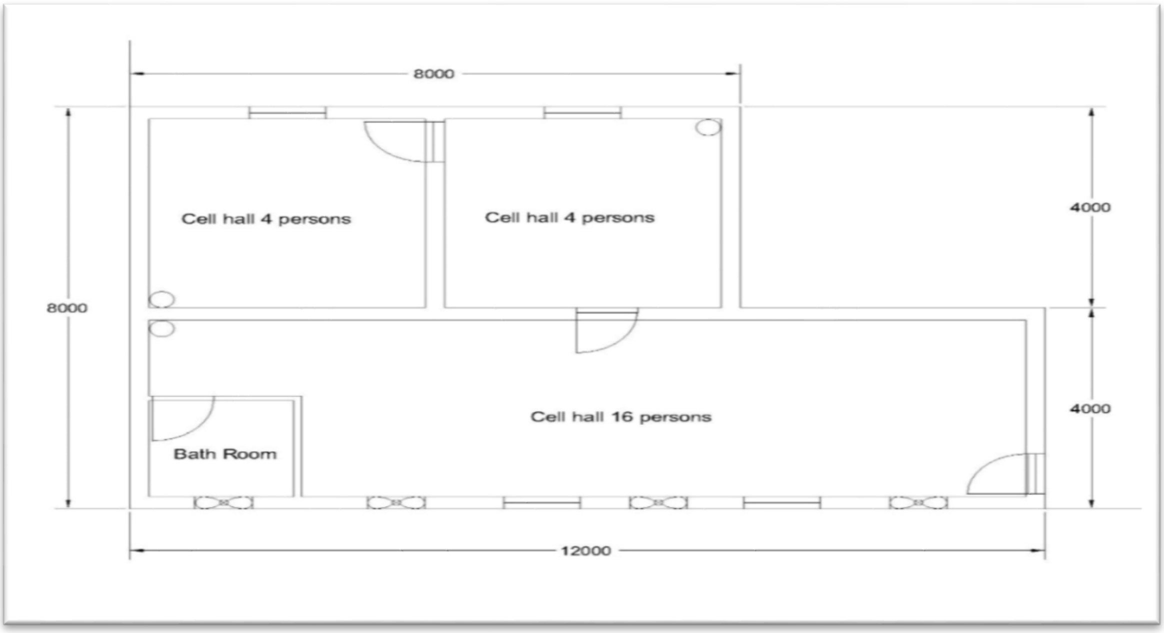

Figure 4.4: Layout of Rusafa ,the Simulated Cells Halls.

4.6 THE SOFT WARE

To simulate the mentioned building for the above-mentioned systems, a widely used simulation software was used which is the **Hourly Analysis Program** (**HAP**) introduced by Carrier Corporation **Version 4.90-2014**, as illustrated in Figure 4.5.

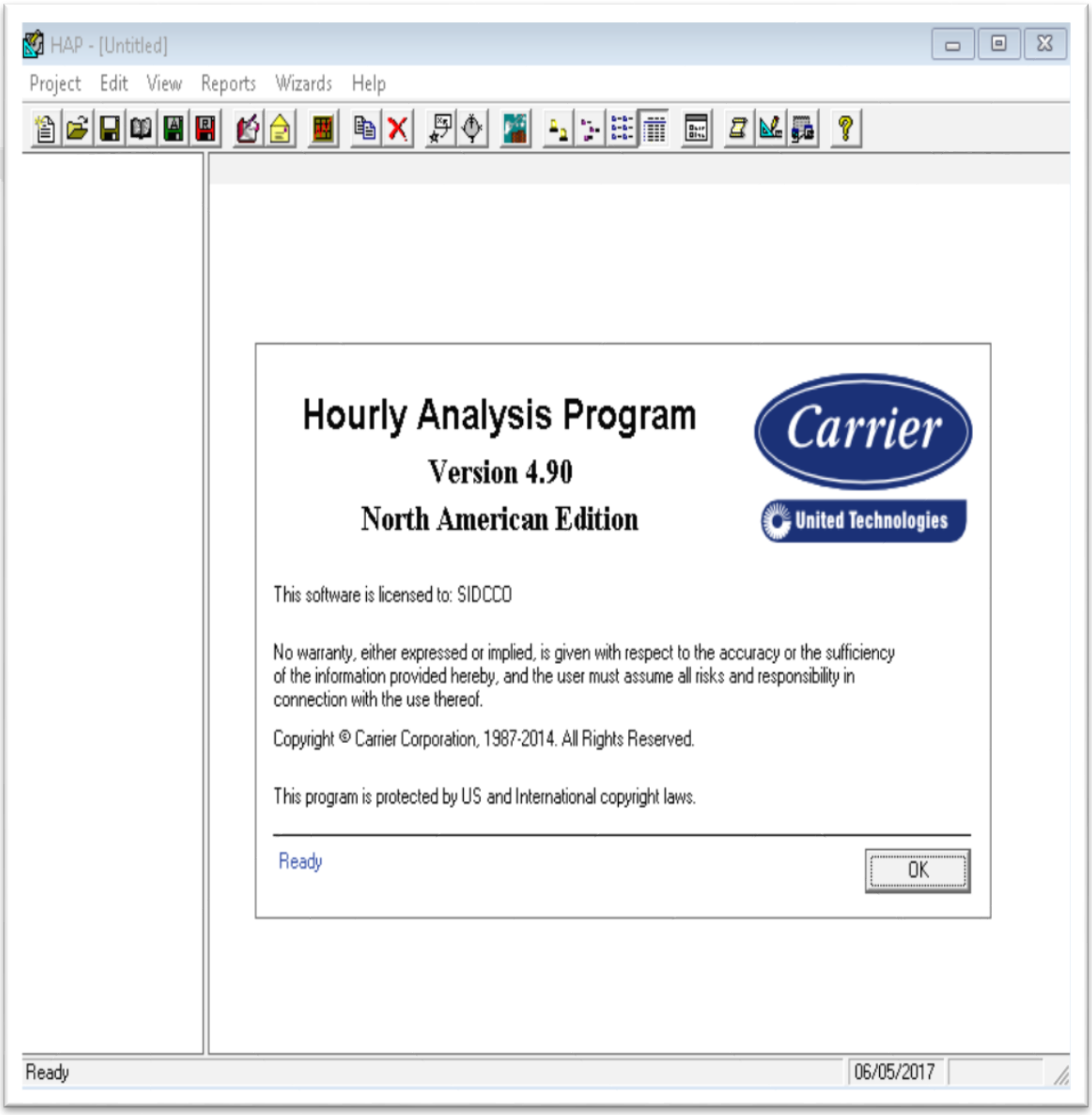

Figure 4.5: HAP Software Introduction Page.

This software was selected because it is simple, reliable, well known, and perform the detailed required load calculations as well as energy and power simulation all in one software. It also gives detailed hour-by-hour calculation and power/energy simulation for any selected period. As out lined in Figure 4.6. This flexibility and wide range of analysis together with the simplicity of this software made it the perfect choice for our case. Preview of the main widows that can be used to perform the simulation is listed below is shown in Figure 4.7.

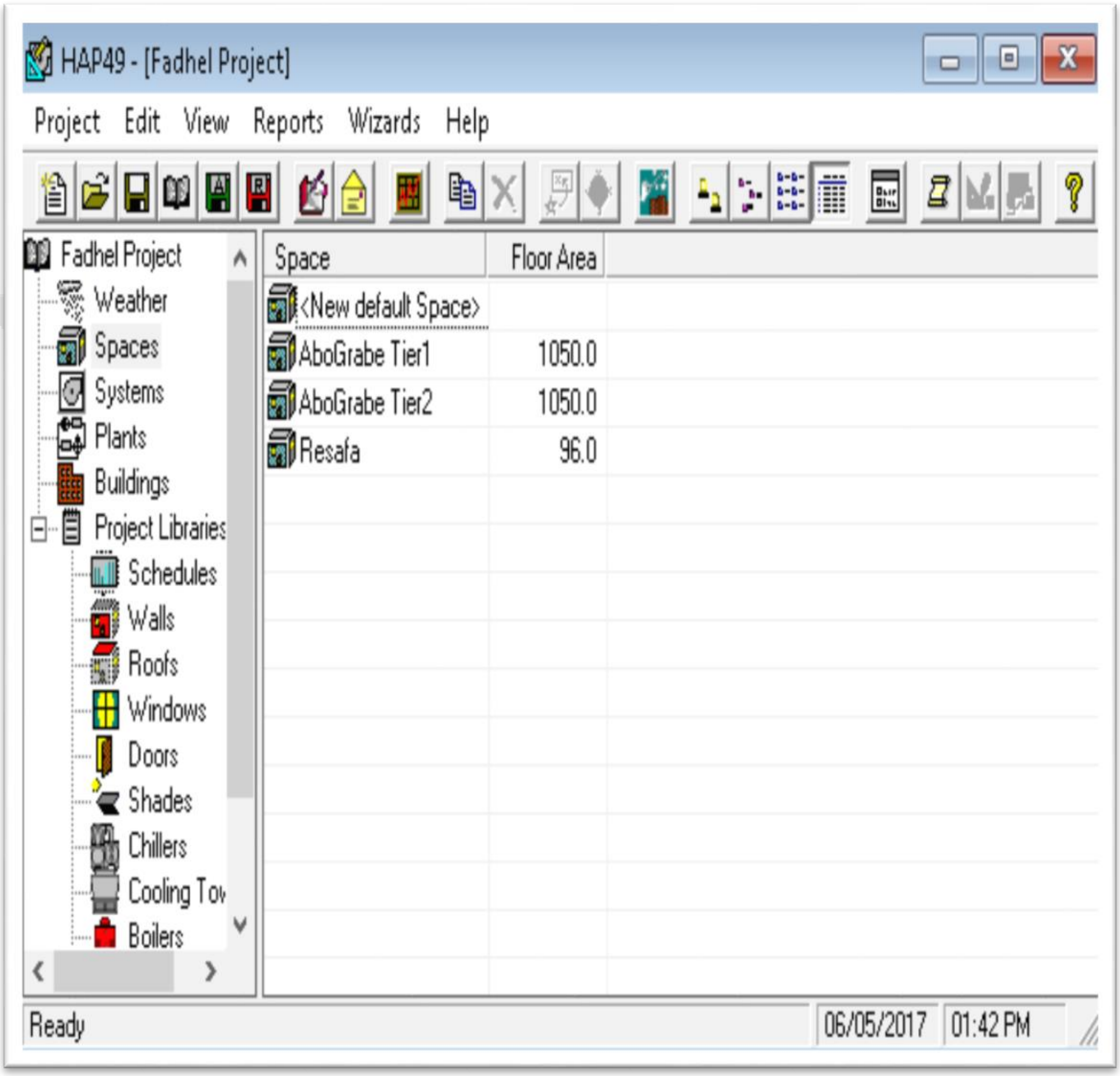

Figure 1.6: Simulated Space Definition.

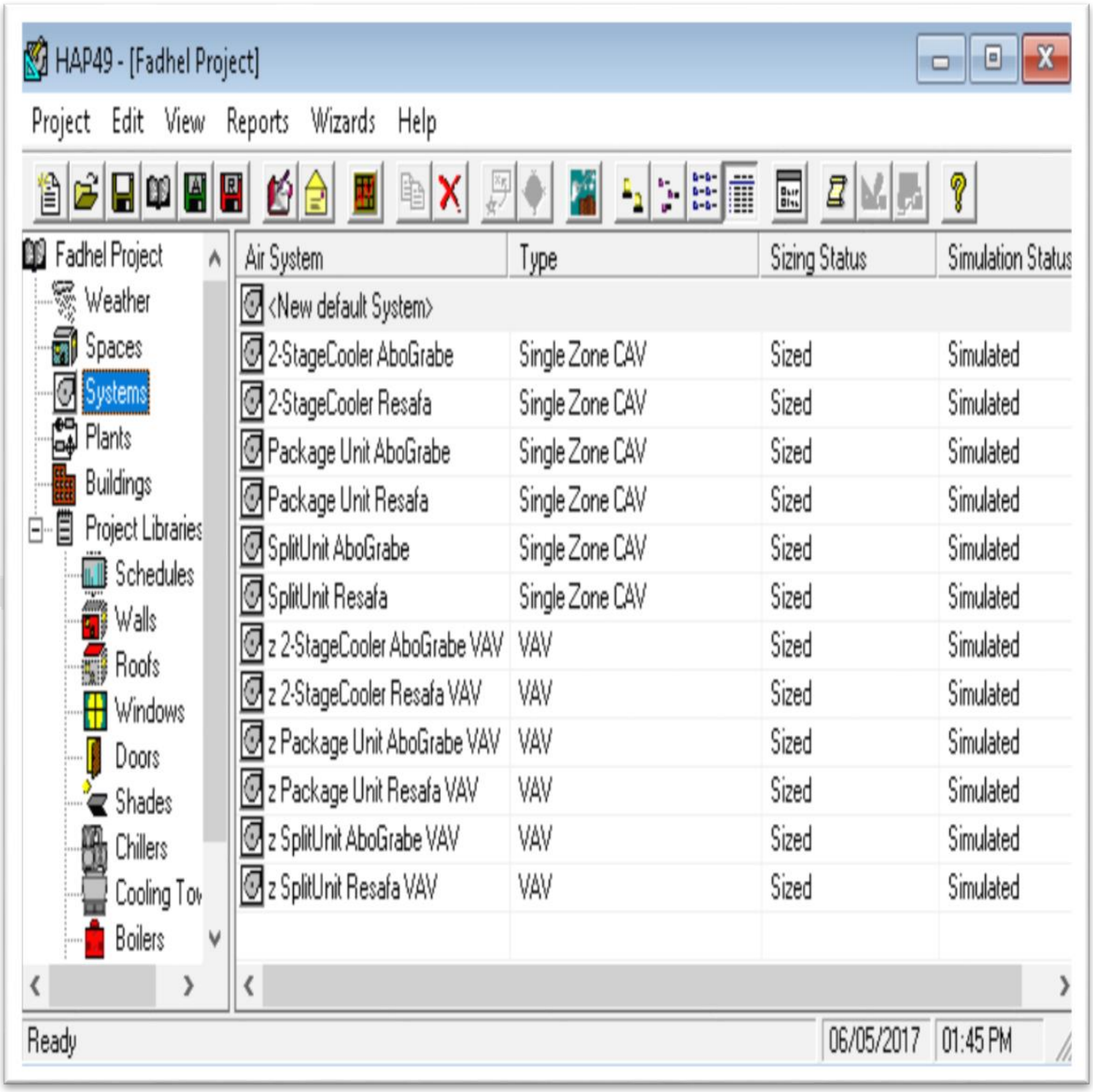

Figure 4.7: Simulated Systems for the Spaces.

4.7 CASE STUDIES

Control Principles for Energy Conservation of VAV

VAV systems are typically designed to supply constant-temperature air at all times. To conserve central plant energy, the temperature of supply air can be raised in response to demand from the zone with the greatest load (load analyzer control). However, because more cool air should be supplied to identify a given load, the mechanical cooling energy is kept, may be offset by an increase in fan energy.

Equipment operating efficiency should be studied closely before implementing temperature reset in cooling-only VAV systems, as seen in Figure 4.8.

Outside, return, and exhaust air ventilation dampers are controlled by the flow air temperature controller to provide free cooling as the first phase in the cooling sequence. When outside air temperature rises to the point that it can no longer be used for cooling, an outside air limit “economizer” control overrides the discharge controller and moves ventilation dampers to the minimum ventilation position. An enthalpy control system can substitute outside air limit control in some weathers, is shown in Figure 4.9,4.10.

After the general wants of a building have been found and the building and system subdivision has been made, the mechanical system and its control process can be considered. Designing systems that keep energy requires information of (1) the building, (2) its operative table, (3) the systems to be installed, and (4) ASHRAE Standard 90.1. The essentials or basics that conserve energy are as follows:

- Run equipment only when needed. Schedule HVAC unit operation for occupied periods. Turn on heat at night only to maintain indoor temperature between 13°C and 16°C to deny freezing. Start morning warm-up as late as possible to achieve design internal temperature by occupancy time, considering residual space temperature, outside temperature, and equipment capacity (optimum start control).
- Calculate shutdown time so that space temperature does not drift out of the selected comfort zone before occupancy ends.
- Under most extreme conditions, the equipment can be closed down some time before the finish of occupation, comprising on interior and outer loads and space temperature.
- Sequence heating and cooling do not supply heating and cooling simultaneously unless it is requested for humidity control. Central fan systems should use cool outside air in sequence between heating and cooling. Zoning and system selection should eliminate, or at least minimize, simultaneous heating and cooling. Also, humidification and dehumidification should not take place concurrently.

- Provide only the heating or cooling actually needed, reset the supply temperature of hot and cold air.
- Supply heating and cooling from the most efficient source, by using free or low-cost energy sources first, then higher-cost sources as necessary.
- Apply outside air control. When on minimum outside air, use no less than that recommended by ASHRAE Standard 62.1. In areas where it is cost-effective, use enthalpy rather than dry-bulb temperature to determine whether outside or return air is the most energy-efficient air source for the cooling mode.

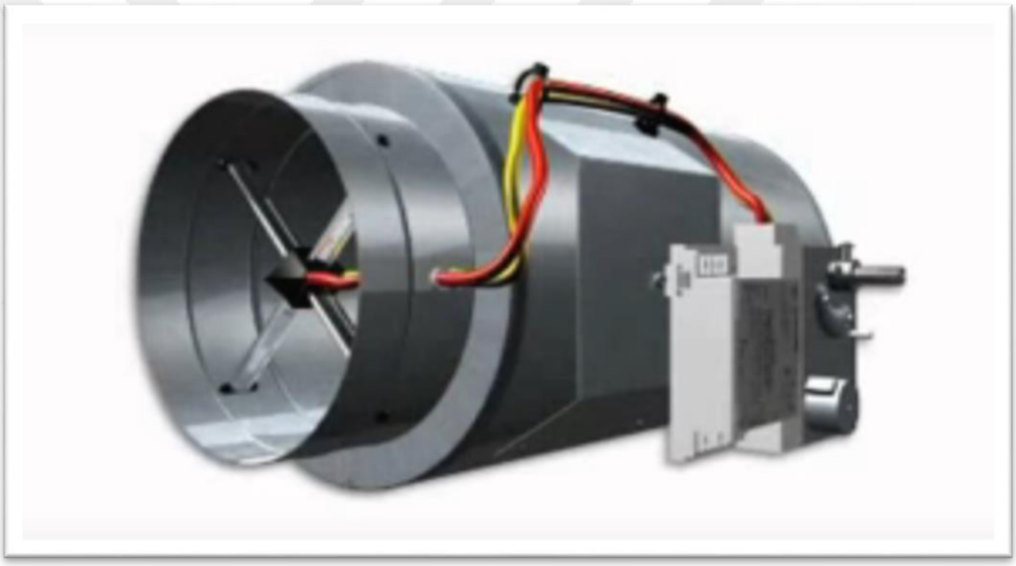

Figure 4.8: VAV Mechanism.

Figure 4.9: VAV Section with Damper.

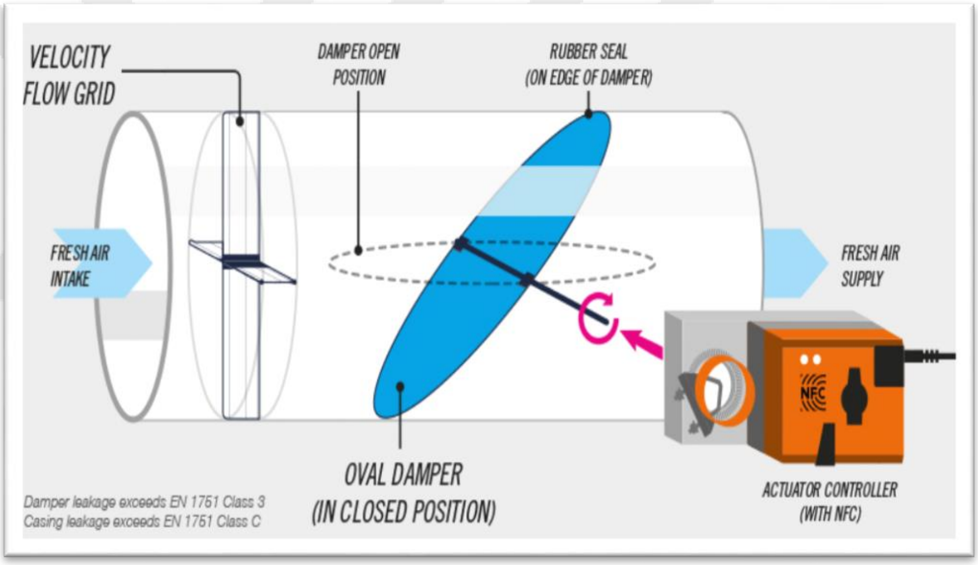

Figure 4.10: VAV Schematic with Actuator Controller.

The most common variable-airflow method is a closed-loop proportional-with-integral (PI) control, using the pressure measured at a selected point in the duct system. Most often, the set point is a constant, selected by the designer and confirmed by the balancer during system commissioning. If the pressure loop and flow loops are sufficiently decoupled, this is a solid, stable design. However, this control strategy is based on the readings of a single sensor that is assumed to represent the pressure available to all VAV boxes. Choosing duct pressure sensor location can be difficult. Though widely used, this fixed-pressure design uses more energy than necessary, even when applied. The Figures 4.11,4.12, are explaining the control system of VAV mechanism.

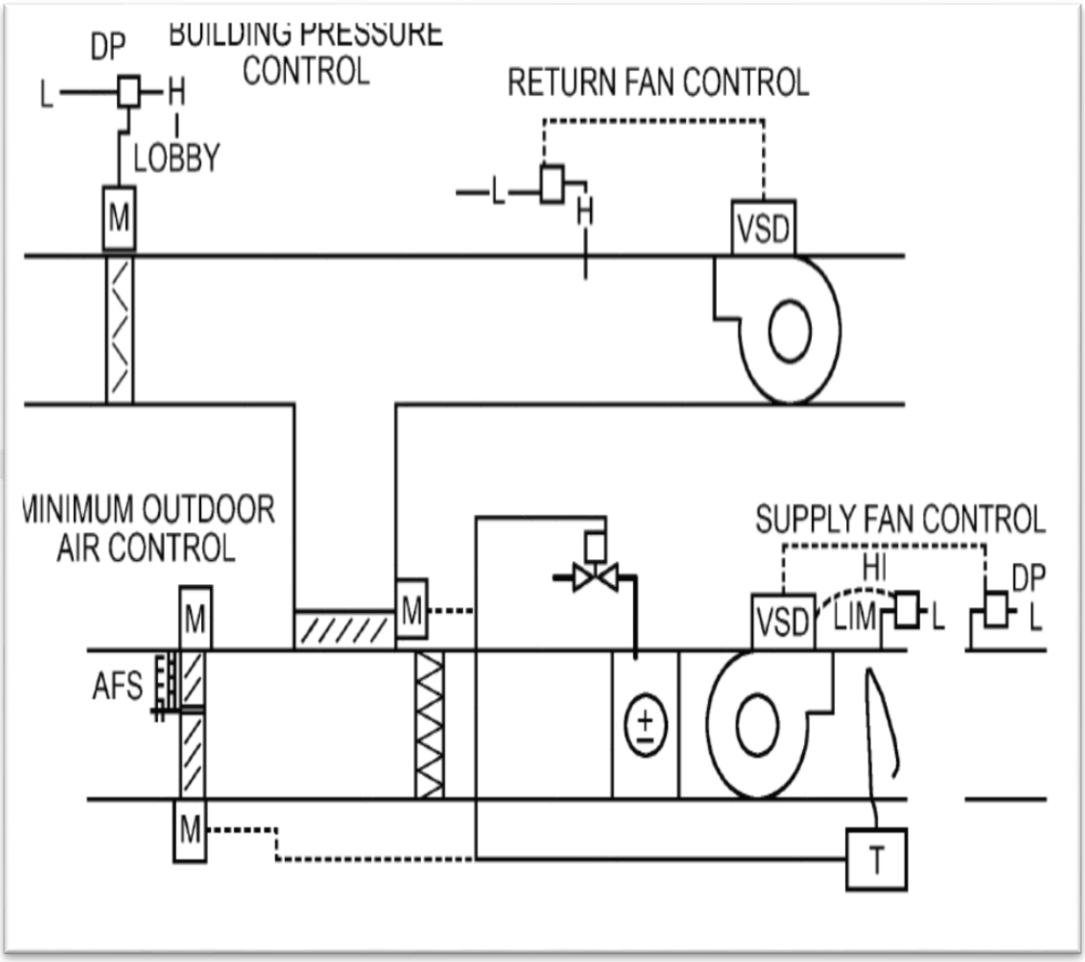

Figure 4.11: Schematic of VAV Control System.

Reported energy savings, monitored over weeks or months, have ranged from 30 to 50% of fan energy used by the same system running with a constant-pressure set point. All of these reset designs use data from terminal controllers to alter fan operation.

Most reset strategies use zone control data to adjust the set point of the duct pressure control loop. This makes the location of the pressure sensor much less important. In many cases, it makes the most sense to measure duct pressure near the fan outlet.

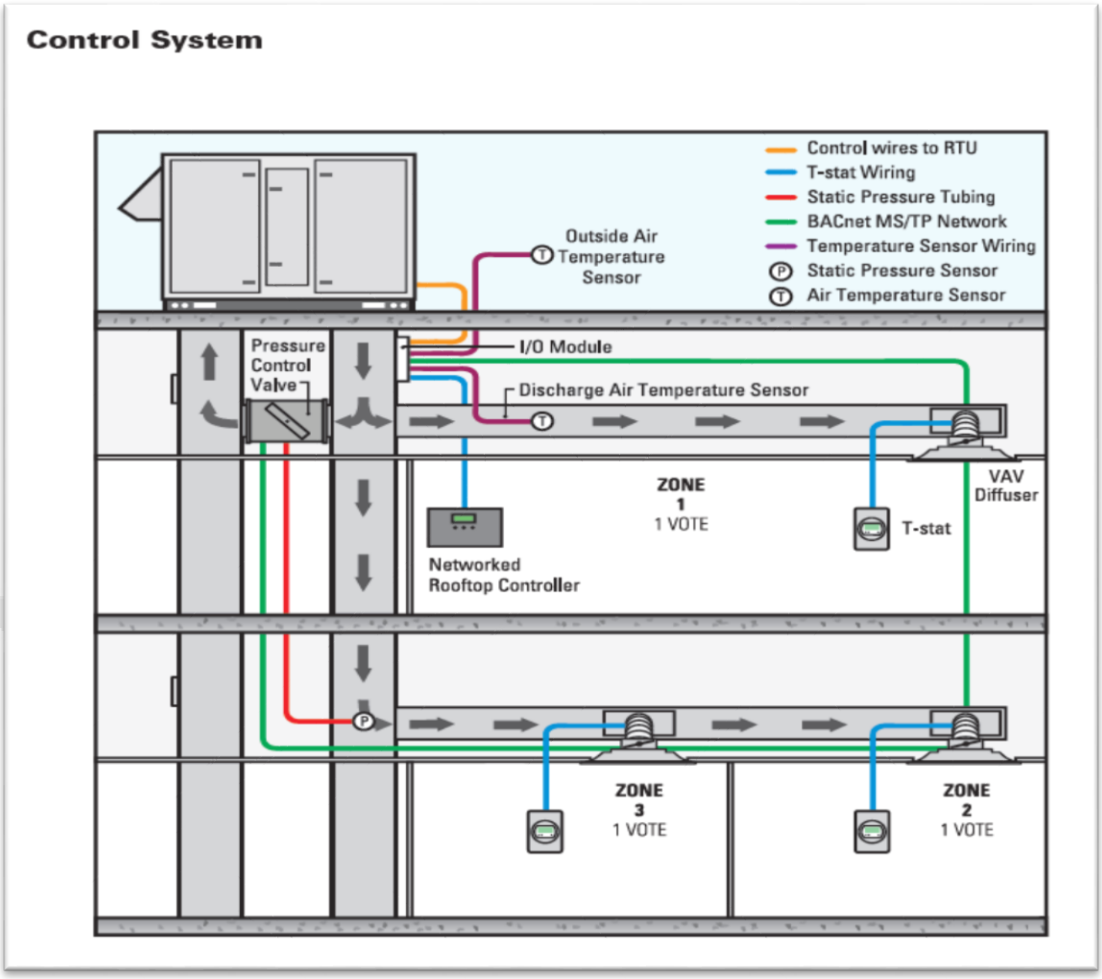

Figure 4.12: Layout of VAV Control System.

4.8 RESULTS

The total energy consumption by the different systems and buildings are as follow in Table 4.1 and Figure 4.13 for Abu Ghraib, Table 4.2 and Figure 4.14 for Rusafa.

Table 4.1: Total One-Year Energy Consumption in kWh for Abu Ghraib

HVAC System	Total Energy Consumption in kWh for one year
2 Stage Cooler CAV	408461
Package CAV	423236
Split CAV	412155
2 Stage Cooler VAV	282000
Package VAV	391397
Split VAV	371992

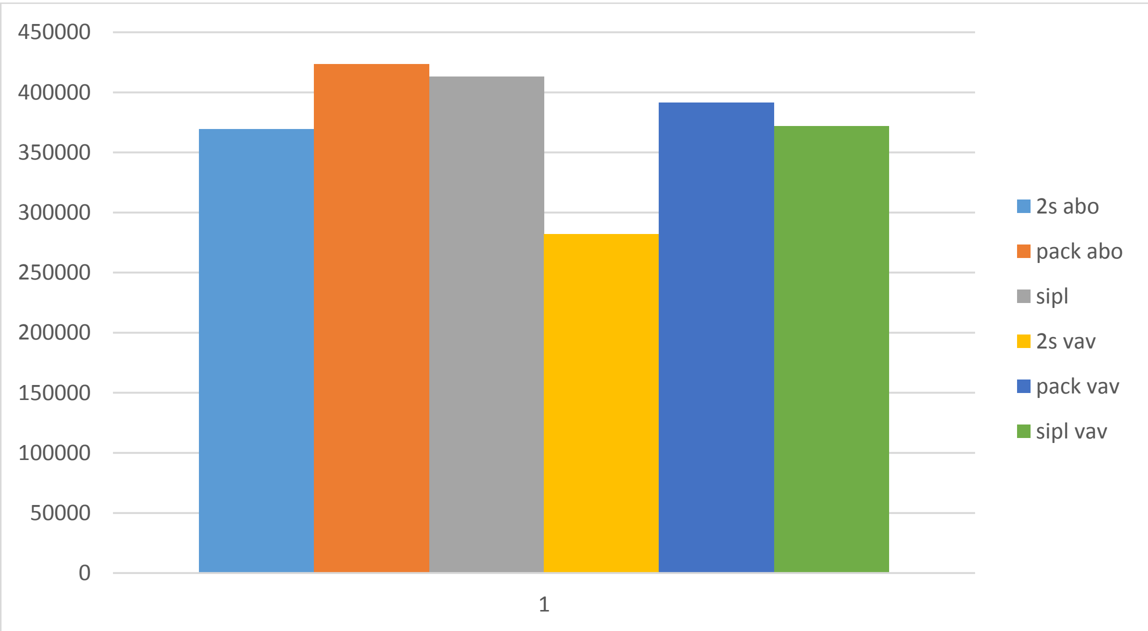

Figure 4.13: Total One-Year Energy Consumption in kWh for Abu Ghraib.

Table 4.2: Total One-Year Energy Consumption in kWh for Rusafa

HVAC system	Total Energy Consumption in kWh for one year
2 Stage Cooler CAV	39070
Package CAV	48228
Split CAV	50402
2 Stage Cooler VAV	21050
Package VAV	39186
Split VAV	42783

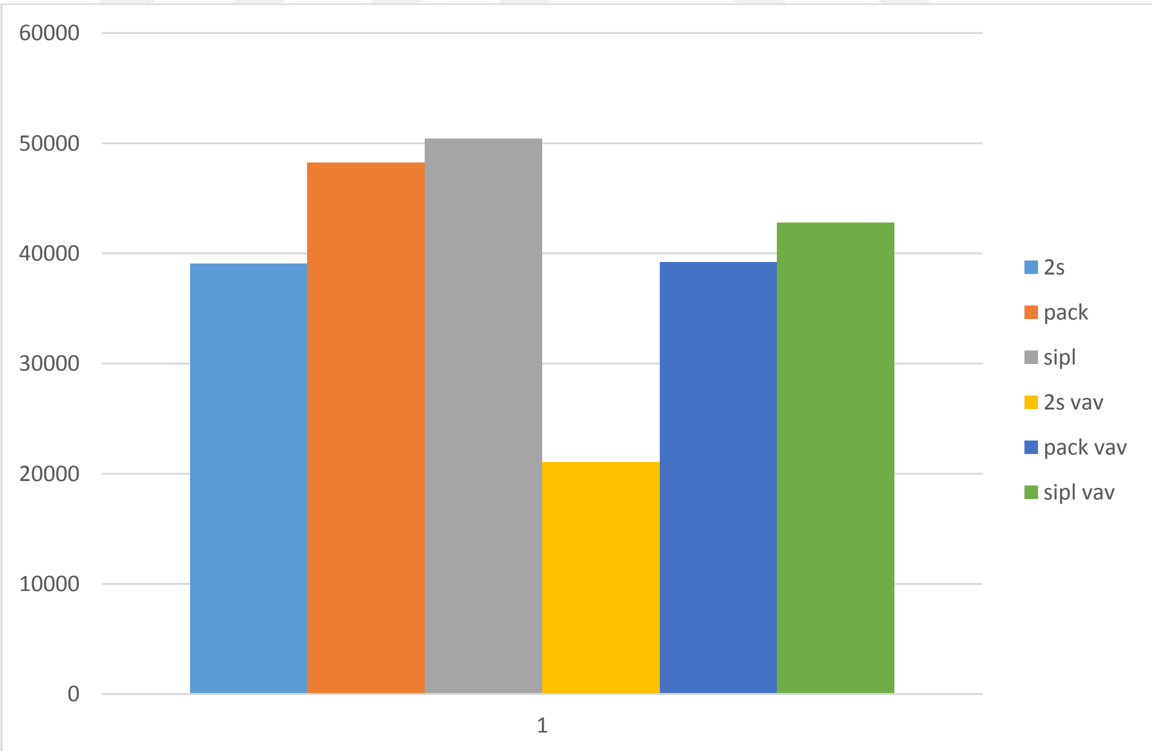

Figure 4.14: Total One-Year Energy Consumption in kWh for Rusafa.

It can be seen from figures 4.13 and 4.14 that the aggregate vitality utilization was diminished with around (10%-15%) for the split and package unit systems by utilizing the VAV control. For the used IDEC system, the energy consumption may be reduced within (30%-40%) since the main energy is the fan energy and no compressors are used in this system.

For the structures used, its seen that using the IDEC may decrease the consumed control fundamentally together with the VAV control we may reduce the imperativeness 30%-50% diverging from various frameworks. This is a considerable amount that drops the attention to this system even if its initial cost is high.

The Monthly energy consumption for CAV for Abu Ghraib, can be seen in Tables 4.3,4.4,4.5.

Table 4.3: Energy Consumption for 2 stage cooler CAV of Abu Ghraib

Month	Central Unit Clg Input (kWh)	Central Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)
January	0	682	18944	15624	35250
February	0	386	17110	14112	31608
March	0	0	18944	15624	34568
April	1	0	18333	15120	33454
May	3	0	18944	15624	34571
June	4	0	18333	15120	33457
July	4	0	18944	15624	34572
August	4	0	18944	15624	34572
Septembe r	3	0	18333	15120	33456
October	2	0	18944	15624	34570
November	0	0	18333	15120	33453
December	0	362	18944	15624	34930
Total	**21**	**1430**	**223048**	**183960**	**408461**

Table 4.4: Energy Consumption for Package CAV of Abu Ghraib

Month	Central Unit Clg Input (kWh)	Central Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)
January	0	4209	11506	15624	31339
February	0	2893	10392	14112	27397
March	0	55	11506	15624	27185
April	6221	0	11135	15120	32476
May	11715	0	11506	15624	38845
June	15790	0	11135	15120	42045
July	18819	0	11506	15624	45949
August	18447	0	11506	15624	45577
Septembe r	14042	0	11135	15120	40297
October	8534	0	11506	15624	35664
November	0	0	11135	15120	26255
December	0	3077	11506	15624	30207
Total	**93568**	**10235**	**135472**	**183960**	**423236**

Table 4.5: Energy Consumption for Split CAV of Abu Ghraib

Month	Central Unit Clg Input (kWh)	Central Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)
January	0	8747	5753	15624	30124
February	0	6325	5196	14112	25633
March	0	676	5753	15624	22053
April	8582	0	5567	15120	29269
May	17107	0	5753	15624	38484
June	23503	0	5567	15120	44190
July	28194	0	5753	15624	49571
August	27603	0	5753	15624	48980
Septembe r	20757	0	5567	15120	41444
October	12140	0	5753	15624	33517
November	0	0	5567	15120	20687
December	0	6826	5753	15624	28203
Total	**137886**	**22574**	**67736**	**183960**	**412155**

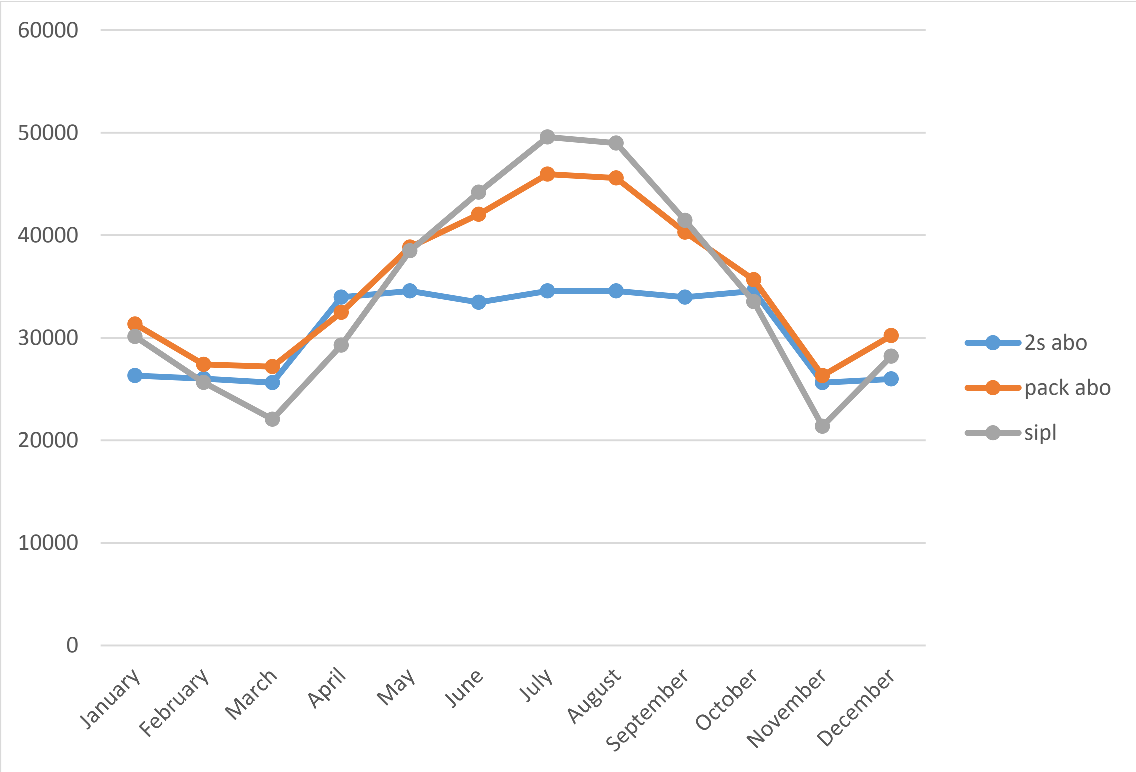

Figure 4.15: CAV Systems, Monthly Consumptions in kWh for Abu Ghraib.

As it is seen in Figures 4.15 & 4.16, the energy consumption is low during the first and fourth quarter of the year, this is because Iraq climatic conditions in Winter is moderate, and sometimes heating is not required at all when the internal load is high or it is crowded. On the other hand, the main load occurs in summer, which is one of the most sever climate on the world.

For the big building Two-speed fan was used with the IDEC system low that is used in low load season and high use with high load season.

The Monthly Energy Consumption for CAV for Rusafa, can be seen in Tables 4.6,4.7,4.8.

Table 4.6: Energy Consumption for 2 Stage Cooler CAV of Rusafa

Month	Central Unit Clg Input (kWh)	Central Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)
January	0	1910	2263	714	4887
February	0	933	2044	645	3622
March	0	12	2263	714	2989
April	1	0	2190	691	2882
May	3	0	2263	714	2980
June	3	0	2190	691	2884
July	4	0	2263	714	2981
August	4	0	2263	714	2981
September	3	0	2190	691	2884
October	2	0	2263	714	2979
November	0	0	2190	691	2881
December	0	1143	2263	714	4120
Total	**21**	**3998**	**26645**	**8407**	**39070**

Table 4.7: Energy Consumption for Package CAV of Rusafa

Month	Central Unit Clg Input (kWh)	Central Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)
January	0	2780	1374	714	4868
February	0	1592	1241	645	3478
March	0	355	1374	714	2443
April	1201	0	1330	691	3222
May	2337	0	1374	714	4425
June	3152	0	1330	691	5173
July	3765	0	1374	714	5853
August	3701	0	1374	714	5789
Septembe r	2815	0	1330	691	4836
October	0	0	1374	714	2088
November	0	44	1330	691	2065
December	0	1900	1374	714	3988
Total	**16971**	**6671**	**16179**	**8407**	**48228**

Table 4.8: Energy Consumption for Split CAV of Rusafa

Month	Central Unit Clg Input (kWh)	Central Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)
January	0	3469	687	714	4870
February	0	2102	621	645	3368
March	0	681	687	714	2082
April	1524	0	665	691	2880
May	3099	0	687	714	4500
June	4239	0	665	691	5595
July	5086	0	687	714	6487
August	4997	0	687	714	6398
Septembe r	3767	0	665	691	5123
October	2213	0	687	714	3614
November	0	233	665	691	1589
December	0	2495	687	714	3896
Total	**24925**	**8980**	**8090**	**8407**	**50402**

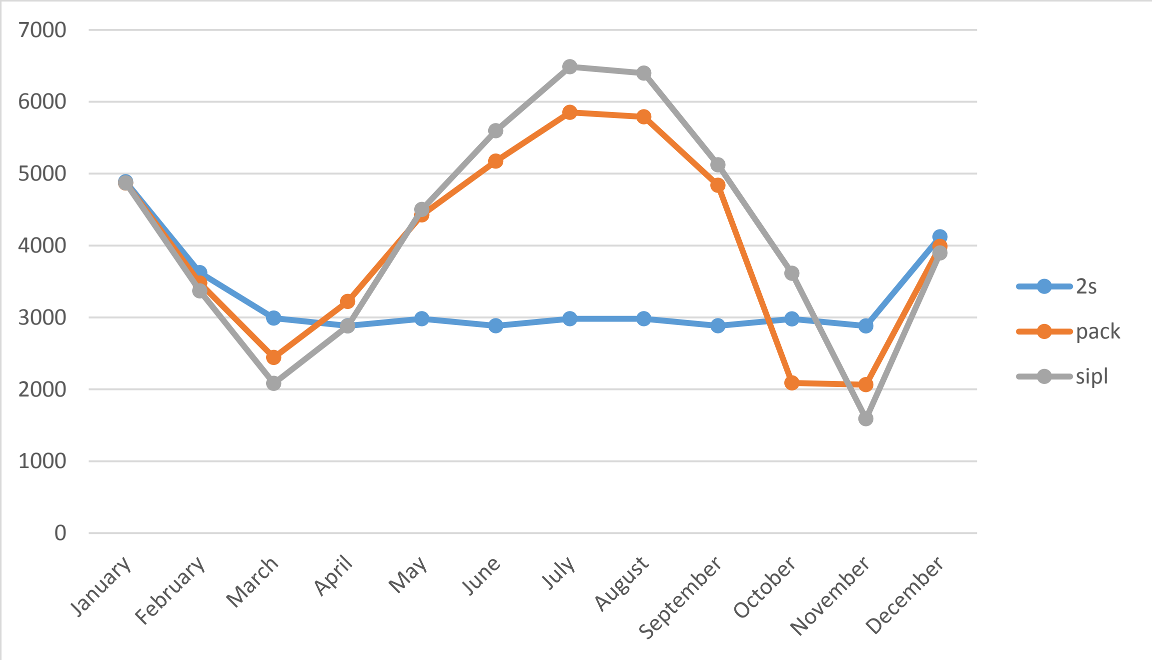

Figure 4.16: CAV Monthly Consumption in kWh for Rusafa.

It is seen that the Split Unit is the highest power consumption system in the high season. IDEC system is the lowest power consumption system among all systems.

The Monthly Energy Consumption for VAV for Abu Ghraib, can be seen in Tables 4.9,4.10,4.11.

Table 4.9: Energy Consumption for 2 Stage Cooler VAV of Abu Ghraib

Month	Central Unit Clg Input (kWh)	Terminal Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)
January	0	5031	4413	15624	25068
February	0	291	3986	14112	18389
March	0	0	5096	15624	20720
April	1	0	6961	15120	22082
May	3	0	9349	15624	24976
June	4	0	10772	15120	25896
July	4	0	11920	15624	27548
August	4	0	11715	15624	27343
Septembe r	3	0	9937	15120	25060
October	2	0	7975	15624	23601
November	0	0	5269	15120	20389
December	0	888	4416	15624	20928
Total	**95202**	**6210**	**91808**	**183960**	**282000**

Table 4.10: Energy Consumption for Package VAV of Abu Ghraib

Month	Central Unit Clg Input (kWh)	Terminal Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)
January	739	5573	2863	15624	24799
February	1047	478	2586	14112	18223
March	3027	0	3165	15624	21816
April	9461	0	4243	15120	28824
May	16885	0	5739	15624	38248
June	22287	0	6663	15120	44070
July	26150	0	7390	15624	49164
August	25880	0	7259	15624	48763
Septembe r	19898	0	6131	15120	41149
October	12291	0	4873	15624	32788
November	4164	0	3234	15120	22518
December	1224	1323	2864	15624	21035
Total	**143053**	**7375**	**57011**	**183960**	**391397**

Table 4.11: Energy Consumption for Split VAV of Abu Ghraib

Month	Central Unit Clg Input (kWh)	Terminal Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)
January	720	5573	1432	15624	23349
February	1026	478	1293	14112	16909
March	3063	0	1583	15624	20270
April	9828	0	2122	15120	27070
May	17806	0	2869	15624	36299
June	23687	0	3331	15120	42138
July	28524	0	3695	15624	47843
August	28130	0	3630	15624	47384
September	21089	0	3065	15120	39274
October	12840	0	2436	15624	30900
November	4242	0	1617	15120	20979
December	1198	1323	1432	15624	19577
Total	**152153**	**7375**	**28505**	**183960**	**371992**

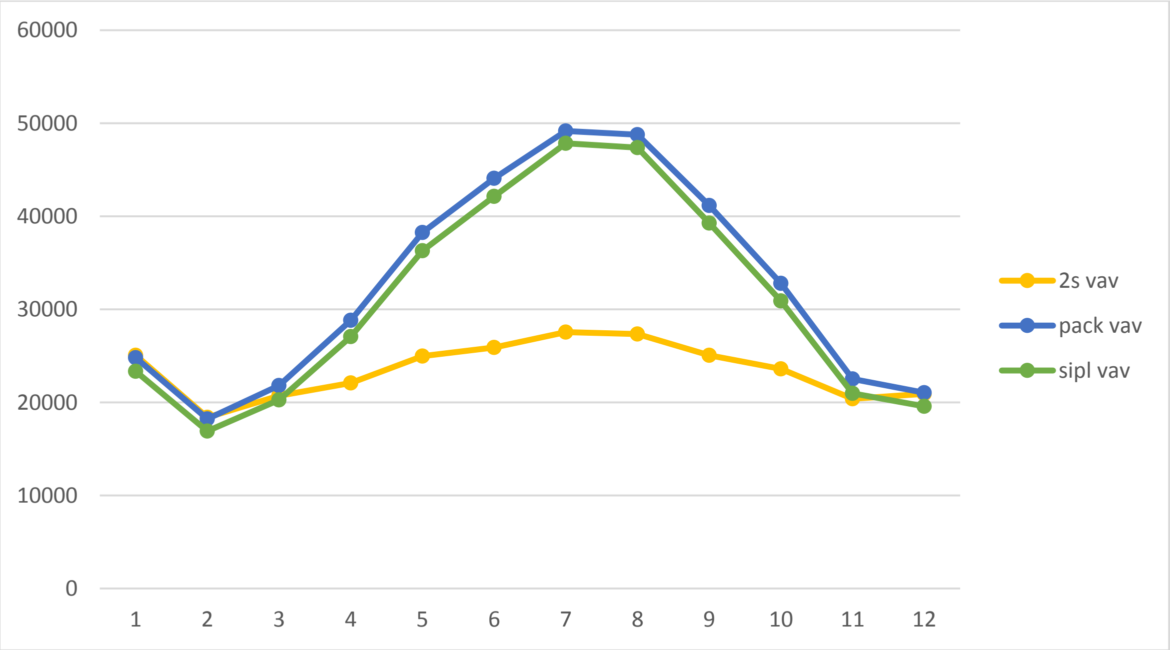

Figure 4.17: VAV Monthly Consumption in kWh for Abu Ghraib.

The Monthly Energy Consumption for VAV for Rusafa, can be seen in Tables 4.12,4.13,4.14.

Table 4.12: Energy Consumption for 2 Stage Cooler VAV of Rusafa

Month	Central Unit Clg Input (kWh)	Terminal Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)
January	1	52	624	714	1391
February	1	2	585	645	1233
March	1	0	775	714	1490
April	2	0	982	691	1675
May	3	0	1259	714	1976
June	4	0	1419	691	2114
July	5	0	1552	714	2271
August	5	0	1528	714	2247
Septembe r	4	0	1323	691	2018
October	3	0	1095	714	1812
November	1	0	781	691	1473
December	1	4	631	714	1350
Total	**31**	**58**	**12554**	**8407**	**21050**

Table 4.13: Energy Consumption for Package VAV of Rusafa

Month	Central Unit Clg Input (kWh)	Terminal Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)
January	170	604	436	714	1924
February	247	102	394	645	1388
March	597	0	483	714	1794
April	1552	0	602	691	2845
May	2533	0	779	714	4026
June	3220	0	881	691	4792
July	3877	0	967	714	5558
August	3855	0	951	714	5520
Septembe r	2930	0	820	691	4441
October	1900	0	672	714	3286
November	792	0	481	691	1964
December	290	208	436	714	1648
Total	**21963**	**914**	**7902**	**8407**	**39186**

Table 4.14: Energy Consumption for Split VAV of Rusafa

Month	Central Unit Clg Input (kWh)	Terminal Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)
January	221	604	218	714	1757
February	323	102	197	645	1267
March	788	0	242	714	1744
April	2073	0	301	691	3065
May	3404	0	389	714	4507
June	4337	0	441	691	5469
July	5238	0	483	714	6435
August	5208	0	475	714	6397
September	3945	0	410	691	5046
October	2543	0	336	714	3593
November	1051	0	241	691	1983
December	380	208	218	714	1520
Total	**29511**	**914**	**3951**	**8407**	**42783**

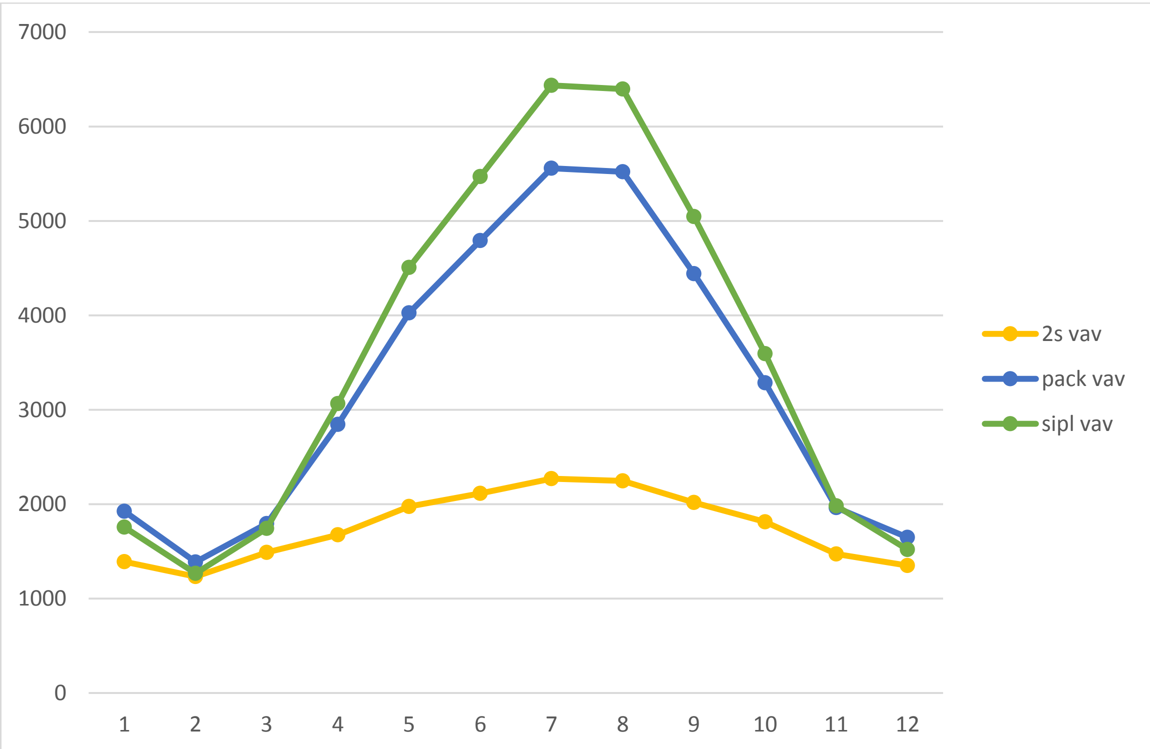

Figure 4.18: VAV Monthly Consumption in kWh for Rusafa.

Figures 4.17 & 4.18, show the Monthly consumption using variable air volume strategy. Again, the same results but with lower values for all systems. It can be seen that the IDEC with VAV control strategy shows much better results. It is also seen that in a big building where large air volume is needed split unit gives similar results to the package unit.

The reduction in power consumption can be seen very clearly using the VAV strategy with IDEC system and since the air flow varies with load the annual consumption was reduced more than the monthly consumption.

On the following Figures,4.19,4.20 CAV & VAV were shown together for each system and for each building this may show the benefit of using the VAV system and how much reduction on energy it will cause. From these figures, it is shown that using VAV strategy will reduce the energy considerably for the IDEC system while in other systems the reduction was not high by reducing the air volume only as compared with IDEC.

It can be seen that the power at winter on CAV is higher than expected especially for the small building Rusafa, this happened because of high air volume that was calculated to overcome the cooling load in summer. This high air volume required more heating to keep the air at the required temperature.

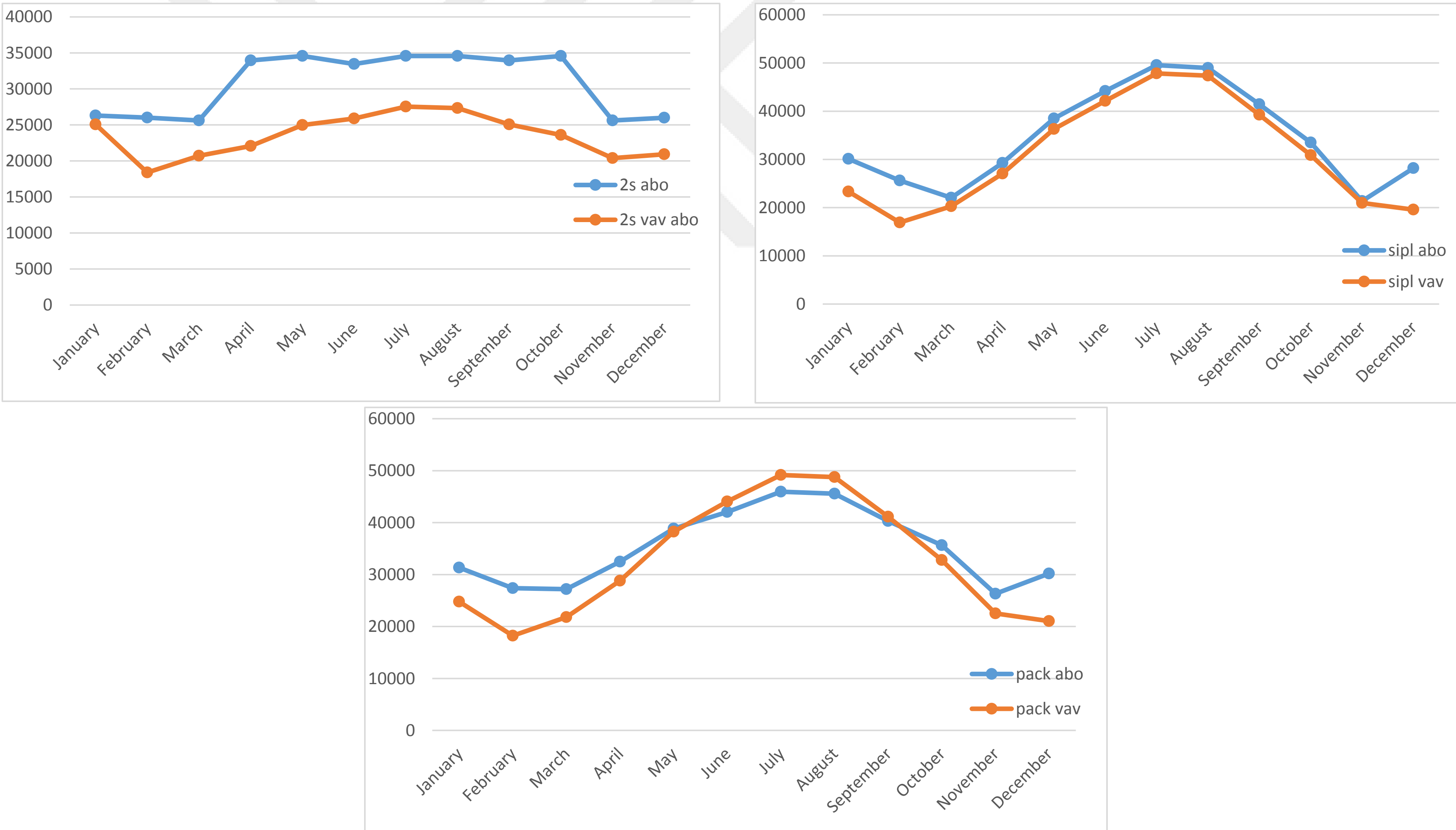

Figure 4.19: Energy Consumption Between CAV & VAV in kWh for 2stage, package, spilt for Abu Ghraib.

Figure 4.20: Energy Consumption Between CAV & VAV in kWh for 2stage, package, spilt for Rusafa.

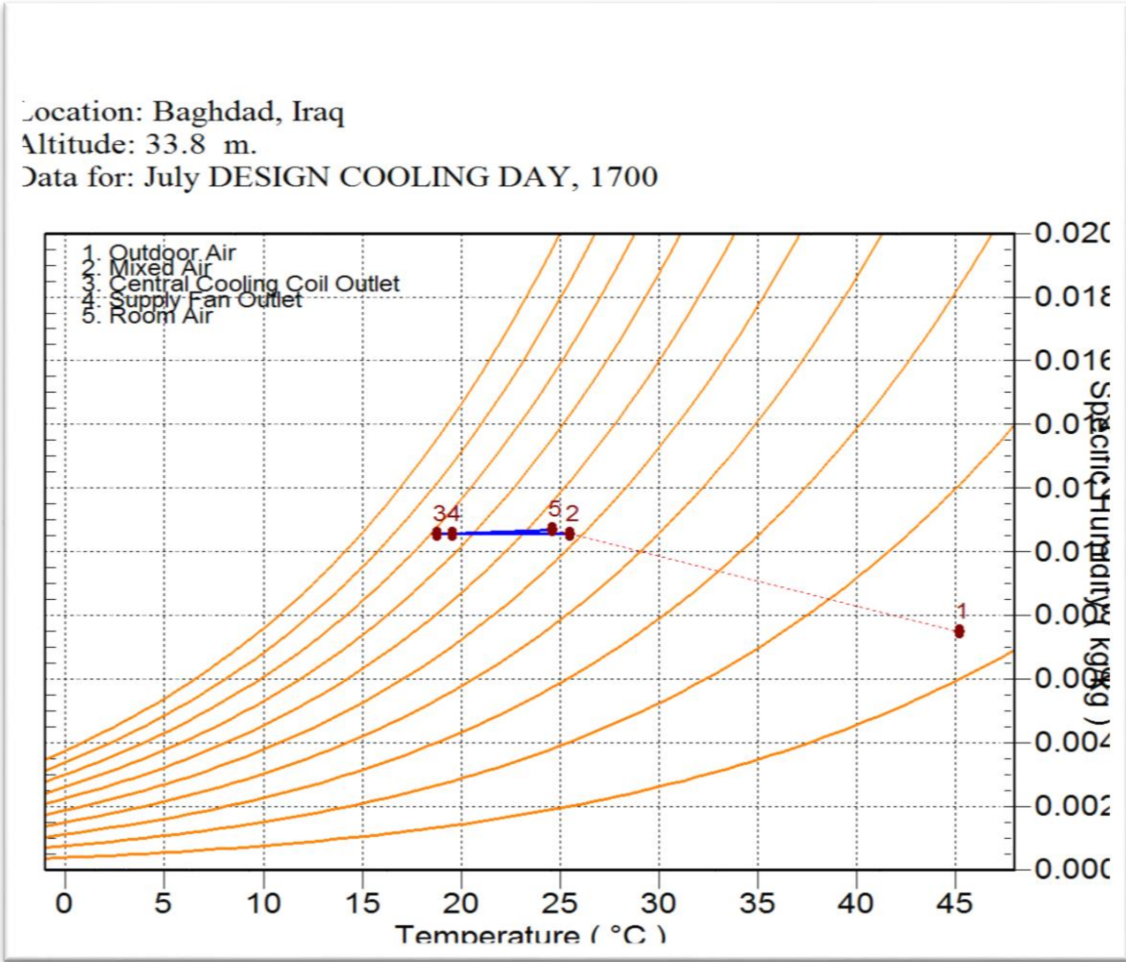

Figure 4.21: Psychometric Chart for The Hottest Day.

The above chart in Figure 4.21, is the Psychometric chart for the hottest day which shows the conditions of each point of the conditioned space, from which we can know the temperature energy change of the supplied air, where:

1. Outdoor temperature.
2. Mixed Air.
3. Cooling Coil Outlet.
4. Supply Fan Outlet.
5. Room Air.

On this well-known chart the horizontal axis represents the dry bulb temperature the curved lines represent the humidity percentage, the vertical axis represents the water content. From this chart we can understand the cooling or heating process on air and how the properties are changed on each part of the system, for more information about Psychometric chart we can see in Figure 3.10 / chapter three.

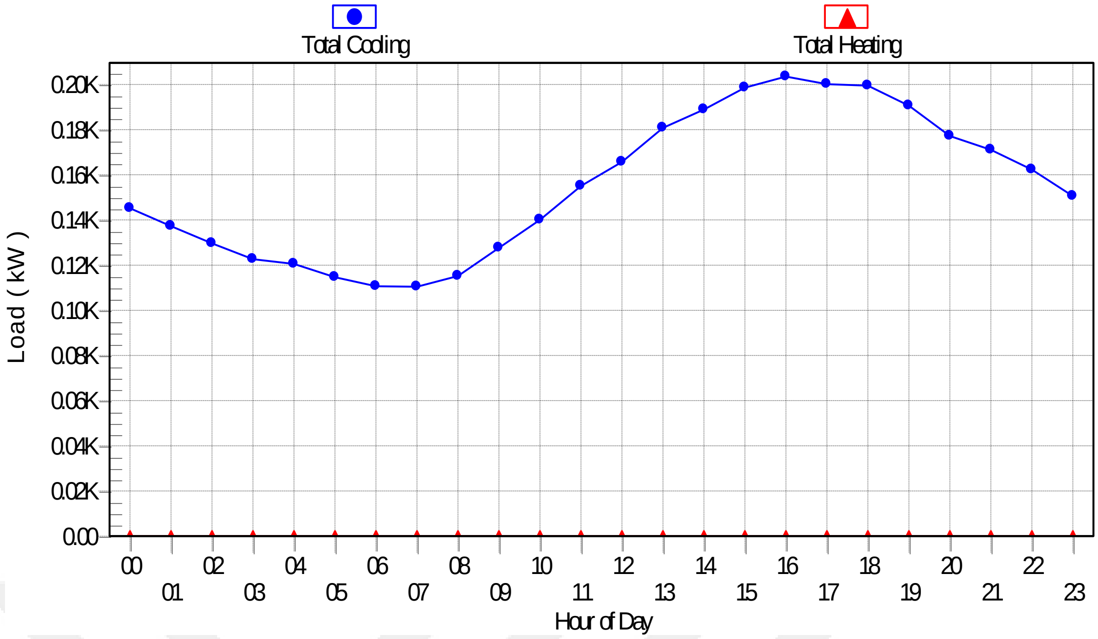

Figure 4.22: Hourly Air System Design Day Loads at July for Abu Ghraib.

Data of January

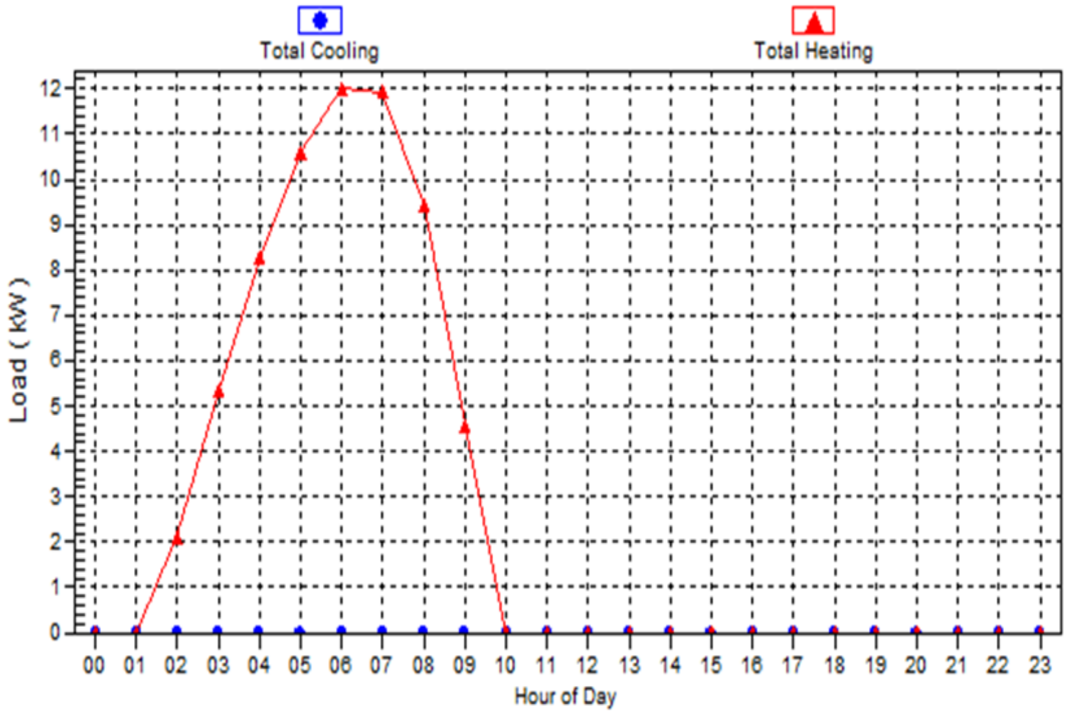

Figure 4.23: Hourly Air System Design Day Loads at January for Abu Ghraib.

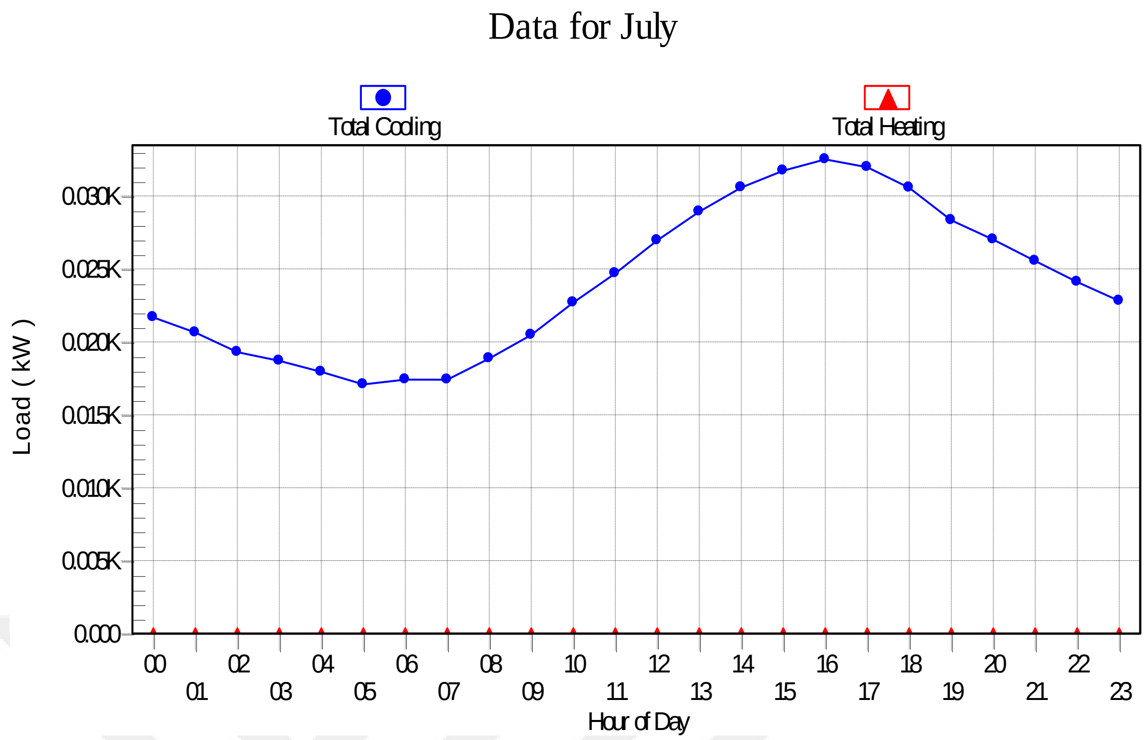

Figure 4.24: Hourly Air System Design Day Loads at July for Rusafa.

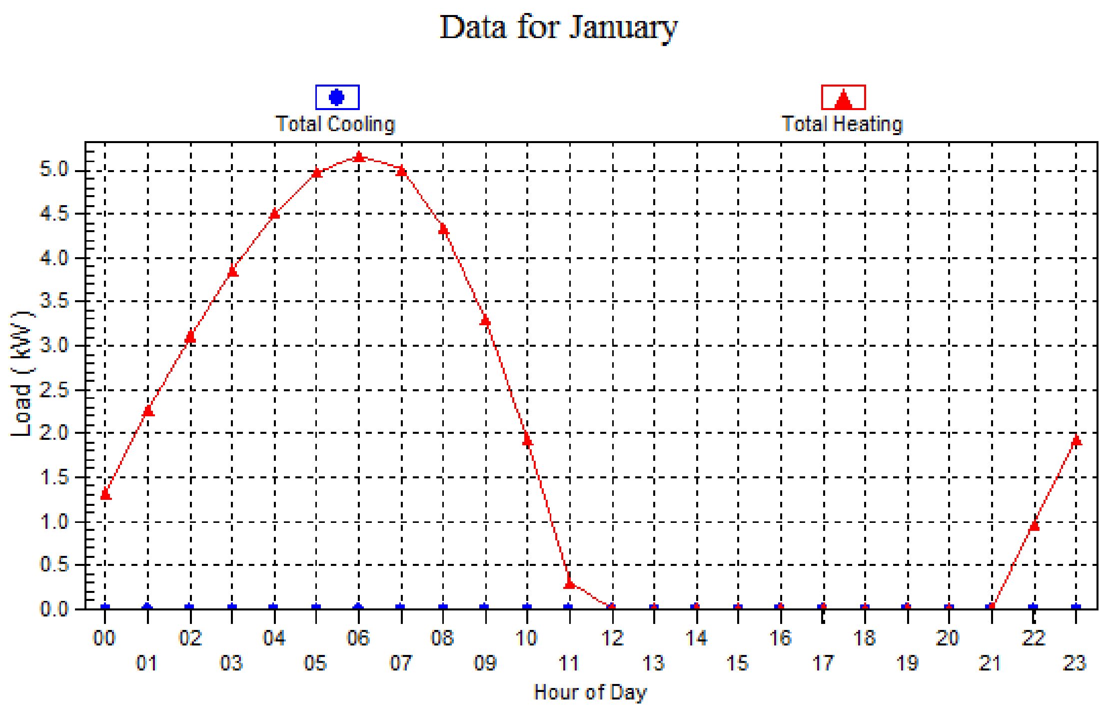

Figure 4.25: Hourly Air System Design Day Loads at January for Rusafa.

It can be seen from the above the charts in Figures 4.22,4.23,4.24,4.25, that show how much the fluctuation of energy consumption occur through the day. This fluctuation may be utilized to reduce the consumption by many ways, one of the most effective ways is to change the air flow with the load. This is known as Variable Air Volume (VAV) so the fan power may be reduced and this may affect the power consumption rapidly.

We should calculate the incremental in power consumption in every kind of systems, the calculate the selling price and cost price to see the value of the benefit for each system as a statistic. From the total energy the software HAP can calculate the cost of the power according to the billing prices, it also can calculate complex billing charges depending on demand, categories, consumption, season, or even all. We have put the cost of governmental consumption on the software as shown in Tables 4.15,4.16,4.17,4.18,4.19,4.20, Figure 4.26, for Abu Ghraib, this was as per latest Ministry of Electricity billing criteria, which was as follow:

From 1-5000 kwh cost (125) IQD that is (0.1) USD

From 5001-10000 kwh cost (150) IQD that is (0.12) USD

From 10001-20000 kwh cost (175) IQD that is (0.14) USD

From 20001-40000 kwh cost (200) IQD that is (0.16) USD

Above 40001 kwh cost (225) IQD that is (0.18) USD.

Table 4.15: Total Billing Price of 2 Stage Cooler CAV in Abu Ghraib

Month	Central Unit Clg Input (kWh)	Central Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)	Avg Unit Price USD $	Total Billing Price USD $
January	0	682	10000	15624	26306	0.11	1100
February	0	386	10000	15624	26010	0.11	1100
March	0	0	10000	15624	25624	0.11	1100
April	1	0	18333	15624	33958	0.1307	4916
May	3	0	18944	15624	34571	0.1317	5117
June	4	0	18333	15120	33457	0.1307	4916
July	4	0	18944	15624	34572	0.1317	5117
August	4	0	18944	15624	34572	0.1317	5117
Septembe r	3	0	18333	15624	33960	0.1307	4916
October	2	0	18944	15624	34570	0.1317	5117
November	0	0	10000	15624	25624	0.11	1100
December	0	362	10000	15624	25986	0.11	1100
Total	**21**	**1430**	**180775**	**186984**	**369210**		**40716**

Table 4.16: Total Billing Price of Package CAV in Abu Ghraib

Month	Central Unit Clg Input (kWh)	Central Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)	Avg Unit Price USD $	Total Billing Price USD $
January	0	4209	11506	15624	31339	0.1273	3,998
February	0	2893	10392	14112	27397	0.1254	3,445
March	0	55	11506	15624	27185	0.1253	3,416
April	6221	0	11135	15120	32476	0.1277	4,157
May	11715	0	11506	15624	38845	0.1317	5,127
June	15790	0	11135	15120	42045	0.1339	5,639
July	18819	0	11506	15624	45949	0.1361	6,264
August	18447	0	11506	15624	45577	0.1359	6,204
Septembe r	14042	0	11135	15120	40297	0.1328	5,359
October	8534	0	11506	15624	35664	0.1292	4,618
November	0	55	11135	15120	26310	0.1248	3,286
December	0	3077	11506	15624	30207	0.1268	3,839
Total	**93568**	**10235**	**135472**	**183960**	**423291**	**0.1305**	**55352**

Table 4.17: Total Billing Price of Split CAV in Abu Ghraib

Month	Central Unit Clg Input (kWh)	Central Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)	Avg Unit Price USD $	Total Billing Price USD $
January	0	8747	5753	15624	30124	0.1268	3,828
February	0	6325	5196	14112	25633	0.1244	3,198
March	0	676	5753	15624	22053	0.1219	2,698
April	8582	0	5567	15120	29269	0.1264	3,708
May	17107	0	5753	15624	38484	0.1315	5,069
June	23503	0	5567	15120	44190	0.1351	5,982
July	28194	0	5753	15624	49571	0.1378	6,843
August	27603	0	5753	15624	48980	0.1376	6,749
Septembe r	20757	0	5567	15120	41444	0.1335	5,543
October	12140	0	5753	15624	33517	0.1281	4,303
November	0	676	5567	15120	21363	0.1207	2,506
December	0	6826	5753	15624	28203	0.1259	3,559
Total	**137886**	**22574**	**67736**	**183960**	**412831**		**53986**

Table 4.18: Total Billing Price of 2Stage Cooler VAV in Abu Ghraib

Month	Central Unit Clg Input (kWh)	Terminal Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)	Avg Unit Price USD $	Total Billing Price USD $
January	0	5031	4413	15624	25068	0.1254	3,431
February	0	291	3986	14112	18389	0.1195	2,326
March	0	0	5096	15624	20720	0.1217	2,664
April	1	0	6961	15120	22082	0.1231	2,913
May	3	0	9349	15624	24976	0.1252	3,386
June	4	0	10772	15120	25896	0.1258	3,549
July	4	0	11920	15624	27548	0.1267	3,812
August	4	0	11715	15624	27343	0.1266	3,778
Septembe r	3	0	9937	15120	25060	0.1253	3,411
October	2	0	7975	15624	23601	0.1243	3,156
November	0	0	5269	15120	20389	0.1215	2,625
December	0	888	4416	15624	20928	0.122	2,716
Total	**95202**	**6210**	**91808**	**183960**	**282000**	**0.1242**	**37767**

Table 4.19: Total Billing Price of Package VAV in Abu Ghraib

Month	Central Unit Clg Input (kWh)	Terminal Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)	Avg Unit Price USD $	Total Billing Price USD $
January	739	5573	2863	15624	24799	0.1239	3,082
February	1047	478	2586	14112	18223	0.1181	2,161
March	3027	0	3165	15624	21816	0.1217	2,665
April	9461	0	4243	15120	28824	0.1262	3,645
May	16885	0	5739	15624	38248	0.1313	5,031
June	22287	0	6663	15120	44070	0.1351	5,963
July	26150	0	7390	15624	49164	0.1377	6,778
August	25880	0	7259	15624	48763	0.1375	6,714
Septembe r	19898	0	6131	15120	41149	0.1333	5,495
October	12291	0	4873	15624	32788	0.1278	4,201
November	4164	0	3234	15120	22518	0.1223	2,763
December	1224	1323	2864	15624	21035	0.1211	2,555
Total	**143053**	**7375**	**57011**	**183960**	**391397**		**51053**

Table 4.20: Total Billing Price of Split VAV in Abu Ghraib

Month	Central Unit Clg Input (kWh)	Terminal Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)	Avg Unit Price USD $	Total Billing Price USD $
January	720	5573	1432	15624	23349	0.1229	2,879
February	1026	478	1293	14112	16909	0.1164	1,977
March	3063	0	1583	15624	20270	0.1203	2,448
April	9828	0	2122	15120	27070	0.1253	3,400
May	17806	0	2869	15624	36299	0.1298	4,720
June	23687	0	3331	15120	42138	0.1339	5,654
July	28524	0	3695	15624	47843	0.137	6,567
August	28130	0	3630	15624	47384	0.1368	6,493
September	21089	0	3065	15120	39274	0.132	5,195
October	12840	0	2436	15624	30900	0.1271	3,936
November	4242	0	1617	15120	20979	0.121	2,547
December	1198	1323	1432	15624	19577	0.1196	2,351
Total	**152153**	**7375**	**28505**	**183960**	**371992**		**48167**

The following values for cost were obtained in Tables 4.21,4.22,4.23,4.24,4.25,4.26, Figure 4.27, for Rusafa, these charts tend just like energy but the cost will increase if the building was bigger and consumption was more.

Table 4.21: Total Billing Price of 2 Stage Cooler CAV in Rusafa

Month	Central Unit Clg Input (kWh)	Central Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)	Avg Unit Price USD $	Total Billing Price USD $
January	0	1910	2263	714	4887	0.1	488.7
February	0	933	2044	645	3622	0.1	362.2
March	0	12	2263	714	2989	0.1	298.9
April	1	0	2190	691	2882	0.1	288.2
May	3	0	2263	714	2980	0.1	298
June	3	0	2190	691	2884	0.1	288.4
July	4	0	2263	714	2981	0.1	298.1
August	4	0	2263	714	2981	0.1	298.1
September	3	0	2190	691	2884	0.1	288.4
October	2	0	2263	714	2979	0.1	297.9
November	0	0	2190	691	2881	0.1	288.1
December	0	1143	2263	714	4120	0.1	412
Total	**21**	**3998**	**26645**	**8407**	**39070**		**3907**

Table 4.22: Total Billing Price of Package CAV in Rusafa

Month	Central Unit Clg Input (kWh)	Central Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)	Avg Unit Price USD $	Total Billing Price USD $
January	0	2780	1374	714	4868	0.1	486.8
February	0	1592	1241	645	3478	0.1	347.8
March	0	355	1374	714	2443	0.1	244.3
April	1201	0	1330	691	3222	0.1	322.2
May	2337	0	1374	714	4425	0.1	442.5
June	3152	0	1330	691	5173	0.1	517.3
July	3765	0	1374	714	5853	0.1	585.3
August	3701	0	1374	714	5789	0.1	578.9
September	2815	0	1330	691	4836	0.1	483.6
October	0	0	1374	714	2088	0.1	208.8
November	0	44	1330	691	2065	0.1	206.5
December	0	1900	1374	714	3988	0.1	398.8
Total	**16971**	**6671**	**16179**	**8407**	**48228**		**4822.8**

Table 4.23: Total Billing Price of Split CAV in Rusafa

Month	Central Unit Clg Input (kWh)	Central Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)	Avg Unit Price USD $	Total Billing Price USD $
January	0	3469	687	714	4870	0.1	487
February	0	2102	621	645	3368	0.1	336.8
March	0	681	687	714	2082	0.1	208.2
April	1524	0	665	691	2880	0.1	288
May	3099	0	687	714	4500	0.1	450
June	4239	0	665	691	5595	0.1	559.5
July	5086	0	687	714	6487	0.1	648.7
August	4997	0	687	714	6398	0.1	639.8
September	3767	0	665	691	5123	0.1	512.3
October	2213	0	687	714	3614	0.1	361.4
November	0	233	665	691	1589	0.1	158.9
December	0	2495	687	714	3896	0.1	389.6
Total	**24925**	**8980**	**8090**	**8407**	**50402**		**5040.2**

Table 4.24: Total Billing Price of 2 Stage Cooler VAV in Rusafa

Month	Central Unit Clg Input (kWh)	Terminal Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)	Avg Unit Price USD $	Total Billing Price USD $
January	1	52	624	714	1391	0.1	139.1
February	1	2	585	645	1233	0.1	123.3
March	1	0	775	714	1490	0.1	149
April	2	0	982	691	1675	0.1	167.5
May	3	0	1259	714	1976	0.1	197.6
June	4	0	1419	691	2114	0.1	211.4
July	5	0	1552	714	2271	0.1	227.1
August	5	0	1528	714	2247	0.1	224.7
September	4	0	1323	691	2018	0.1	201.8
October	3	0	1095	714	1812	0.1	181.2
November	1	0	781	691	1473	0.1	147.3
December	1	4	631	714	1350	0.1	135
Total	**31**	**58**	**12554**	**8407**	**21050**		**2105**

Table 4.25: Total Billing Price of Package VAV in Rusafa

Month	Central Unit Clg Input (kWh)	Terminal Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)	Avg Unit Price USD $	Total Billing Price USD $
January	170	604	436	714	1924	0.1	192.4
February	247	102	394	645	1388	0.1	138.8
March	597	0	483	714	1794	0.1	179.4
April	1552	0	602	691	2845	0.1	284.5
May	2533	0	779	714	4026	0.1	402.6
June	3220	0	881	691	4792	0.1	479.2
July	3877	0	967	714	5558	0.1	555.8
August	3855	0	951	714	5520	0.1	552
Septembe r	2930	0	820	691	4441	0.1	444.1
October	1900	0	672	714	3286	0.1	328.6
November	792	0	481	691	1964	0.1	196.4
December	290	208	436	714	1648	0.1	164.8
Total	**21963**	**914**	**7902**	**8407**	**39186**		**3918.6**

Table 4.26: Total Billing Price of Split VAV in Rusafa

Month	Central Unit Clg Input (kWh)	Terminal Heating Coil Input (kWh)	Supply Fan (kWh)	Lighting (kWh)	Total (kWh)	Avg Unit Price USD $	Total Billing Price USD $
January	221	604	218	714	1757	0.1	175.7
February	323	102	197	645	1267	0.1	126.7
March	788	0	242	714	1744	0.1	174.4
April	2073	0	301	691	3065	0.1	306.5
May	3404	0	389	714	4507	0.1	450.7
June	4337	0	441	691	5469	0.1	546.9
July	5238	0	483	714	6435	0.1	643.5
August	5208	0	475	714	6397	0.1	639.7
Septembe r	3945	0	410	691	5046	0.1	504.6
October	2543	0	336	714	3593	0.1	359.3
November	1051	0	241	691	1983	0.1	198.3
December	380	208	218	714	1520	0.1	152
Total	**29511**	**914**	**3951**	**8407**	**42783**		**4278.3**

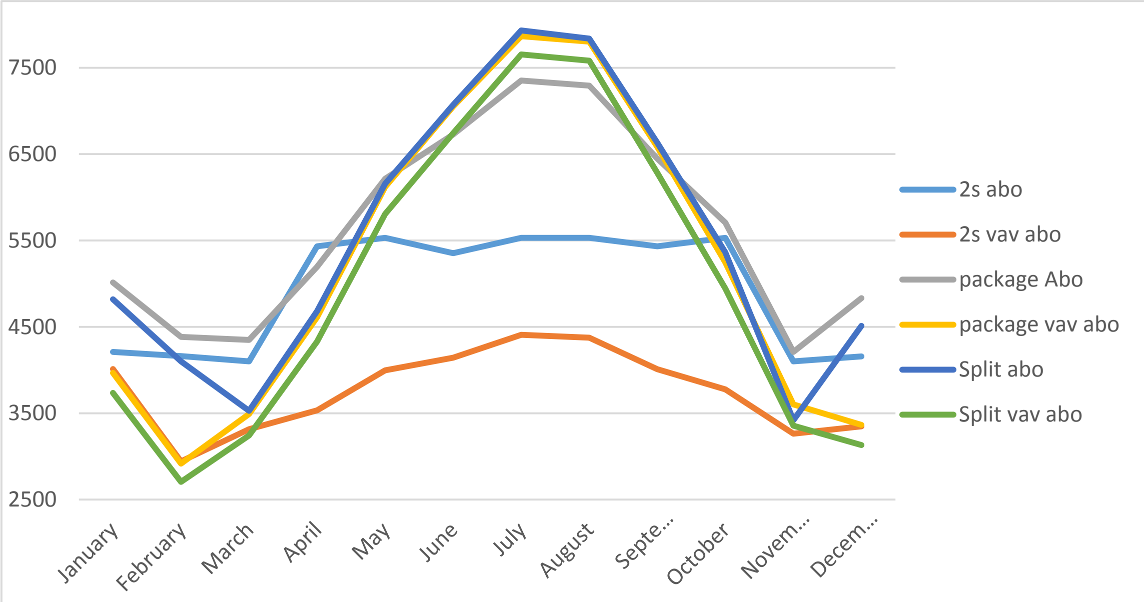

Figure 4.26: Total Cost for Abu Ghraib HVAC Systems in **USD $**

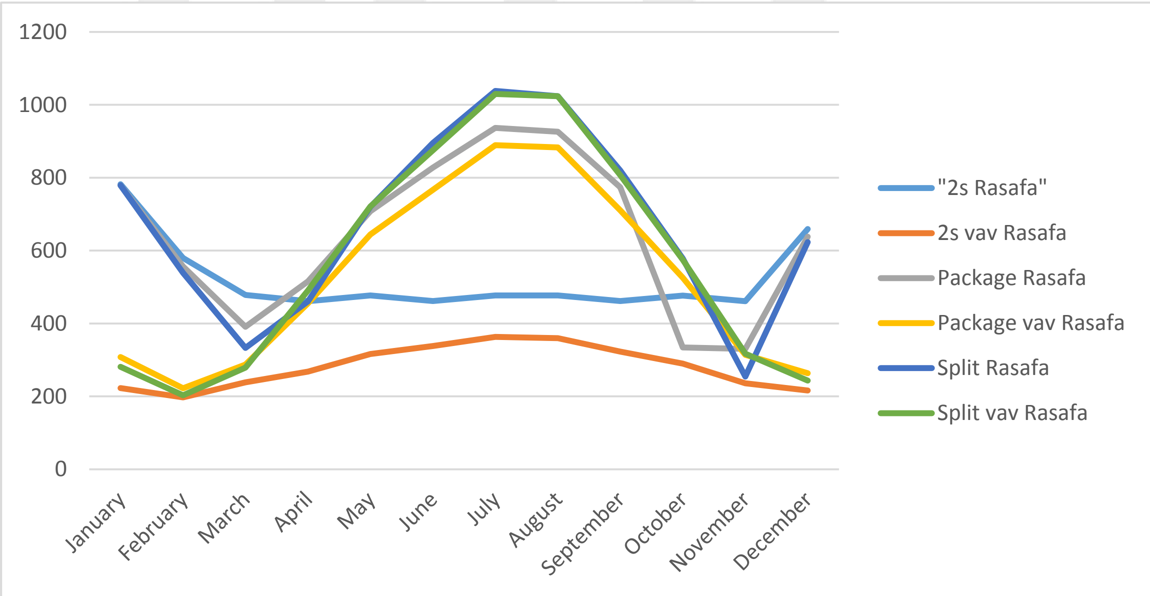

Figure 4.27: Total Cost for Rusafa HVAC Systems in **USD $**

Detail results may be found on annexures that show the simulation results obtained from the HAP program.

4.9 SUMMERY & DISCUSSION

From the previous sections, it has seen that:

- Split units are the highest energy demand on all cases. If we connect this with the fact that the largely used Split systems are with no fresh air at all, and crowded places need more fresh air. This will increase the energy consumption even more, plus the presence of the indoor units inside the building of the prison, which represent more challenges. We may say that this system is the worst choice to be used in cell halls. Providing that it should never be used with any patient halls without the required filters and fresh air as per standards.

- For package units it is the first choice a designer may think of. This system is good especially for medium buildings (very large systems will use chilled water AHU for each zone). Although, the ducting system will be larger. That means more energy is needed to supply the required air. However, compacted one place equipment is much more efficient and the total energy will be less in most cases than the split unit. When the load is minimum and the unit is working then the energy will represent the air supply and it will be less efficient for constant air volume. Since the difference between these systems is the supplying air energy then the use of Variable air system will reduce the energy for supplying air at less demand points. This will give an advantage for the package units over the split units regarding the energy consumption.

- For the 2 Stage (one media) Direct Indirect Evaporative coolers IDEC, energy is consumed due to air supply manly, all other accessories like pumps, dampers, etc. do not represent any important figure comparing with large air quantities required for keeping the indoor conditions same as other systems. Although, air quantity will be some times as large as three times other systems required but the absence of the compressor will give the advantage for this systems especially on high season. Again since we are talking about air supply the it is seen that using variable air volume will decrease the required energy dramatically and it is shown that the total yearly energy consumption can be reduced as low as 40% of any other system.

- Keeping on mined that the IDEC will utilize totally natural air this will be ideal for patient prisons, doctor's facilities, therapeutic focuses, and swarmed jail, cell halls.

CHAPTER FIVE

Conclusions and Suggestion for Future Works

5.1 Conclusions

The studies in this thesis are mainly focused on a protocol of the reduction of energy in HVAC systems in prisons halls in Iraq, the following conclusions are reached during this investigation:

1. Using VAV system is recommended since it can reduce energy considerably.
2. Changing to new efficient systems like IDEC coolers rather than the classic systems should be considered, when possible, and may provide a huge reduction on energy that can reach to about 50% or even more.
3. Using modern control and operating systems is recommended and will control the environment and reduce energy.
4. In some cases, rooms can be conditioned using out door air only without using any cooling or heating equipment, like execution platform.
5. Connecting the ventilation with conditioning system will by more efficient and controllable than using a separate system for ventilation.
6. For large facilities, split small units should be avoided and one central unit should be used instead.
7. To reduce the power and keep a good performance a reputable engineering company should be consultant to select the suitable efficient system.

5.2 Future Works Can Be Carried Out On the Following Issues

1. The effect of installing a Building Energy Management System (BMS or BEMS), which offers close control and monitoring of building services performance, including Heating, Ventilation and Air Conditioning. Displayed on a computer screen in real time and allowing system performance to be monitored and settings to be changed quickly and easily, so called smart prisons, can be studied.

2. Ventilation VAV control depending on indoor environment can be introduced to the system and the energy reduction due to it may be involved.

3. Hybrid systems using solar cells, sun heater, etc. can be studied and further energy consumption reduction may be provided.

4. The effect of new building materials and new architecture designs may be studied, since it may offer a good reduction on the conditioning energy.

REFERENCES

[1] John, jarolimek,1977” *Social Studies in Elementary Education”6th education mac Millan publishing companying, Inc.*

[2] Bahel and Sriniv, 2001*” A Computer Study for Estimation of Sensual Energy Consumption and Selection of Air-Conditioner in Six Saudi Arabian Cities”, energy conversion and management and international journal Vol 28 No,3.*

[3] Glenn Reynders Dirk,2004 “*Numerical and Experimental Evaluation of Ventilation in Laboratories a Case Study” international conference 12th on air distribution in rooms, pp (1-9).*

[4] Lamki ,2013 *“Packaged Hermetic Reciprocating Chillers” Trane air conditioning company.*

[5] C.P. Arora,2005*,” Refrigeration and Air Conditioning “Tata MC Grow Hill, second edition.*

[6] Shank. Wang ,2001” *Handbook of Air Conditioning and Refrigeration” MC Grow-Hill, Second edition.*

[7] Simith, P.R 2014 “*The Role of Using the Standards” computer and education journal, Ser.10, NO3.*

[8] W. P Jones, January,2001" *Air Conditioning Engineering Elsevier Science & Technology Books", fifth edition.*

[9] Khalid. K. zhran, June 2014" *The Concept of Energy Conservation in HVAC Systems" Ministry of Petroleum, Egypt.*

[10] Scott.Mc. Millan, PE, member ASHRAE, July,2010" *HVAC for prisons" ASHRAE journals.*

[11] Philip.R. Austin, B.A.E, M.S.A.E,1970" *Design and Operation of Clean Rooms" Business, new publishing company Birmingham, Michigan.*

[12] Mark. Seymour ,2003" *Practical Air Flow Design and Optimization in Clean Rooms Using Mathematical Modeling "England.*

[13] H. B, Awbi,2003" *Ventilation of Buildings" span press, second edition.*

[14] Smith. P.R.and D. Polland,1986 "*The Role of Computer Simulation in Engineering Education" computer and education, Vol.1, NO3.*

[15] Anesa. M. Hassan,1979 "*Using a System Approach to Design the Education "journal of technology and education, third Issue.*

[16] Surrander Naganathan, P.E, member ASHRAE, October 2010 "*HVAC for prisons "ASHRAE journal.*

[17] Grall. G.C.et.al ,1975“*A Computer Simulation and Validation of a Building Heating Ventilating and Air Conditioning System” ASHRAE Transactions, NO 2361, Vol.81 part 1-PP 66-69.*

[18] A. D. Gosman, 1999” *Teaching Computer Aided Design of Fluid Flow and Heat Transfer Engineering “computer and education, Vol.3, PP 111-130.*

[19] Zhu .et.al.and Dongzhao Yuan ,2006 “*Numerical Analysis of a Cold Air Distribution System “the six international conference for enhanced building operations,china,IV-2-3.*

[20] Wei. Cai. XuboYu, and Danjun Wang ,2002” *CFD Simulation of Cold Air Distribution System with Different Supply Patterns”2nd international conference on computer engineering and technology, china (ICCET) 2nd conference.*

[21] Hamilton, J.f and Millev, J.L, 2014 “*A simulation program for modeling and air -conditioning system” ASHRAE Transaction, Part 1, PP 210-222.*

[22] Carbon trust journal ,2016 *“HVAC System: Switch to Energy-Efficient Technologies*” www.Carbontrust.com.

[23] Polack E. RR, 1998 “*Design Optimization of Air Conditioning System “ASHRAE Transactions NO,2511, Vol 84, part 5.*

[24] Silvermen, G.L, 2006 “*Modeling and Optimization of HAVC System Using Network Concept” ASHRAE Transactions, Cl-81-3, NO7, Vol 87, part 4.*

[25] Ramiz M. Saleem, 2009 "*An Optimization Technique for A Process Cooling System Selection" A PhD Thesis submitted to the college of engineering, university of Baghdad.*

[26] Hamilton and QingyanChen,2012" *Comparison of Air Flow and Contaminant Distribution in Rooms with Traditional Displacement Ventilation and Under-Floor Air Distribution Systems" ASHRAE Transections,115(2), PP (300-324).*

[27] Al-Jobori, Ali Salman, 2016 "*Effect of The Base Temperatures on HAVC Power" journal of Military Engineering College, Baghdad, Six Issue.*

[28] FOJ,1985"*ASHRAE Handbook of Fundamentals "American Society of Heating, Refrigeration and Air-Conditioning Engineering.*

[29] Kvenn, Thomas, 1998 "*Thermal Environmental engineering" 3rd edition.*

[30] Jones, W.P, 1998 "*Air Conditioning Applications and Design "Second edition, Edward Arnold publishers, London.*

[31] Bartak McCormack,2010 M,**"** *Experimental and Numerical Study of Local Mean Age of Air* **"***Building simulation PP (770-790).*

[32] Ali.A.AL-Tememi, 2004 "*Reduction of Energy Consumption in Air Conditioner" Master Thesis, submitted to the council of department of Technical education in the university of Technology.*

[33] Stoecker.F. Wilbert, 2011 “*Refrigeration and Air Conditioning” 2nd edition, international student edition, Mc Grow-Hill book company, Network.*

[34]” *Packaged Air Conditioners Roof Top Units* “1984, *carrier international corporation, U.S.A.*

[35] Jennings, Burgess H ,1990*“Enviromental Engineering Analysis and Practice” international and text book company, New York.*

[36] Fisher, DE, and C. Chantrasrisalai,2006 “*Lighting Heat Gain Distribution in Building” report ASHRAE 1282-RP. Atlanta.*

[37] Ranlyu, 2016 “*Low-Energy LEO Lighting Heat Gain Distribution in Buildings” Science and Technology for the built environment 26 October*.

[38] *“Hand Hook of air Conditioning System Design”* 2000, *carrier air conditioning company, McGraw-Hill book company, New York.*

[39] Liping Wang, 2013 *“Monitoring-Based HVAC Commission of an Exiting Office Building for Energy Efficiency” Journal Homepage:WWW.elsevier.com\locate\apenergy*.

[40] US ,2012 “*Environmental protection agency” psychometrics charts*. *Website http:\\www.energystan.govlindex.cfm? c=evaluate-performance. Bus-portfolio manager*.

[41] chadd erton. David, 2011 "*Air Conditioning a Practiced Introduction" 2nd edition, Great Britain by the alden poess, oxford.*

[42] Stoecker, wil pertf,1989 "*ASHRAE Manual Procedures for Simulating the Performance of Competence and System for Energy Calculations".*

[43]" *System Design Manual" ,1999, carrier international corporation, U.S.A.*