PERFORMANCE STUDY BETWEEN VARIABLE REFRIGERANT FLOW AND CHILLED WATER AIR CONDITIONING SYSTEM

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KUALA LUMPUR

2019

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SUBMITTED TO THE FACULTY OF ENGINEERING, UNIVERSITY OF MALAYA IN PARTIAL FULFILMENT OF THE REQUIREMENT FOR THE DEGREE OF MASTER OF MECHANICAL ENGINEERING

2019

UNIVERSITY OF MALAYA
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Name of Degree: Master of Mechanical Engineering

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ABSTRACT

Currently, the world been a witnessed on how the evolution of air conditioning industries takes place. From an idea of centrifugal water chiller by Willis Carrier back in 1921, today we can see how the chiller industry have booming the market. The variety of chiller was introduce and to the latest there are a modular chiller system and a chiller that been installed with a variable speed drive technology in a way to increase the chiller performance itself. On top of that, despite the chiller system, there is another air conditioning system that has captured the attention of this industry. The system known as Variable Refrigerant Flow. The concept of this system was basically connecting a single or modular outdoor system to a number of indoor system. Currently, Hitachi variable refrigerant flow system has taken the lead in producing it most efficient variable refrigerant flow system in the market. The competition among the air condition brand in Malaysia is very competitive since Malaysia has the highest demand in air conditioning units. Due to this fact, the study towards the performance of this two type of system was done. Throughout the study, variable refrigerant flow system has offer the lowest equipment cost to be compared with chiller, but this system generate higher operational cost. This due to instability of the refrigerant charge throughout the system. Using water as a primary source of refrigerant is the best idea since it can be control precisely rather than other refrigerant such as in this case is R410A which is used for variable refrigerant flow system. This also led to a result that chiller give higher coefficient of performance if to be compared with variable refrigerant floe system.

Keywords: VRF, Chiller, Energy, Air Conditioning, VSD

ABSTRAK

Hari ini, dunia telah menyaksikan bagaimana evolusi industri penghawa dingin berlaku. Dari idea penyejuk air empar oleh Willis Carrier pada tahun 1921, kini kita dapat melihat bagaimana industri penyejuk telah berkembang pesat. Pelbagai penyejuk telah diperkenalkan dan terkini terdapat sistem penyejuk modular dan penyejuk yang telah dipasang dengan teknologi pemacu laju berubah-ubah dalam cara untuk meningkatkan prestasi penyejuk itu sendiri. Selain itu, walaupun terdapat sistem penyejuk, terdapat juga sistem penyaman udara lain yang telah menarik perhatian industri. Sistem yang dikenali sebagai aliran penyejuk berubah-ubah. Konsep sistem ini pada dasarnya menghubungkan sistem luaran tunggal atau modular kepada beberapa sistem dalaman. Pada masa ini, sistem aliran penyejuk berubah-ubah dari Hitachi telah memimpin dalam menghasilkan sistem aliran penyejuk pembolehubah yang paling berkesan di pasaran. Persaingan dalam kalangan jenama penyaman udara di Malaysia sangat kompetitif kerana Malaysia mempunyai permintaan yang tinggi dalam unit penyaman udara. Oleh diseabkan itu, kajian terhadap prestasi dua jenis sistem ini telah dilakukan. Sepanjang kajian, sistem aliran penyejuk berubah-ubah telah menawarkan kos peralatan terendah berbanding dengan penyejuk, tetapi sistem ini menjana kos operasi yang lebih tinggi. Ini disebabkan ketidakstabilan caj penyejuk di seluruh sistem. Penggunaan air sebagai sumber penyejuk utama adalah idea terbaik kerana ia boleh mengawal tepat kadran caj dan juga tenaga masuk berbanding penyejuk lain seperti dalam kes ini adalah R410A yang digunakan untuk sistem aliran penyejuk berubah-ubah. Ini juga menyebabkan keputusan bahawa penyejuk memberi pekali prestasi yang lebih tinggi jika dibandingkan dengan sistem aliran penyejuk berubah-ubah.

ACKNOWLEDEGEMENT

Praise to Allah SWT for He is the most Beneficent Merciful. This project would not be a success without the help and the kind guidance of my supervisor, Prof. Dr. Kazi Md Salim Newaz. Thank you for being not only a mentor, but a friend in need. The amount of love and support came from the devoted and most caring person in my life, my *Ummi* (mother), Puan Nor Hashimah binti Ahmad. I would not be where I am without you, and I am most definitely nothing without you. Equally, my beautiful siblings had been my backbone and the reason to keep fighting, especially my lovely Nur Amalin Waheeda, I pray for the best in the world for you.

Anne Frank wrote that "*No one has ever become poor by giving*." A special thanks to my employer, Johnson Control Hitachi Air Conditioning Sales Malaysia Sdn Bhd for not being tired of giving the opportunities and chances to complete this project whilst serving your company. Your kindness can never be repaid, but by His Gracious. It is true what is being said about family; it is not about what is in the blood, but what is in the heart. Thank you to my friends for whom without them, the world never feels as whole.

Knowledge is the source of progress, and progress is what I intended to strive for, now and forever. Highest gratitude to University of Malaya who not only believes in me, but also our generation. The effort made in preparing a greater generation for the future is what keeping this nation moving forward.

I am truly indebted in all of you who had aspired and assisted me in completing this project, both direct and indirect contributions. May His Lordship grant us His blessings in this world and in the hereafter.

TABLE OF CONTENT

LIST OF FIGURES

LIST OF TABLES

LIST OF SYMBOL & ABBREVIATION

Symbol	Description
VRF	Variable refrigerant flow
COP	Coefficient of performance
^{O}C	Degree Celsius
kW	Kilowatt
TR	Tones Refrigerant (Tonnage)
VSD	Variable speed drive
HVAC	Heating, ventilation and cooling
EEV	Electronic expansion valve
DB	Dry bulb
WB	Wet bulb
U	Factor
BF	Ballast factor
AHU	Air handling unit
V	Volts
Ph	Phase
Hz	Hertz (Frequency)
CMH	Cubin meter per hour
PHT	Polytechnics Hulu Terengganu
MDHT	Majlis Daerah Hulu Terengganu
BTU	British thermal unit

LIST OF APPENDICES

CHAPTER 1

INTRODUCTION

1.1 Background of Research

The world today is witnessing how global warming has become one of the issues that borders the whole nation. This global warming is related to the process of increasing the surface temperature of the earth. Through a report released by Intergovernmental Panel on Climate Change (2001) states that there has been an increase in carbon dioxide levels since the last five decades. This has led to the phenomenon of global warming. This phenomenon has also increased the global warming temperature by 0.6 $°C$ and its impact has caused a change in weather to occur in extreme, particularly on the rain and the thunderstorm pattern. Due to this fact, an energy efficient driven type of technologies need to be developed to cater this problem.

In today environmental issue and crisis on the energy consumption, scientists are searching for a solution in inventing an energy efficient system. The utilization of petroleum derivatives such as fossil fuel and the emanations of ozone harming substances that lead to greenhouse effect lead to extensive financial expenses and ecological results. Air conditioning are the biggest consumer in energy; a flip-side of an innovation improvement in heating, ventilation and cooling (HVAC) system is at its urgent state. (Doo et al. 2017).

Variable refrigerant flow (VRF) system change the channel of refrigerant to indoor units dependent on the set up capacity. This system have an ability to control the measure of refrigerant charge that is given to the evaporator unit or indoor unit placed all over the building makes the variable refrigerant system is the most efficient system for a multiple number of zone with variety of load (Carrier, 2013). Figure 1.1 show the zoning layout for VRF System.

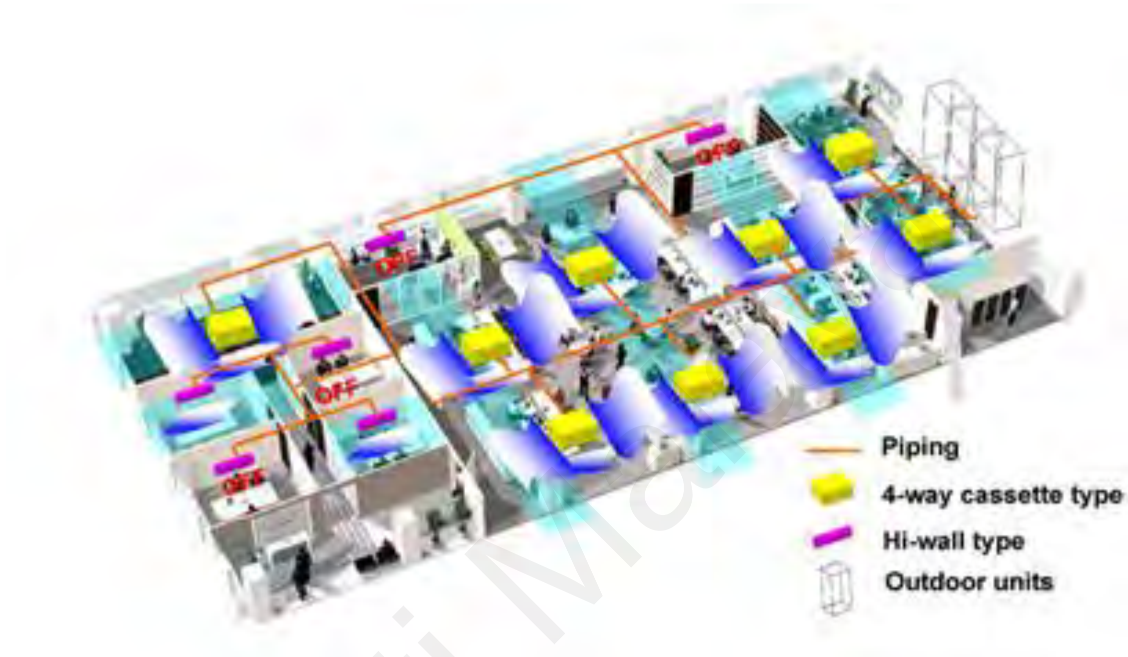


Figure 1.1: Zoning Layout for VRF System.

(Source: Variable Refrigerant Flow (VRF) System, Carrier Corp. New York, 2013)

Chilled water system chillers are classified as a refrigeration system that cools a process fluid or dehumidifies air in commercial and industrial facilities. A chiller will remove heat from a liquid via a vapour-compression or absorption refrigeration cycle. This liquid can then be circulated through a heat exchanger to cool equipment, or another process stream (such as air or process water). As a necessary by product, refrigeration creates waste heat that must be exhausted to ambience, or for greater efficiency, recovered for heating purposes. Chilled water is used to cool and dehumidify air in mid- to large-size commercial, industrial, and institutional facilities. Water chillers can be water-cooled, air-cooled, or evaporative cooled.

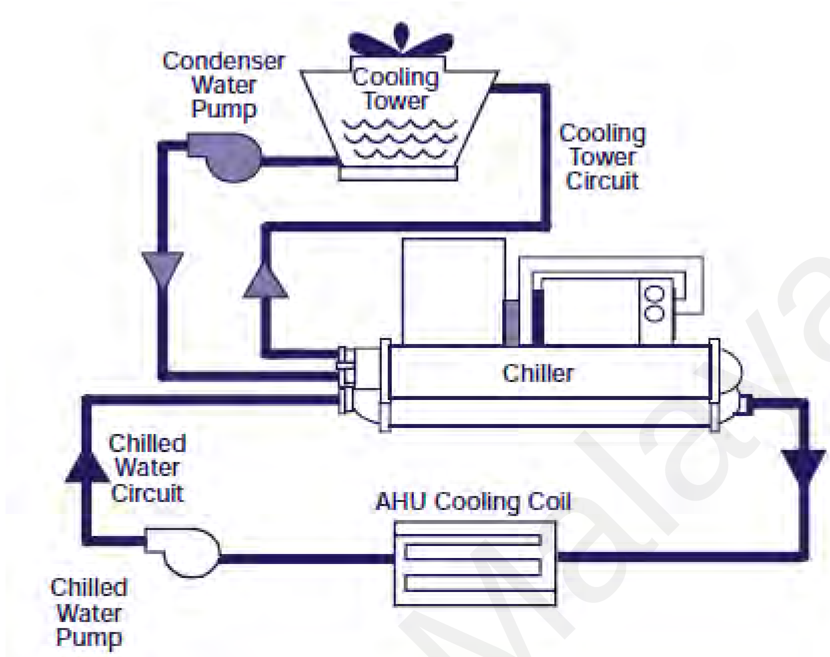


Figure 1.2: Typical water chiller diagram

The basic cooling cycle is the same for both vapour-compression and absorption chillers. Both systems utilize a liquid refrigerant that changes phase to a gas within an evaporator which absorbs heat from the water to be cooled. The refrigerant gas is then compressed to a higher pressure by a compressor or a generator, converted back into a liquid by rejecting heat through a condenser and then expanded to a low- pressure mixture of liquid and vapour that goes back to the evaporator section. The cycle is repeated.

1.2 Problem Statement

The current scenario of today's rising energy consumption has become a major focus on scientists to study what is the most appropriate method for controlling energy consumption and at the same time fulfilling current requirements. This is also supported by a report released by the Energy Commission which shows energy sales increase to 90,770 GWh, an increase of 8.8% when compared to 83,411 GWh in 2009 (Energy Commission, 2010). Malaysia has a strong GDP growth of 4.6% in 2011 which is rare among Asian countries. This actually brought Malaysia far ahead of them.

1.3 Aims

To evaluate the performance of Variable Refrigerant Flow (VRF) and Chilled Water Air Conditioning System in reduction of energy consumption and cost-benefits for economy and socials balance

1.4 Objectives

1. To design and evaluate the Variable Refrigerant Flow and Chilled Water Air Conditioning System.
2. To compare the performance (energy consumption) of both system
3. To evaluate the economic reciprocity and socials readiness.

1.5 Scope

1. This study is mainly for Malaysia, the system that was selected are Variable Refrigerant Flow (VRF) and Chilled Water Air Conditioning System.
2. The performance is compared under the same standard using Malaysia Test Standard and ASHRAE reference.
3. This research strongly hold on 2 platform which is first about the evaluation of two system which is Variable Refrigerant Flow (VRF) and Chilled Water System and their advancement in energy consumption comparison, and second, the potential of this proposed technology improve social balance and reciprocated the economy.
4. The selected building to study the performance of the design system is Polytechnic Hulu Terengganu.
5. The variable refrigerant flow was design by using Hitachi Variable Refrigerant Flow Series. The chilled water system and its air handling unit (AHU) was design using Johnson Control Inc. with York series and also one of Daikin Applied Sdn Bhd software.

CHAPTER 2

LITERATURE REVIEW

2.1 Introduction

In this chapter, five main parts that related to this study will be overview. First, is about the energy consumption behavior in Malaysia, second about the air conditioning case study in the market, third is a deep introduction on chiller system, and lastly is a discussion on the variable refrigerant flow system.

2.2 Energy Consumption in Malaysia

In Malaysia there a quite a number of sector that consume energy such as industries, transportation, agriculture, residential/commercial and etc. Through the study conducted by Shaharudin (1997) and also Siti Zakiah (2004), states that, with the increase of urban population and reduction of vegetation coverage and coupled with the use of air-conditioning in Malaysia have produced a number of heat energy that enhance the discomfort of the city to its inhabitants. This can be refer in Figure 2.1 which shows the state of the increase of global temperature anomalies released by the Meteorological Department.

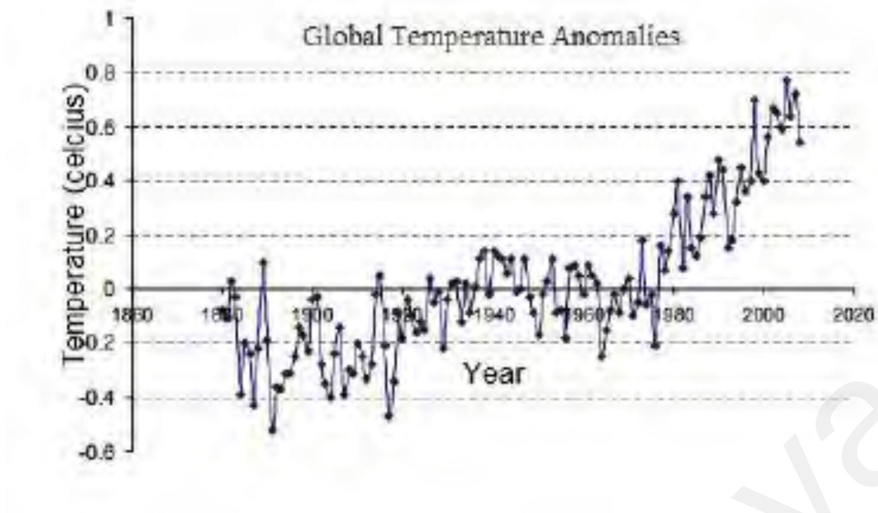


Figure 2.1: Global Temperature Anomalies

(Source: Meteorological Department)

At the same time, referring to 2010 performance and statistics published by the Energy Commission in 2010, states that electricity demand especially in peninsular Malaysia in 2010 increased from 7.8% in 2009 to 94, 748 GWh in 2009 to 102, 139 GWh. In 2010, the maximum demand in peninsular Malaysia recorded an increase of 5.8% from 14, 245 MW in 2009 to 15, 072 MW in 2010 which was recorded on May 24, 2010. This can be refer at figure 2.2. (Energy Commission, 2010)

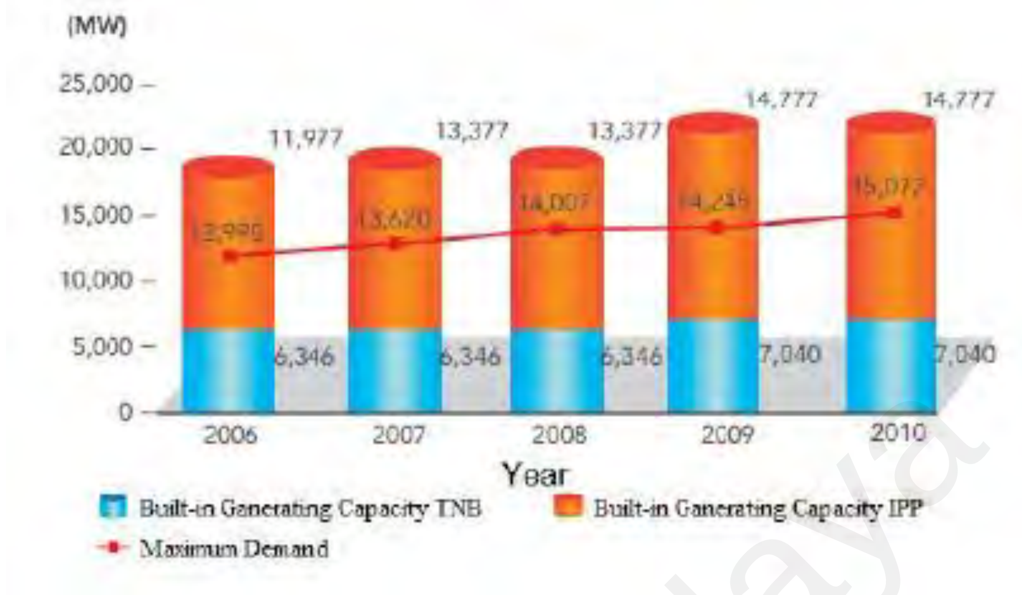


Figure 2.2: Report by Energy Commission, 2010

(Source: Energy Commission Report, 2010)

On the other hand, from *Malaysian National Statistic Handbook* stated several sector had involve in energy usage in Malaysia. This can be refer from figure 2.3. This figure show that industrial sector and transportation sector is the heavy consumer of fuel. Figure 2.4 shows the data of energy consumed by sector in year 1995 and year 2015. In 1995, the final energy consumption was 21,883 ktoe, and up to 2015, the energy consumed was increased up to 51,806 ktoe. (Energy Commission, 2018))

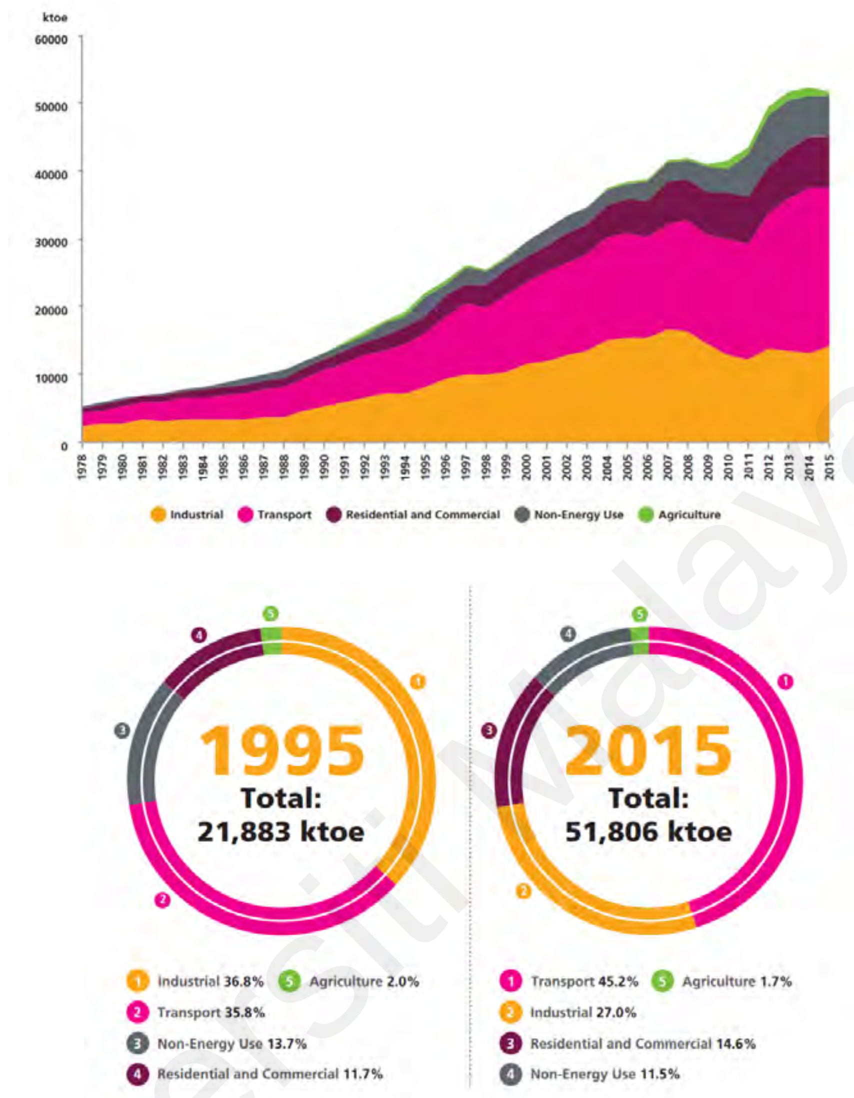


Figure 2.3 & 2.4: Final Energy Consumption by Sector

(Source: National Energy Balance, 2015)

On the other hand, according to the Global Economic, Data, Indicators, Charts & Forecast, (CEIC), Malaysia's electricity information was stay competitive. Malaysia electricity consumption information was accounted for at 12,310.231 kWh/month in February 2019. This records an abatement from the past number of 12,520.361 kWh/month for January 2019. Malaysia's Power Consumption information is review month to month, averaging 6,425.750 kWh/month from January 1989 to February 2019, with 362 perceptions. The information achieved an unequaled high of 12,785.769 kWh/month in August 2018 and a record low of 1,385.300 kWh/month in February 1989. Malaysia's Electricity Consumption information stays dynamic status in CEIC and is accounted for by Department of Statistics. The information is sorted under Global Database's Malaysia. (Department of Statistic, 2019). Figure 2.5 shown the behavior of energy consumption in Malaysia.

Figure 2.5: Electricity Consumption in Malaysia

(Source: Department of Statistic)

In heating, ventilation and cooling (HVAC) frameworks a major commitment to save the energy can be done by the establishment of a component with a capacity to fit the motor speed to the original exigencies of the structure request, which generally throughout the year is altogether lower than the actual design utilized for estimating the system (Carrier, 2005). Plus, the part load can cause wasteful activity during throughout the year. Hence, at this point variable speed drive (VSD) are today most reliable component to shift the rotational speed of the electric motor and thus of the parts with motor driven such as fans, pumps and so on, (ABB, 2011). Parts joined to a variable speed motor, for example, fans, pumps and blowers act contradistinction as their speed changes. In this way VSDs can be connected to chillers, air handlers and heat pumps to seek after increasingly proficient system for a central system. VSD innovation empowers these parts in parallel or autonomous with the building management system. (Hydraulic Institute, Euro Pump and U.S. Department of Energy, 2004)

2.3 Air Conditioning Case Study

Today's air conditioning has become one of the equipment act as a necessity for every home. The growing demand has led to the emergence of various air conditioning brands in the market. Among the brands of air conditioning available on the market are LG, Sharp, Panasonic, Hitachi, Samsung, Elba, Haier, York and Daikin. Each brand of air conditioning offers a variety of functions and features such as antibacterial, energy saving, deodorizer, and more.

Any cases and aspirations for increasingly economical fates in Asia are seriously traded off by the broad and quick take-up of vitality escalated techniques for cooling indoor spaces. Over the coming two decades Asia will be the primary driver of a 40% expansion in worldwide vitality utilization, multiple quarters of which will keep on originating from fossil fuels. (Fernando et. al, 2008)

Where once AC was viewed as an extravagance, in a couple of brief decades it has turned into a profoundly regular innovation for controlling the temperature and humidity levels of indoor spaces all through the locale's tropical and subtropical zones, especially in urban areas. With this pattern set to proceed, in Southeast Asia the mechanical cooling (and drying) of the assembled condition will be a critical factor adding to an interest in vitality that is outpacing a great part of the world, expanding from current levels by around 75% by 2030. (IEA, 2009)

In the UK the heating, cooling and ventilation of indoor spaces is in charge of a critical extent of total vitality request. Of these warming includes by a wide margin the significant part, however the vitality utilized for cooling in non-local structures has expanded, with the quantity of introduced molding frameworks and units developing uniquely since the 1970. (DECC, 2013)

Late history has demonstrated that air conditioning system possession can develop more quickly than monetary development in warm-climate nations. In 1990, not exactly a percent of urban Chinese family units claimed a forced air system; by 2003 this number rose to 62%. The proof proposes a comparative blast of air conditioning system use in numerous other nations isn't a long ways behind. Room climate control system buys in India are presently developing at 20% every year, with about portion of these buys ascribed to the private part. (McNeil & Letschert, 2007)

2.4 Chilled Water System.

In air conditioning industries, chilled water system is by far the oldest system that are still in demand currently. Back in 1921, Willis Carrier is an inventor for the very first centrifugal water chiller. This chiller was then been patented under his name. Previously, before the existing of this centrifugal chiller, the reciprocating type was commonly used to move the gas or refrigerant through the system. The centrifugal compressor is the main part for this centrifugal chiller. Same goes to the reciprocating chiller type which used the reciprocate compressor. This design of the first centrifugal compressor is almost similar to the design of a centrifugal blades in a water pump. In 1950, the explosion of plastic industries has led to the invention of the first industrial chiller system. (John Buie, 2011)

2.4.1 Type of chiller

In general, the chiller system can be divided into two major system which is vapor compression chiller and vapor absorption chiller. The vapor compression chillers use an electrically driven mechanical compressor to force a refrigerant around the system. These are the most common types of chillers. There are two sub categories for vapor compression chillers which are water cooled or air-cooled chillers. On the other hand, the vapor absorption chillers will use a heat source to move the refrigerant around the system rather than using a mechanical compressor. The refrigerant in these chillers move around between areas of different temperature and pressure. (Paul Evans, 2018)

Taking a side at vapor compression chillers, the two fundamental sorts of system are air cooled and water cooled. The two kinds of chillers have a similar basic parts which are, the evaporator, the blower, the condenser and the capillary tube or expansion component. When we talk about air cooled or water cooled, this basically alludes to the manner by which the undesirable warmth is rejected out from the system by means of the chillers condenser. The working rule for both air cooled and water cooled chillers is actually the same. A blower drives a refrigerant round within the chiller between the condenser, expansion component, evaporator and back to the blower. The main contrast is that with an air cooled chiller, fans power air over the uncovered containers of the condenser which reject the heat. Water cooled chillers have a fixed condenser and water is pumped through the system to remove the heat load and scatter this through the cooling tower. The cooling tower will likewise utilize a fan to reject the heat. (Paul Evan, 2018).

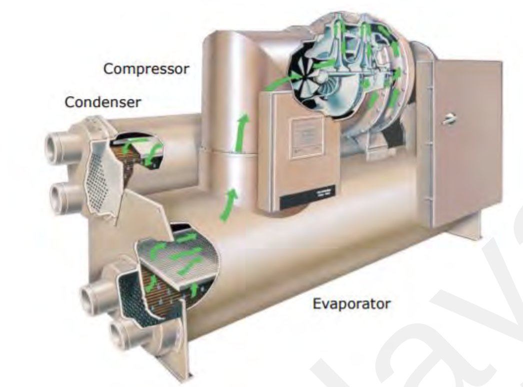


Figure 2.6: Vapor Compression Chiller

(Source: http://www.tranebelgium.com)

There are a variety of water chiller types. Most commonly, they are absorption, centrifugal, helical rotary, and scroll. Some reciprocating chillers are also available. Chillers can be either air- or water-cooled. (Trane, 2009). Table 2.1 shows different type of chiller.

Table 2.1: Type of Chillers

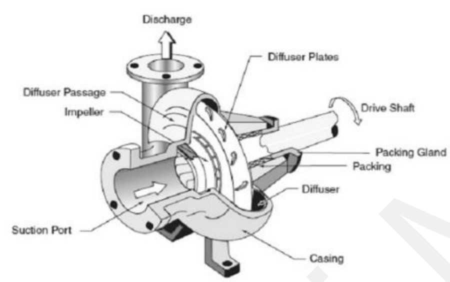
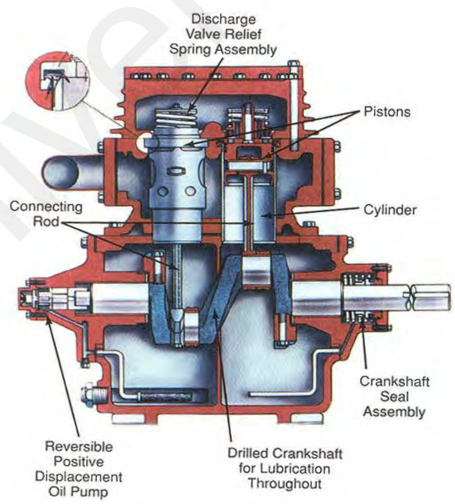
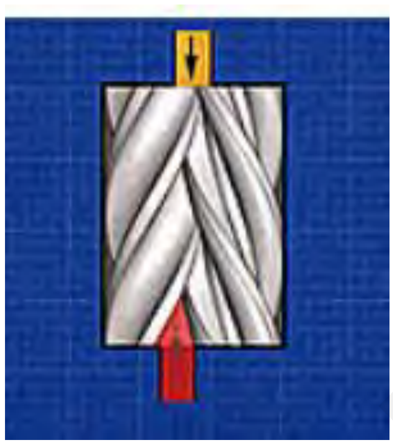
No	Type	Description
1	Centrifugal Chillers  *Figure 2.7: Centrifugal Chiller* *(Source: https://ihc2015.info/skin/centrifugal-compressor-animation.akp)*	This chiller usually available as a water cooled chiller. Often been used in medium to large cooling load with typically available within 150 – 6,000 TR ~ 530 – 21,000 kW. Usually, this system could generate in total coefficient of performance around 5.8 to 7.1. The centrifugal chiller usually used only one compressor, but in certain cases such as for a large capacity system, the number of compressor could increase to two. This system work at best at full loading and the VSD can be fitted to improve the part load. On top of that, this system use one or two rotating impeller to compress the refrigerant and force it around the chiller.
2	Turbocor Chiller	This Type of chiller can be operates as in air cooled system or water cooled system. It used in all cooling loads from large to small application with a normal

	 *Figure 2.8: Turbocor Chiller* *(Source: https://theengineeringmindset.com/chiller-types-and-application-guide/)*	range of 60 – 1500 TR~ 210 – 5,200 kW in it cooling capacity. The generated coefficient of performance of this system usually start from 4.6 up to 10. One or more compressor is use, while the stage and speed of the system is varied. This system also consist of variable speed controller, soft starter, magnetic bearings, plus there is only one moving part and it is an oil free system. This type of chiller also used two rotating impellors to compress the refrigerant.
3	Reciprocating Chiller  *Figure 2.9: Reciprocating Chiller*	The medium used for this type of chiller usually air or water cooled chillers. This system is an old technology and seldom been used nowadays. This chiller was use in small to medium cooling load, but commonly used in simple low-cost refrigerator. The available load capacity usually start from 50 – 500 TR ~ 170 – 1,700 kW. The rated coefficient of performance generally at 4.2 to 5.5. On top of that, this system used a piston and

	(Source: https://berg-group.com/engineered-solutions/the-science-behind-refrigeration/)	also a chamber to compress the refrigerant, plus the capacity control through the compressor staging.
4	Scroll Chiller  *Figure 2.10: Scroll Chiller* (Source: https://theengineeringmindset.com/chiller-types-and-application-guide/)	In general, scroll chiller can be design with air cooled system of water cooled system. Usually, this system was used for a small to medium cooling loads with a typical load from 40 – 400 TR ~ 140 – 1,400 kW. The air cooled scroll chiller usually rate the coefficient of performance reading from 3.2 to 4.86, while the water cooled scroll chiller, the rated performance was 4.45 to 6.2. This system could use one or more than one compressor, with either a fixer or variable speed, staged controlled. Two spiral plates was used to compress the refrigerant, where one of it will be fixed in place while the other one will rotate. The capacity if this type of chiller is controlled by momentarily separating scrolls with solenoid valve and also an electronic modulation.

5	Screw Chiller  *Figure 2.11: Screw Chiller* *(Source: https://theengineeringmindset.com/chiller-types-and-application-guide/)*	Air cooled screw chiller and water cooled screw chiller are the option of system offered by this type of chiller. Typically, it used in small to medium cooling loads with a range of 70 – 600 TR ~ 250 – 2,100 kW. For an air cooled type, the coefficient of performance rated by this system is around 2.9 – 4.15, while for water cooled system, the performance rated at 4.7 to 6.07. Usually, this system will just use one compressor as for water cooled type, while for air cooled type, usually this system will used one or two compressor. This system use two interlocking rotating helical rotors to compress the refrigerant, and the capacity of the system is controlled by speed control.

2.4.2 Chilled water primary system components

In general, chilled water system consist of several functional equipment or part. In a way to cool the water, chiller is one of the major equipment needed for this chilled water system. Next is the coil which will justify the load carry by the system. To send the chilled water to satisfy the load, a distribution pumps and pipes of chilled water was used. On the other hand, on the condenser part, the cooling tower or condenser fan (air cooled water chiller system), condenser water pumps and pipes was used to reject the heat to ambient. Lastly is a control system that will coordinate the mechanical component together as a system. (Trane, 2009)

2.5 Variable Refrigerant Flow (VRF) System

VRF innovation is one of a kind for its capability to screen each cooling zone temperature by fluctuating the mass flow rate of the refrigerant charge circulated in the indoor unit of that zone. The adjustment in this flow rate is a consequence of concurrent variability in blower and fan speeds. On the other side, chillers work by circulating the chilled water into the indoor zones, which then chill off the indoor air achieving the desired temperature. On top of that, ordinary split air conditioning system allot a different condenser (open air unit) for each evaporator (indoor unit), since they do not have the benefit of connecting more than one evaporator to a condenser system. Truth be told, VRFs are believed to be progressed and inventive type of the regular split cooling system. (Goetzler, 2007)

Lately, the variable refrigerant flow (VRF) system have been generally utilized in both business and private structures because of its various favorable circumstances, including high part load productivity, different indoor units with various limits and setups, individual and adaptable zone warm controls, basic development for establishment and extraordinary capacity to give synchronous warming and cooling in independent zones, (Aynur, 2010), (Lin & Lee et. al, 2015). It has come to more than 100 million units with a comparing estimation of US $70 billion around the world in 2016 (Cooling Post, 2017). In addition, the VRF had a piece of the pie of 53.4% of market share in the China central cooling system (Mechanical and Electrical Information, 2017)

Given the unprecedented performance and adaptable zone level controls, the variable refrigerant flow system have risen as an extraordinary answer for applications requiring individualized cooling configuration. Thus, the variable refrigerant flow system have increased much attention and are ending up more broadly utilized with deals blasting around the world. (Karanukaran et al. 2010)

On top of that, as the variable refrigerant flow system has extended into both new development and retrofit in structures, since it is being seen as a system that gives a great arrangement than the different sorts of system that being considered. As an effective reliable system of heating, ventilation and cooling (HVAC) system, a large number of study about this system configuration were done especially on improving the variable refrigerant control presentation and advancing its application. (Zahi and Abu O Lu et al. 2017)

2.5.1 Type of variable refrigerant flow system.

Variable Refrigerant Flow (VRF) system are refrigerant based system , which are for the most part involved a single condensing unit tp serve different or a numbers of indoor units associated by a refrigerant channeling system. Typically, there are two standard variable refrigerant flow types, which is the heat recovery type and also a heat pump type. The heat recovery type of VRF system essentially can supply heating and cooling at the same time, while for feat pump variable refrigerant flow type, it just can operate both heating and cooling in a time. Therefore, the variable refrigerant flow can be divided into two type of system which either through air source or by water source which highly dependent on the cooling method approach at the condenser unit. The variable refrigerant flow system changes the refrigerant stream utilizing variable speed blowers in the condensing unit and electronic expansion valves (EEVs) attached in each indoor unit. In addition, the advanced variable refrigerant flow system can adjust the temperature on the evaporator side to meet the required load of indoor units (Yun et al. 2016)

On the other hand, the potentiality of variable refrigerant flow system to control the refrigerant mass flow rate as indicated by cooling load or heating has empowered the coordination of upwards of 60 indoor units (evaporator unit) with assorted capacity connecting to an outdoor system with either one or more than one compressor installed at the condensing unit. This opened the likelihood of the zone level individual control, with synchronous cooling and heating in various zones, and warmth recuperation from one zone to another. (Chua et al. 2013). Figure 2.12 shows a typical installation of a variable refrigerant floe system for an office floor. Basically the system configuration will have one outdoor system connect with multiple indoor unit. Usually the indoor unit offered come with a variety type such as wall mounted, ceiling cassette, ceiling exposed and ceiling concealed. On the other hand, figure 2.13 show cycle of a variable refrigerant flow system. Basically the cycle of variable is the same as the single split unit system, just that for VRF the number of indoor are different.

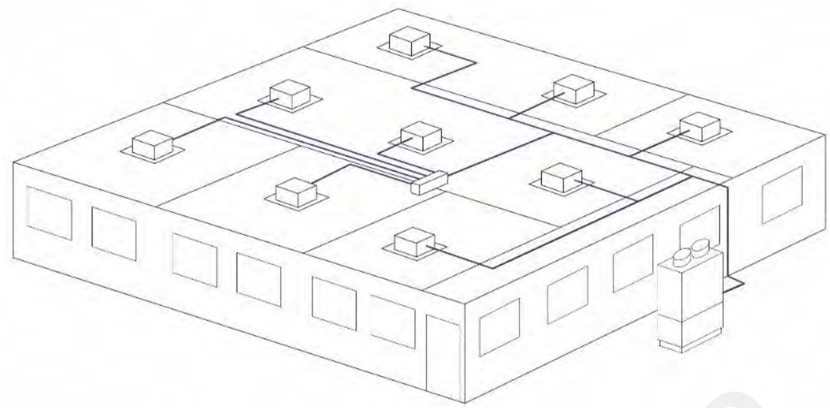


Figure 2.12: Typical Installation of VRF System

(Source: Goetzler W. Variable refrigerant flow systems. ASHRAE Journal. April, 2007)

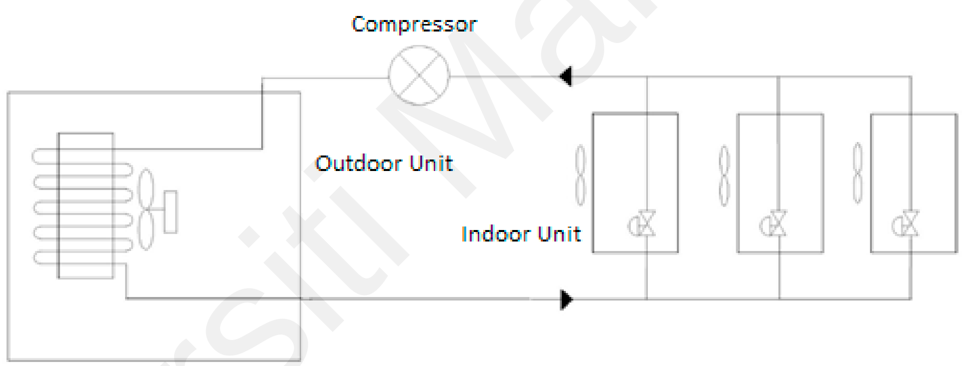


Figure 2.13: Refrigerant Cycle of VRF System

(Source: Goetzler W. Variable refrigerant flow systems. ASHRAE Journal. April, 2007)

CHAPTER 3

METHODOLOGY

3.1 Introduction

This chapter will discussed the methods used in completing this research project. Basically, there are several parts in this research need to be done as below.

1. Building load estimation
2. Variable refrigerant flow system design
3. Chilled water air conditioning system design
4. Performance evaluation of variable refrigerant flow system and chilled water air conditioning system.
5. The operational cost analysis of each system
6. The payback period analysis for chilled water system.

Hence, there are six item with different method will be apply on this research.

3.2 Building Load Estimation

Polytechnics Hulu Terengganu has been selected as the study area for this project. This project was estimated to fill up 75,000 ft^2 of a free space at Kampung Buluh, 21700 Kuala Berang, Terengganu. This project is located at 5.0703° N, 103.0102° E. Thus several condition is estimating the load can be refer from table 3.1. The calculation will be done using the standard load estimation sheet by proposed through the Training Manual from *Technomedia Solution Private Ltd*. (Technomedia, 2019). The precondition in estimating the load can be seen from table 3.1. On top of that, Figure 3.1 shows the load estimation sheet used in estimating the load of an area of Polytechnic Hulu Terengganu.

Table 3.1: Parameter for Load Calculation

Location	5.0703° N, 103.0102° E
Moisture Content Condition	Outdoor Air Condition : 74.7 GR/LB Indoor Air Condition : 71.5 GR/LB
Outdoor Air Inlet Temperature Condition	35 ºC DB
Indoor Air Inlet Temperature Condition	27 ºC DB / 19.5 ºC WB
Relative Humidity of Room	55 %
Solar Gain Glass Factor	0.56
Solar & Trans Gain Walls & Roof	U value : 0.36 Btu/h. ft^2
Ceiling Trans, Gain	U Value : 0.32
Outdoor Air Infiltration (Sensible)	Ballast Factor : 0.15 Factor : 1.08
Outdoor Air Infiltration (Latent)	Ballast Factor : 0.15 Factor : 0.68
Internal Heat Gain Factor	People (sensible) : 245 People (latent) : 205 Light (Ballast factor) : 1.10 Light and other equipment load (factor) : 3.41

BTU ESTIMATION SHEET

JOB NAME:	**Politeknik Hulu Terengganu**						Date: 16/5/2019
SPACE FOR:	**SCU5-01 Group Study 6**			HT.(Ft.)	9.8		Estimated By: Nurshafiq
SIZE:	Width		ft				
	Length		ft				
	Total	**0.00**	**Sq ft**	**0**	**Cu ft**	Estimate for:	**SUMMER**

SOLAR GAIN GLASS				**HEAT GAIN**
ITEM	Area	Sun Gain	Factor	**Btu/hour**
	(Sq ft)	(Btu/h.sq ft)		
N - Glass	0	14	0.56	0
NE - Glass	0	12	0.56	0
E - Glass	0	12	0.56	0
SE - Glass	0	12	0.56	0
S - Glass	0	12	0.56	0
SW - Glass	0	100	0.56	0
W - Glass	0	164	0.56	0
NW - Glass	0.00	123	0.56	0
Sky light(NW)				0
SOLAR & TRANS. GAIN WALLS & ROOF				
ITEM	Area	Eq. temp. diff.	U	
	(Sq ft)	(°F)	(Btu/h.sq ft)	
N - Wall	0	25	0.36	0
NE - Wall	0	31	0.36	0
E - Wall	0	39	0.36	0
SE - Wall	0	39	0.36	0
S - Wall	0	37	0.36	0
SW - Wall	0	35	0.36	0
W - Wall	0	33	0.36	0
NW - Wall	0.00	27	0.36	0
Roof Sun	0	55	0.12	0
Roof Shaded				0
TRANS. GAIN EXCEPT WALLS & ROOF				
ITEM	Area	Temp. diff.	U	
	(Sq ft)	(°F)	(Btu/h.sq ft)	
All Glass	0	14.4	1.13	0
Partition	0	9.4	0.32	0
Ceiling	0	9.4	0.32	0
Floor	0	9.4	0.40	0
INFILTRATION AND OUTSIDE AIR				
nflatration CFM	°F	BF	FACTOR	
0	14	-	1.08	0
utdoor Air CFI	°F	BF	FACTOR	
0	14	0.15	1.08	0
INTERNAL HEAT GAIN				
People	0	Nos x	245	0
Light	0	1.10	3.41	0
Eq. Load	0	W x	3.41	0
		ROOM SENSIBLE HEAT (RSH)		0
Supply duct heat gain +	Supply duct leak. loss +	Heat gain from fan HP(%)	0.0%	0
		Safety factor (%)	5%	0
		EFFECTIVE ROOM SENSIBLE HEAT (ERSH)		0
LATENT HEAT				
People	0	Nos X	205	0
		ROOM LATENT HEAT (RLH)		0
Supply duct leakage loss +		Safety factor %	7.5%	0
nflatration CFM	GR/LB	BF	FACTOR	
0	3.2	-	0.68	0
utdoor Air CFI	GR/LB	BF	FACTOR	
0	3.2	0.15	0.68	0
		EFFECTIVE ROOM LATENT HEAT (ERLH)		0
		EFFECTIVE ROOM TOTAL HEAT (ERTH)		0
OUTSIDE AIR HEAT (SENSIBLE)				
CFM	°F	1 - BF	FACTOR	
0	14	0.85	1.08	0
OUTSIDE AIR HEAT (LATENT)				
CFM	GR/LB	1 - BF	FACTOR	
0	3.2	0.85	0.68	0
		HEAT SUB TOTAL		0
Return duct heat gain & leak. loss +	HP Pump +	Dehum. & Pipe loss (%)	3%	0
TR	**0.00**	**GRAND TOTAL HEAT**		0

CONDITION	**DB (°F)**	**WB (°F)**	**% RH**	**DP (°F)**	**GR/LB**
OUTSIDE	95	75.2			74.7
ROOM	80.6	66.2	55		71.5
DIFFERENCE	**14**	XXXX	XXXX	XXXX	**3.2**

OUTSIDE AIR (VENTILATION)			
25	CFM/Person	CFM VENTILATION	0
0.06	CFM/Sq. Ft	CFM VENTILATION	0
6	ACPH	CFM VENTILATION	0
		REQUIRED VENTILATION CFM	**0**

EFF. SENSIBLE HEAT FACTOR (ESHF) =		#DIV/0!
Indicated ADP =	54 °F	
Selected ADP =	51 °F	
Dehum. temp rise =	25.16 °F	
DEHUMIDIFIED CFM =	**0**	

NOTES

Occupancy =		Persons
Light =	1.0	Watt/Sqft
Eq. Load =	0.0	KW
Required Fresh ai	300	CFM

CHECK FIGURES

CFM/Sq. ft. =	#DIV/0!
Sq. Ft./Ton =	#DIV/0!
CFM/Ton =	#DIV/0!

Figure 3.1: Load Estimation Sheet

3.3 Variable Refrigerant Flow (VRF) System Design

The variable refrigerant flow system design was conduct after the building load estimation is done. The design was done through '*Global VRF Selection Software by Hitachi-Johnson Control Air Conditioning Inc. version 4.6.0*' and the steps as below.

Step 1: Create a project – In this step, we need to choose the region and also the equipment brands. As per figure 3.2, the region selected was Malaysia and the brands of equipment is Hitachi. Then it will get us to the new project page as in figure 3.3. Then the name of the project is created inside this page.

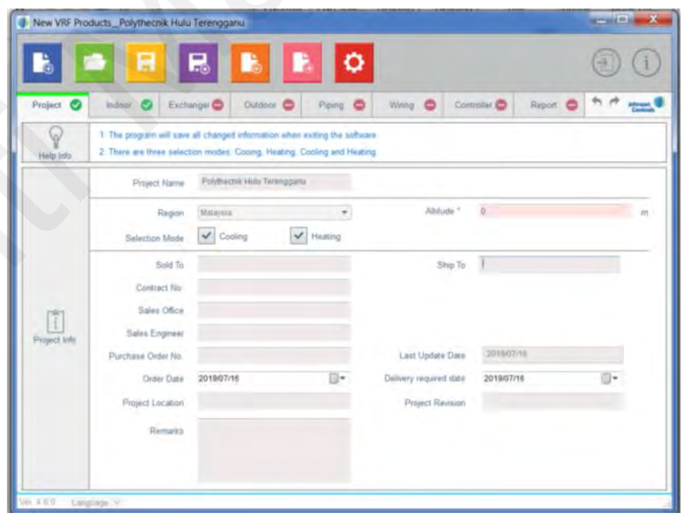


Figure 3.2: Region & Brands

Figure 3.3: Project Page Information

Step 2: Variable Refrigerant Flow Indoor Selection – Hitachi have quite a number of series offer for VRF in industries such as heat pump type (HNCQ), cooling only type (CNCQ), residential type (HNRQ) and etc. Figure 3.4 shows the page of indoor selection. The CNCQ series has been selected for this research since it suit to Malaysia condition that only need cooling supply. Then the indoor unit selection is done by selecting the indoor that offered by Hitachi with a capacity matched to the building load cooling load demand.

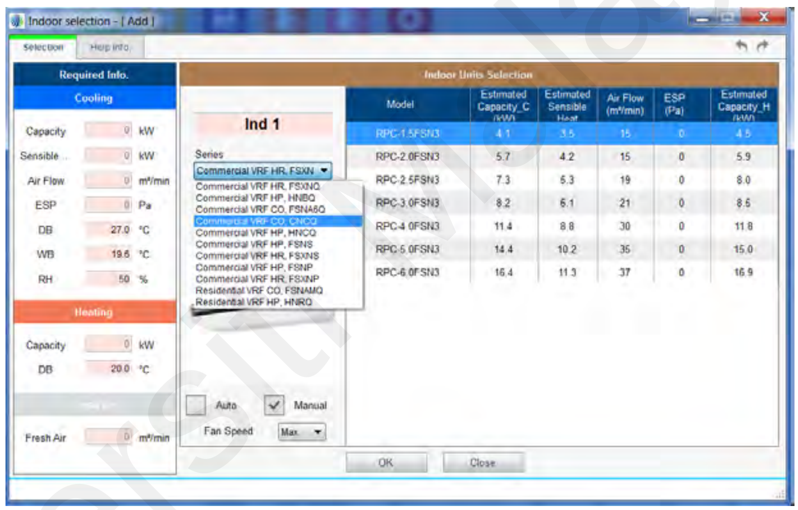


Figure 3.4: Selection of Indoor Unit for VRF System

Step 3: Variable Refrigerant Flow Outdoor Selection – As the indoor equipment has complete its selection process, the outdoor or condensing part for VRF system was then selected. Figure 3.5 show the selection page for outdoor/condensing unit for VRF system in the software. The outdoor model will automatically suggested by the software depend on the indoor capacity of a system.

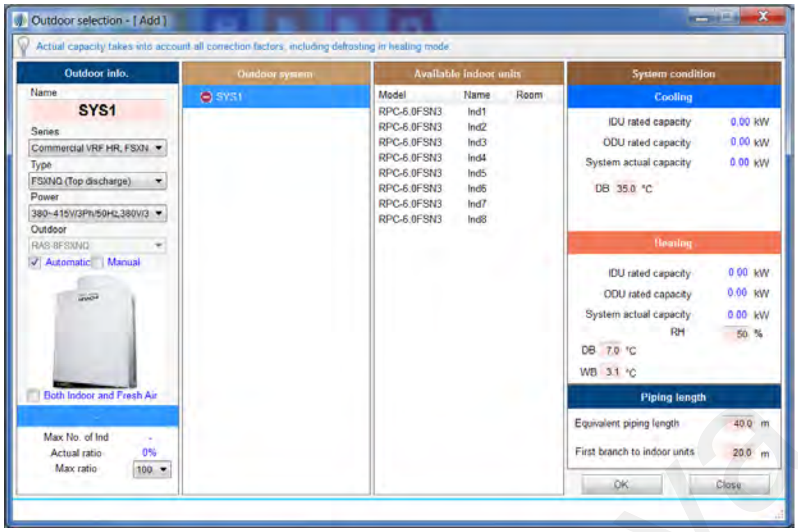


Figure 3.5: Selection of Outdoor Unit for VRF System

Step 4: Refrigerant Piping Design – As all the indoor and outdoor equipment have been selected, from the software itself, we can design the piping based on the floor plan of the building. Figure 3.6 shows the page of a piping design. Basically, from this piping design we can identify the piping size of the system at each joint.

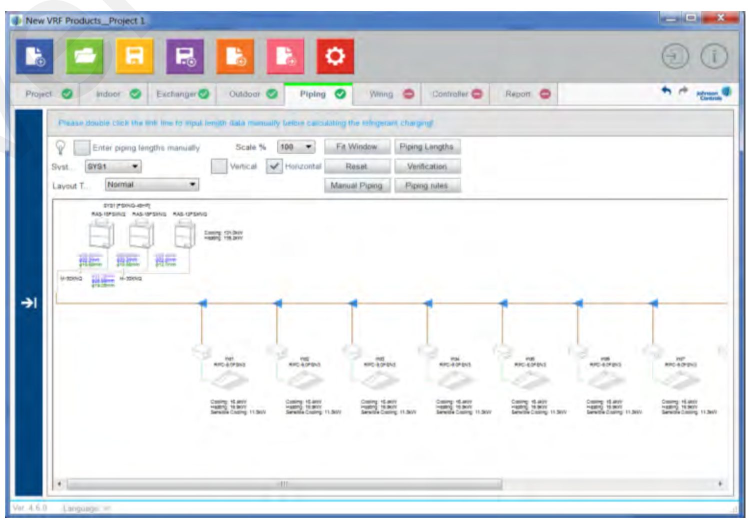


Figure 3.6: Piping Design for VRF System

Step 5: Report: After all the piping design been verified, the report can also be generate throughout this system. Figure 3.7 show the page to generate a report of a system. Basically this report will include the summary of the selected equipment, refrigerant piping design, wiring diagram of VRF system and performance report of the whole system. Thus, this performance was then been used to discussed further for this project research. The VRF system was design at 35°C outdoor condition and 27 °C (DB)/ 19 °C (WB) indoor condition.

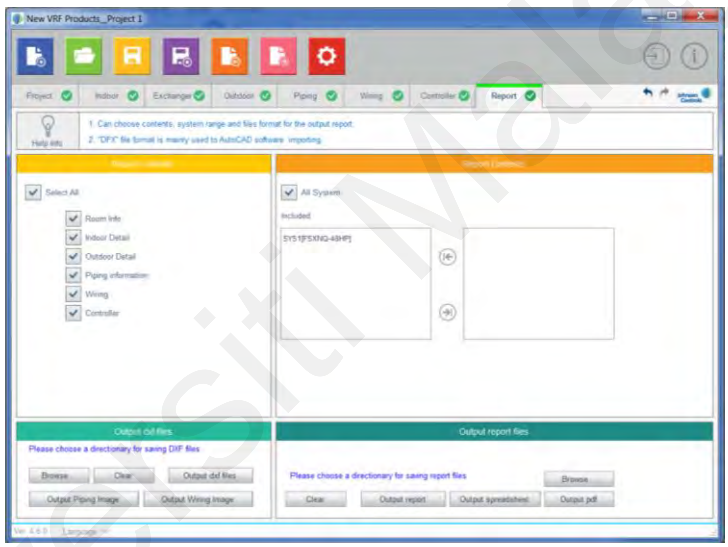


Figure 3.7: Generate Report for VRF System

3.4 Chilled Water Air Conditioning System Design

Chilled water system is another system that will be study in this project research. Before this chilled water is design, the pre-requisite is the same as variable refrigerant flow system which is, we need to know the total cooling capacity demand by the building. To make this chiller complete, we need to take into account the air handling unit of the system. Because in general, this air handling unit will be react as a fan coil unit for chilled water system.

3.4.1 Air Handling Unit (AHU) Design

The air handling unit was design using '*AECworks Selection Software version 8.41 by Johnson Control Inc.*'. Figure 3.8 show the first step in designing the AHU through this software. After the software is open, the AHU will be design with *NewYSM Air Handling Unit* as in a thread from the figure. Next, figure 3.9 will appear to verify the New YSM series was chosen. By clicking the icon, the blank workspace as in figure 3.10 will appear. By click at the new tab on the task bar, figure 3.11 will appear. This is where the precondition of the AHU cabinet was design. For this research, I've used single tier AHU with a condition of 380-415 V/3 Ph. / 50 Hz for all the AHU.

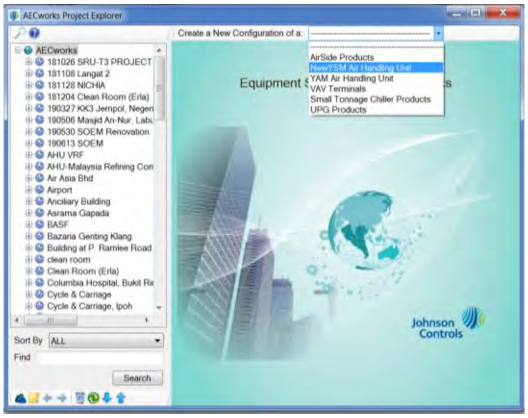


Figure 3.8: AECworks Series selection

Figure 3.9: New YSM Air Handling Unit

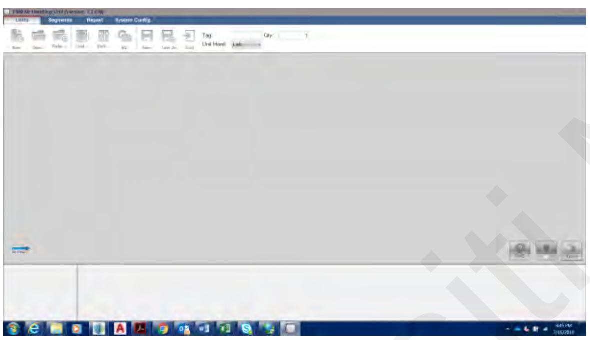


Figure 3.10: NewYSM Workspace

Figure 3.11: AHU Precondition Design.

After the cabinet has been fixed, the unit model of an AHU was design. Figure 3.12 shows the information that need to key in. First is the airflow. The airflow design in 300 CMH per tonnage of the system. The maximum face velocity should always be kept below 2.50 m/s. The inner skin and outer skin material for the whole AHU is design with galvanized steel and painted GI with an inner and outer skin thickness of 0.5mm and Colobond-0.5 mm. The insulation used is 25mm PUF. After all the information has been filled up, and the calculator button was clicked, the software will automatically give a suggestion of suitable cabinet size to use for an AHU.

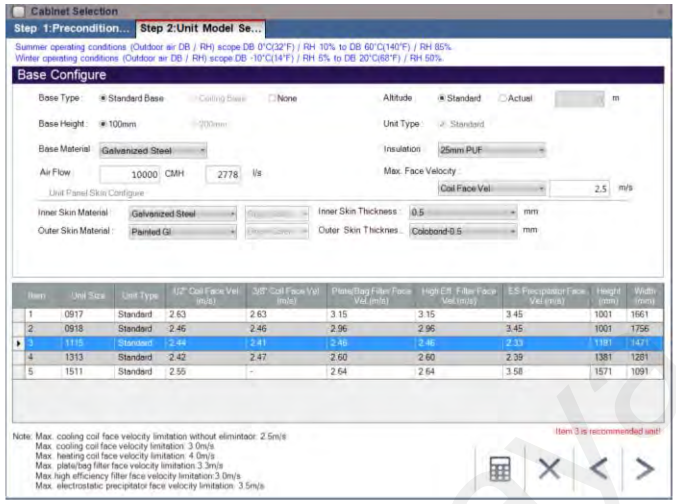


Figure 3.12: AHU Cabinet Selection

After the cabinet is selected, then the process of design the part of the AHU begin. Basically, four main part in AHU will be selected which is first the mixing box (MB), plate filter (PF) or also known a single stage filter, cooling coil (CC) and supply fan. The configuration of the parts can be refer from figure 3.13. The important past to design for AHU is its cooling coil and supply fan. There is several information or parameter that we need to kept constant in designing the cooling coil. Those parameter can be refer from table 3.2.

Table 3.2: AHU Cooling Coil Parameter

The entering air temperature	27 ºC DB / 19.5 ºC WB
The entering relative humidity	48% ~ 60%
Leaving air temperature	13 ºC DB
Entering water temperature	7 ºC
Leaving water temperature	12 ºC
Maximum water pressure drop	60 kPa

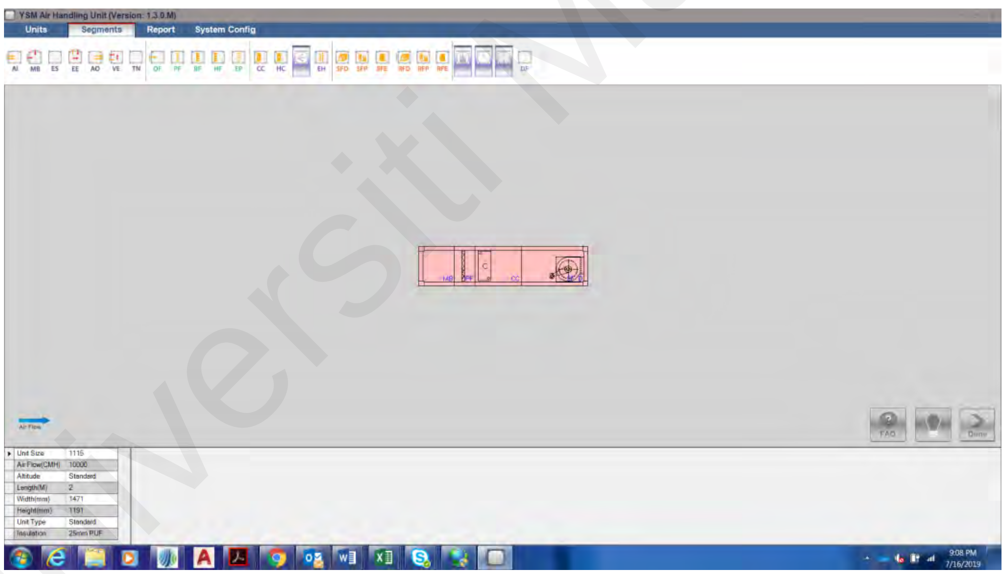


Figure 3.13: AHU Parts Configuration

As for supply fan, the parameter that was taken into account is the external static pressure. For a unit of AHU with more the 100 kW cooling capacity required, the ESP estimated will be 2 in.wg ~ 570 Pa, while if the system below 100 kW, the external static pressure design will be around 1 in.wg ~ 278.78 Pa. The fan type selected for whole AHU is backward curve. On the other hand, the filter used is G3 type. After all the parameter and parts was selected, the report can be generate by clicking the report tab on the task bar. Basically several report we can obtain from this software for the AHU system such as fan report, performance report, drawing and noise report. Figure 3.14 show the possible report that can be generated from this software.

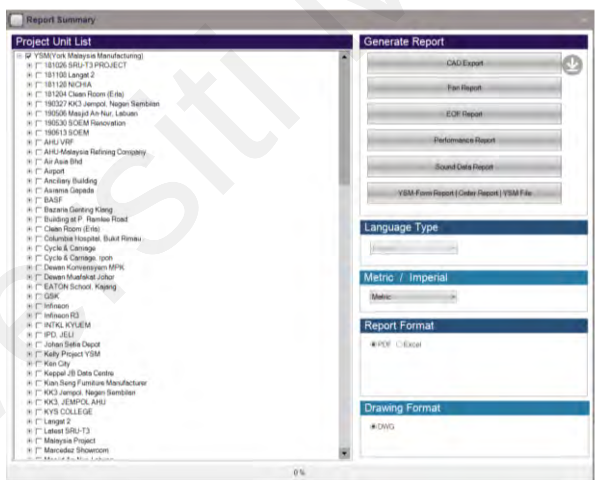


Figure 3.14: Report for AHU

3.4.2 Chiller System Design

The chiller was also design by using the same software as air handling unit which is *AECworks Selection software version 8.41 by Johnson Control Inc.* The different is this time we need to select *Small Tonnage Chiller Product* rather than *NewYSM Air Handling Unit*. This thread can be refer from figure 3.15. For this project two type of chiller has been choose from this software which is the variable water cooled screw chiller system (YVWH) and standard water cooled screw chiller (YGWH) which can be refer from figure 3.16. On the other hand, an air cooled screw chiller was also design but using Daikin chiller selection software. The parameter for chiller design can be refer from table 3.3.

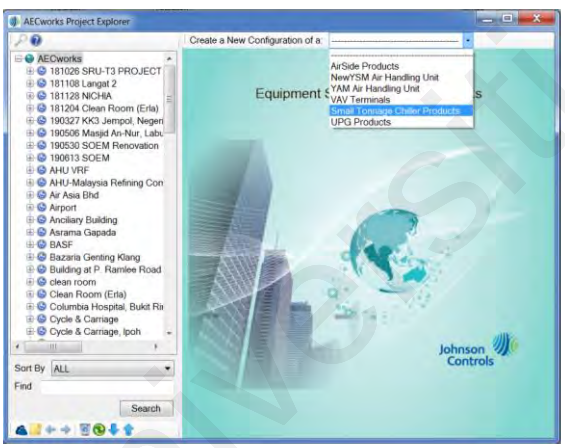


Figure 3.15: Chiller Selection

Figure 3.16: Chiller Model

Table 3.3: Parameter for Chiller Design

Evaporator Leaving Water Temperature	44 ^oF
Condenser Leaving Water Temperature	85 ^oF ~ 90 ^oF
Evaporator Fouling Factor	0.000100
Condenser Fouling Factor	0.000250
Fluid Type	Water
Total Tonnage requires for Chiller	150 Tons
Refrigerant used	R134A

3.5 Performance Evaluation

The performance evaluation is done by determine the coefficient of performance of the system. The formula in calculating the coefficient of performance of a system as below.

COP = Total Cooling Capacity (kW)/ Input Power (kW)

Each system will be evaluate under different part load percentage or data and been compare to identified at which condition will the system perform well.

3.6 Operational Cost Analysis

Block Tariff (per month)	Unit (kWh) allocated based on Prorate Factor	Rate (RM)	Amount (RM)
200	200 x 1.26667 = 253	0.218	55.15
100	100 x 1.26667 = 127	0.334	42.42
300	300 x 1.26667 = 380	0.516	196.08
300	1082 - 253 - 127 - 380 = 322	0.546	175.81
Total	1082		469.46

Figure 3.17: Block Tariff Calculation based on Prorate Factor

Figure 3.17 show the block tariff calculation based on the prorate factor by *Tenaga Nasional Berhad*. The data above was then used in calculating the operational cost of a system. The power input is the key player of determining the rate of the operational cost for a system.

3.7 Payback Period.

The three type of chiller unit are used to compare with a VRF system to obtain the payback period. The formula as per below.

Annual Savings (MYR) = Annual Energy Cost of Standard Unit (VRF operational cost) – Annual Energy Cost of High Efficiency Unit (chiller system)

Payback Period = First Cost (MYR) / Annual Savings (MYR)

CHAPTER 4

RESULTS AND DISCUSSION

4.1 Building Analysis

Polytechnic Hulu Terengganu (PHT) is an institution of polytechnic which owned by the Ministry of High Education Malaysia. Their operation started back in 2008 at Polytechnic Sultan Mizan Zainal Abidin, Dungun which offers 2 main programs; Diploma in Accounting and Diploma in Tourism and Hospitality. In 2010, PHT had move to Hulu Terengganu district, renting a building owned by Majlis Daerah Hulu Terengganu (MDHT) which is located at Bandar Kuala Berang. After several year of operating at the Majlis Daerah Hulu Terengganu building, an initiatives have been done by the government to build a proper building for this institution.

This on-going project align with the mission of PHT to provide recognized and high quality technical and vocational trainings to students around Malaysia. The new construction will take place on the empty 75000 sqft land next to the currently rented building at Kuala Berang and expected to accommodate 250 students at one time.

Figure 4.1: Proposed Location for Construction

(Source: Google Maps)

Polytechnic Hulu Terengganu consist of 4 stories building. The floor plan of this building can be refer from figure 4.2 up to figure 4.5. Currently, this building was designed using variable refrigerant flow (VRF). The analyzing has been done on this building, and I have come out with six different system based on the location and also some limitation of equipment such as below.

1. The piping length should not exceed 90 m.
2. The capacity lost throughout piping.
3. The pressure of supply and return of air and refrigerant.
4. The temperature of supply and return of air and refrigerant.

This six different system was then also been design for six different air handling unit (AHU) that will be used in chilled water system. There are two type of chilled water system design in this project to be compared with variable refrigerant flow system which is air cooled chiller which used air as a cooling medium for its condenser unit and also water cooled chiller which used water as a cooling medium for its condenser unit.

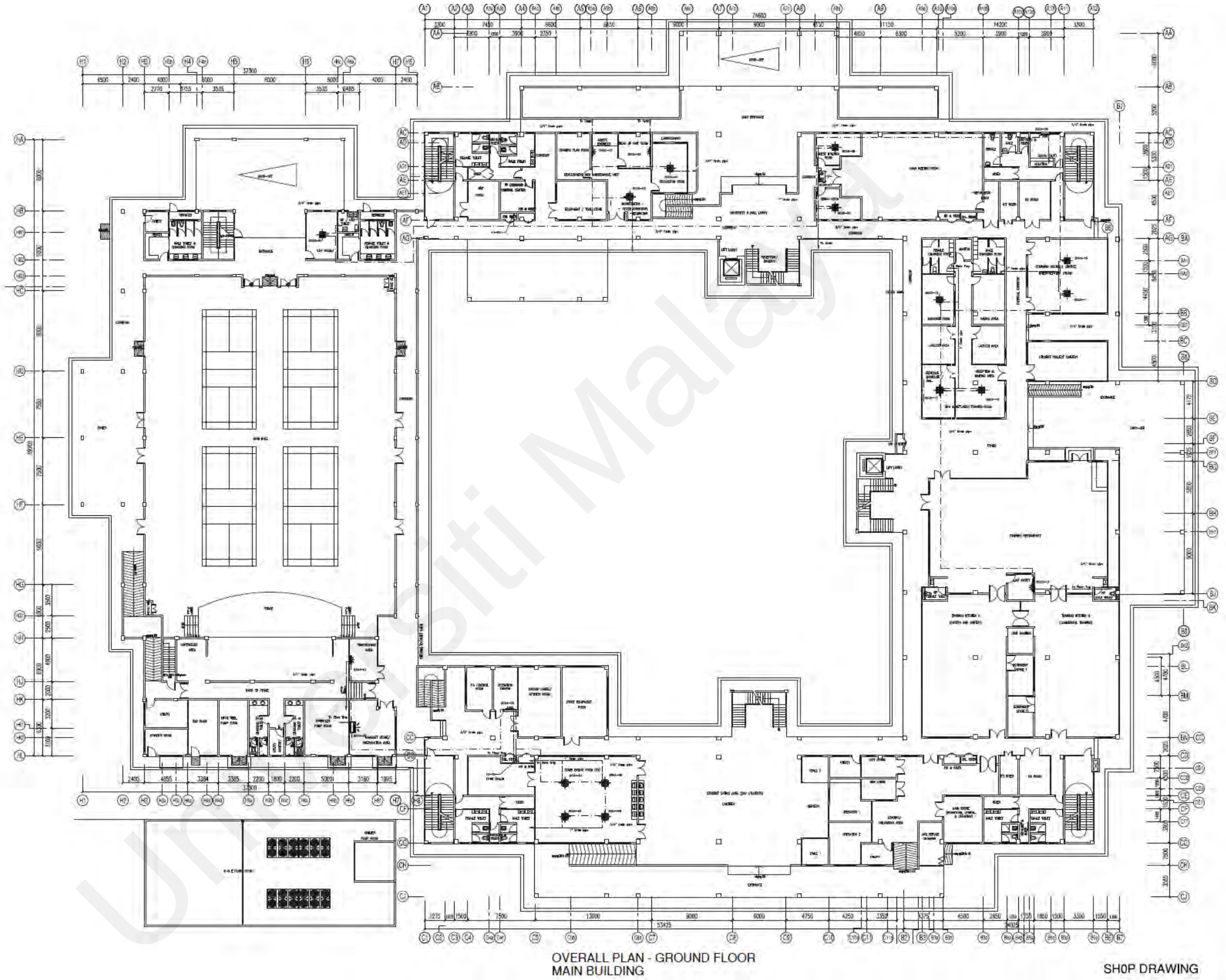


Figure 4.2

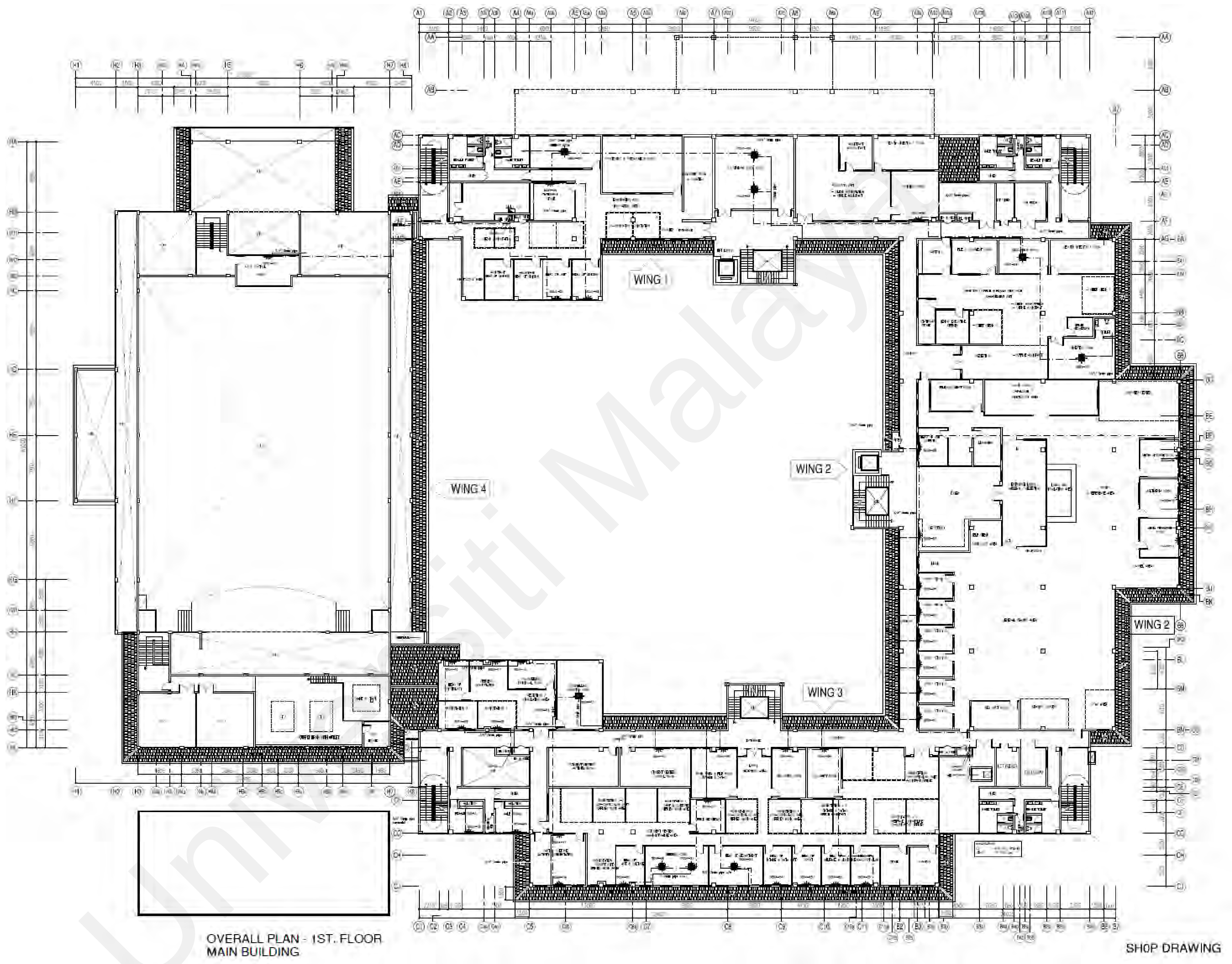
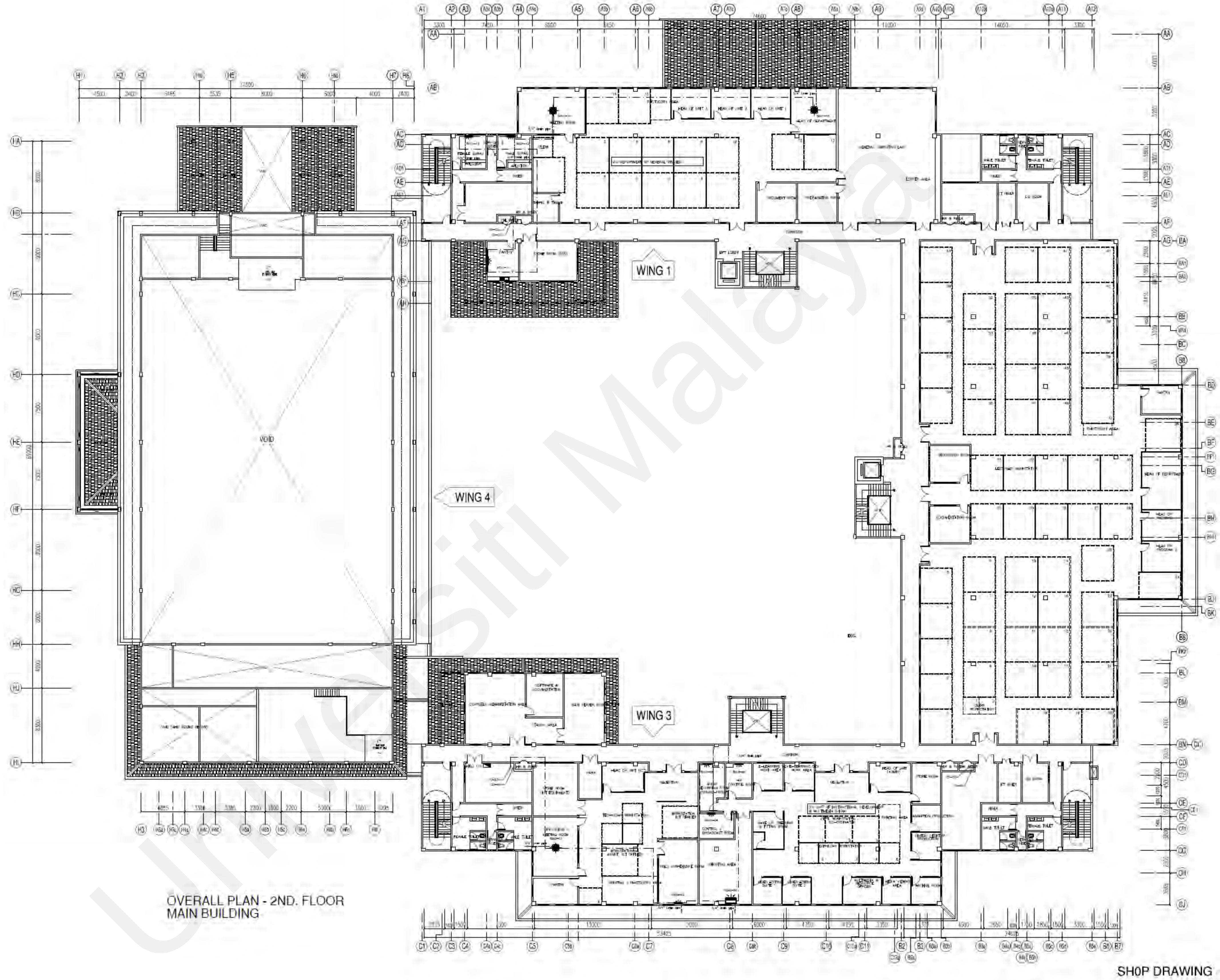

WING 1
WING 2
WING 2
WING 3
WING 4
OVERALL PLAN - 1ST. FLOOR
MAIN BUILDING
SHOP DRAWING

Figure 4.3



WING 1
WING 4
WING 3
VOID
OVERALL PLAN - 2ND. FLOOR
MAIN BUILDING
SHOP DRAWING

Figure 4.4

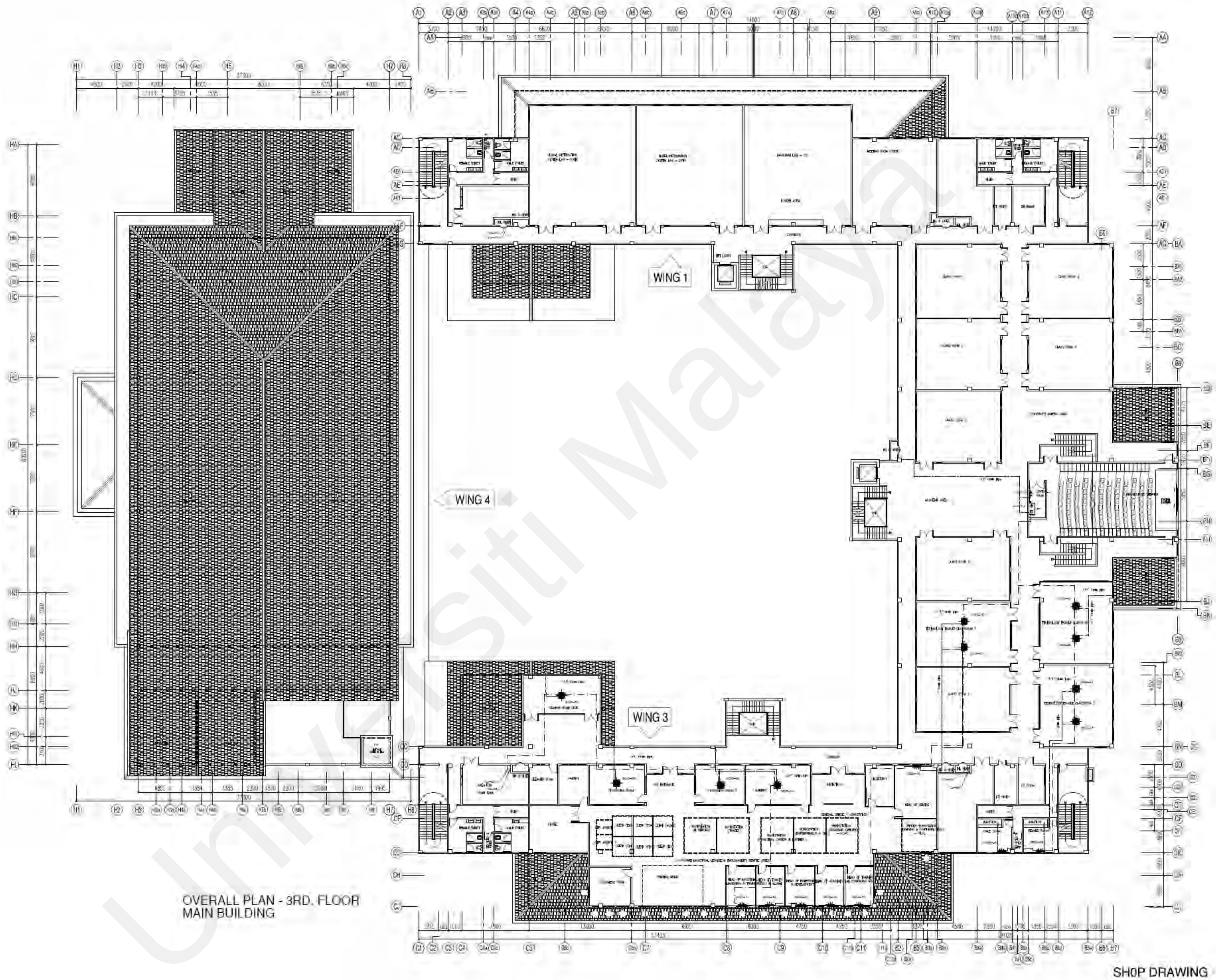

WING 1
WING 4
WING 3
OVERALL PLAN - 3RD. FLOOR
MAIN BUILDING
SHOP DRAWING

Figure 4.5

4.2 Load Estimation

The building consist of four floor which in total have 88 area need to be covered with an air-conditioning system. The load was estimate using the BTU Estimation Sheet that has been created based on the parameter below.

Table 4.1: Building Information

Location	Hulu Terengganu, Terengganu, Malaysia
Number of Stories	4
Building Type	Educational Institution
Gross Building Area	75,000 ft^2
Orientation	South East
Outdoor Air Inlet Temperature	35 ºC DB ~ 95 ºF DB
Indoor Air Inlet Temperature	27 ºC DB / 19 ºC WB ~ 80.6 ºF DB/ 66.2 ºF
Relative Humidity	50 % ~ 60 %
No of System	6 System

Hence from the information above the table in the figure below was generate. The total of 88 individual area has been individually calculated to estimate the cooling load required. Below are the figure 4.6 to figure 4.11 of BTU estimation sheet that was filled up for 6 area to represent the each of the system that was design for this building.

The calculation also involved the number of people, the solar gain for walls and roof, the transition and gain other than wall and roof such as ceiling, outside air condition, the electrical appliance such as light and other equipment load. As the orientation was set to be at the south east, thus the parameter for solar gain was then to be refer at south east condition.

As a result from this calculation, the total cooling load that need to fulfill by the systems is 150 tons. Hence three type of system was then design and their performance was then been evaluated. The summary of estimated load can be refer from table 4.2.

BTU ESTIMATION SHEET

JOB NAME:	**Politeknik Hulu Terengganu**					Date: 16/5/2019
SPACE FOR:	**SCU1-01 AV Room**			**HT.(Ft.)**	9.8	Estimated By: Nurshafiq
SIZE:	Width	23.62	ft			
	Length	11.81	ft			
	Total	**278.99**	**Sq ft**	**2,734**	**Cu ft**	Estimate for: **SUMMER**

SOLAR GAIN GLASS				**HEAT GAIN**
ITEM	Area	Sun Gain	Factor	**Btu/hour**
	(Sq ft)	(Btu/h.sq ft)		
N - Glass	0	14	0.56	0
NE - Glass	0	12	0.56	0
E - Glass	0	12	0.56	0
SE - Glass	0	12	0.56	0
S - Glass	0	12	0.56	0
SW - Glass	0	100	0.56	0
W - Glass	0	164	0.56	0
NW - Glass	0.00	123	0.56	0
Sky light(NW)				0
SOLAR & TRANS. GAIN WALLS & ROOF				
ITEM	Area	Eq. temp. diff.	U	
	(Sq ft)	(°F)	(Btu/h.sq ft)	
N - Wall	0	25	0.36	0
NE - Wall	231.4932	31	0.36	2,583
E - Wall	0	39	0.36	0
SE - Wall	0	39	0.36	0
S - Wall	0	37	0.36	0
SW - Wall	0	35	0.36	0
W - Wall	0	33	0.36	0
NW - Wall	0.00	27	0.36	0
Roof Sun	0	55	0.12	0
Roof Shaded				0
TRANS. GAIN EXCEPT WALLS & ROOF				
ITEM	Area	Temp. diff.	U	
	(Sq ft)	(°F)	(Btu/h.sq ft)	
All Glass	0	14.4	1.13	0
Partition	0	9.4	0.32	0
Ceiling	279	9.4	0.32	839
Floor	0	9.4	0.40	0
INFILTRATION AND OUTSIDE AIR				
Inflatration CFM	°F	BF	FACTOR	
0	14	-	1.08	0
Outdoor Air CFM	°F	BF	FACTOR	
273	14	0.15	1.08	638
INTERNAL HEAT GAIN				
People	2	Nos x	245	490
Light	2	1.10	3.41	8
Eq. Load	558	W x	3.41	1,903
			ROOM SENSIBLE HEAT (RSH)	6,461
Supply duct heat gain +	Supply duct leak. loss +	Heat gain from fan HP(%)	0.0%	0
		Safety factor (%)	5%	323
			EFFECTIVE ROOM SENSIBLE HEAT (ERSH)	6,784
LATENT HEAT				
People	2	Nos X	205	410
			ROOM LATENT HEAT (RLH)	410
Supply duct leakage loss +		Safety factor %	7.5%	31
Inflatration CFM	GR/LB	BF	FACTOR	
0	3.2	-	0.68	0
Outdoor Air CFM	GR/LB	BF	FACTOR	
273	3.2	0.15	0.68	89
			EFFECTIVE ROOM LATENT HEAT (ERLH)	530
			EFFECTIVE ROOM TOTAL HEAT (ERTH)	7,314
OUTSIDE AIR HEAT (SENSIBLE)				
CFM	°F	1 - BF	FACTOR	
273	14	0.85	1.08	3,614
OUTSIDE AIR HEAT (LATENT)				
CFM	GR/LB	1 - BF	FACTOR	
273	3.2	0.85	0.68	506
			HEAT SUB TOTAL	11,434
Return duct heat gain & leak. loss +	HP Pump +	Dehum. & Pipe loss (%)	3%	343
TR	**0.98**		**GRAND TOTAL HEAT**	11,777

CONDITION	**DB (°F)**	**WB (°F)**	**% RH**	**DP (°F)**	**GR/LB**
OUTSIDE	95	75.2			74.7
ROOM	80.6	66.2	55		71.5
DIFFERENCE	**14**	XXXX	XXXX	XXXX	**3.2**

OUTSIDE AIR (VENTILATION)		
25 CFM/Person	CFM VENTILATION	50
0.06 CFM/Sq. Ft	CFM VENTILATION	17
6 ACPH	CFM VENTILATION	273
	REQUIRED VENTILATION CFM	**273**

EFF. SENSIBLE HEAT FACTOR (ESHF) =		0.93
Indicated ADP =	54	°F
Selected ADP =	51	°F
Dehum. temp rise =	25.16	°F
DEHUMIDIFIED CFM =	**250**	

NOTES

Occupancy =	2	Persons
Light =	1.0	Watt/Sqft
Eq. Load =	0.6	KW
Required Fresh ai	300	CFM

CHECK FIGURES

CFM/Sq. ft. =	0.89
Sq. Ft./Ton =	284
CFM/Ton =	254

Figure 4.6: System 1 (AV Room)

BTU ESTIMATION SHEET

JOB NAME:	**Politeknik Hulu Terengganu**					Date: 16/5/2019
SPACE FOR:	**SCU2-01 VIP Room**		**HT.(Ft.)**	9.8		Estimated By: Nurshafiq
SIZE:	Width	13.00	ft			
	Length	19.50	ft			
	Total	**253.50**	**Sq ft**	**2,484**	**Cu ft**	Estimate for: **SUMMER**

SOLAR GAIN GLASS				**HEAT GAIN**
ITEM	Area	Sun Gain	Factor	**Btu/hour**
	(Sq ft)	(Btu/h.sq ft)		
N - Glass	0	14	0.56	0
NE - Glass	0	12	0.56	0
E - Glass	0	12	0.56	0
SE - Glass	0	12	0.56	0
S - Glass	0	12	0.56	0
SW - Glass	0	100	0.56	0
W - Glass	0	164	0.56	0
NW - Glass	0.00	123	0.56	0
Sky light(NW)				0
SOLAR & TRANS. GAIN WALLS & ROOF				
ITEM	Area	Eq. temp. diff.	U	
	(Sq ft)	(°F)	(Btu/h.sq ft)	
N - Wall	0	25	0.36	0
NE - Wall	127.4	31	0.36	1,422
E - Wall	0	39	0.36	0
SE - Wall	0	39	0.36	0
S - Wall	0	37	0.36	0
SW - Wall	0	35	0.36	0
W - Wall	0	33	0.36	0
NW - Wall	0.00	27	0.36	0
Roof Sun	0	55	0.12	0
Roof Shaded				0
TRANS. GAIN EXCEPT WALLS & ROOF				
ITEM	Area	Temp. diff.	U	
	(Sq ft)	(°F)	(Btu/h.sq ft)	
All Glass	0	14.4	1.13	0
Partition	0	9.4	0.32	0
Ceiling	254	9.4	0.32	763
Floor	0	9.4	0.40	0
INFILTRATION AND OUTSIDE AIR				
Inflatration CFM	°F	BF	FACTOR	
0	14	-	1.08	0
Outdoor Air CFM	°F	BF	FACTOR	
248	14	0.15	1.08	580
INTERNAL HEAT GAIN				
People	6	Nos x	245	1,470
Light	2	1.10	3.41	8
Eq. Load	507	W x	3.41	1,729
		ROOM SENSIBLE HEAT (RSH)		5,970
Supply duct heat gain +	Supply duct leak. loss +	Heat gain from fan HP(%)	0.0%	0
		Safety factor (%)	5%	299
		EFFECTIVE ROOM SENSIBLE HEAT (ERSH)		6,269
LATENT HEAT				
People	6	Nos X	205	1,230
		ROOM LATENT HEAT (RLH)		1,230
Supply duct leakage loss +		Safety factor %	7.5%	92
Inflatration CFM	GR/LB	BF	FACTOR	
0	3.2	-	0.68	0
Outdoor Air CFM	GR/LB	BF	FACTOR	
248	3.2	0.15	0.68	81
		EFFECTIVE ROOM LATENT HEAT (ERLH)		1,403
		EFFECTIVE ROOM TOTAL HEAT (ERTH)		7,672
OUTSIDE AIR HEAT (SENSIBLE)				
CFM	°F	1 - BF	FACTOR	
248	14	0.85	1.08	3,284
OUTSIDE AIR HEAT (LATENT)				
CFM	GR/LB	1 - BF	FACTOR	
248	3.2	0.85	0.68	459
		HEAT SUB TOTAL		11,416
Return duct heat gain & leak. loss +	HP Pump +	Dehum. & Pipe loss (%)	3%	342
TR	**0.98**	**GRAND TOTAL HEAT**		11,758

CONDITION	**DB (°F)**	**WB (°F)**	**% RH**	**DP (°F)**	**GR/LB**
OUTSIDE	95	75.2			74.7
ROOM	80.6	66.2	55		71.5
DIFFERENCE	**14**	XXXX	XXXX	XXXX	**3.2**

OUTSIDE AIR (VENTILATION)			
25	CFM/Person	CFM VENTILATION	150
0.06	CFM/Sq. Ft	CFM VENTILATION	15
6	ACPH	CFM VENTILATION	248
		REQUIRED VENTILATION CFM	**248**

EFF. SENSIBLE HEAT FACTOR (ESHF) =	0.82	
Indicated ADP =	54	°F
Selected ADP =	51	°F
Dehum. temp rise =	25.16	°F
DEHUMIDIFIED CFM =	**231**	

NOTES

Occupancy =	6	Persons
Light =	1.0	Watt/Sqft
Eq. Load =	0.5	KW
Required Fresh a	300	CFM

CHECK FIGURES

CFM/Sq. ft. =	0.91
Sq. Ft./Ton =	259
CFM/Ton =	235

Figure 4.7: System 2 (VIP Room)

BTU ESTIMATION SHEET

JOB NAME:	**Politeknik Hulu Terengganu**						Date: 16/5/2019
SPACE FOR:	**SCU3-01 Disc/Meet Room**			**HT.(Ft.)**	9.8		Estimated By: Nurshafiq
SIZE:	Width	23.40	ft				
	Length	30.60	ft				
	Total	**716.04**	**Sq ft**	**7,017**	**Cu ft**	Estimate for: **SUMMER**	

SOLAR GAIN GLASS				**HEAT GAIN**
ITEM	Area	Sun Gain	Factor	**Btu/hour**
	(Sq ft)	(Btu/h.sq ft)		
N - Glass	0	14	0.56	0
NE - Glass	0	12	0.56	0
E - Glass	0	12	0.56	0
SE - Glass	0	12	0.56	0
S - Glass	0	12	0.56	0
SW - Glass	0	100	0.56	0
W - Glass	0	164	0.56	0
NW - Glass	0.00	123	0.56	0
Sky light(NW)				0
SOLAR & TRANS. GAIN WALLS & ROOF				
ITEM	Area	Eq. temp. diff.	U	
	(Sq ft)	(°F)	(Btu/h.sq ft)	
N - Wall	0	25	0.36	0
NE - Wall	229.32	31	0.36	2,559
E - Wall	0	39	0.36	0
SE - Wall	0	39	0.36	0
S - Wall	0	37	0.36	0
SW - Wall	0	35	0.36	0
W - Wall	0	33	0.36	0
NW - Wall	0.00	27	0.36	0
Roof Sun	0	55	0.12	0
Roof Shaded				0
TRANS. GAIN EXCEPT WALLS & ROOF				
ITEM	Area	Temp. diff.	U	
	(Sq ft)	(°F)	(Btu/h.sq ft)	
All Glass	0	14.4	1.13	0
Partition	0	9.4	0.32	0
Ceiling	716	9.4	0.32	2,154
Floor	0	9.4	0.40	0
INFILTRATION AND OUTSIDE AIR				
Inflatration CFM	°F	BF	FACTOR	
0	14	-	1.08	0
Outdoor Air CFM	°F	BF	FACTOR	
702	14	0.15	1.08	1,637
INTERNAL HEAT GAIN				
People	20	Nos x	245	4,900
Light	2	1.10	3.41	8
Eq. Load	1432	W x	3.41	4,883
		ROOM SENSIBLE HEAT (RSH)		16,141
Supply duct heat gain +	Supply duct leak. loss +	Heat gain from fan HP(%)	0.0%	0
		Safety factor (%)	5%	807
	EFFECTIVE ROOM SENSIBLE HEAT (ERSH)			16,948
LATENT HEAT				
People	20	Nos X	205	4,100
		ROOM LATENT HEAT (RLH)		4,100
Supply duct leakage loss +		Safety factor %	7.5%	308
Inflatration CFM	GR/LB	BF	FACTOR	
0	3.2	-	0.68	0
Outdoor Air CFM	GR/LB	BF	FACTOR	
702	3.2	0.15	0.68	229
	EFFECTIVE ROOM LATENT HEAT (ERLH)			4,637
	EFFECTIVE ROOM TOTAL HEAT (ERTH)			21,585
OUTSIDE AIR HEAT (SENSIBLE)				
CFM	°F	1 - BF	FACTOR	
702	14	0.85	1.08	9,276
OUTSIDE AIR HEAT (LATENT)				
CFM	GR/LB	1 - BF	FACTOR	
702	3.2	0.85	0.68	1,298
		HEAT SUB TOTAL		32,159
Return duct heat gain & leak. loss +	HP Pump +	Dehum. & Pipe loss (%)	3%	965
TR	**2.76**	**GRAND TOTAL HEAT**		33,123

CONDITION	**DB (°F)**	**WB (°F)**	**% RH**	**DP (°F)**	**GR/LB**
OUTSIDE	95	75.2			74.7
ROOM	80.6	66.2	55		71.5
DIFFERENCE	**14**	XXXX	XXXX	XXXX	**3.2**

OUTSIDE AIR (VENTILATION)			
25	CFM/Person	CFM VENTILATION	500
0.06	CFM/Sq. Ft	CFM VENTILATION	43
6	ACPH	CFM VENTILATION	702
		REQUIRED VENTILATION CFM	**702**

EFF. SENSIBLE HEAT FACTOR (ESHF) =		0.79
Indicated ADP =	54	°F
Selected ADP =	51	°F
Dehum. temp rise =	25.16	°F
DEHUMIDIFIED CFM =	**624**	

NOTES

Occupancy =	20	Persons
Light =	1.0	Watt/Sqft
Eq. Load =	1.4	KW
Required Fresh a	300	CFM

CHECK FIGURES

CFM/Sq. ft. =	0.87
Sq. Ft./Ton =	259
CFM/Ton =	226

Figure 4.8: System 3 (a) (Discussion/Meeting Room)

BTU ESTIMATION SHEET

JOB NAME:	**Politeknik Hulu Terengganu**						Date: 16/5/2019
SPACE FOR:	**SCU3-17/18 Tech. Enable Classroom**			**HT.(Ft.)**	9.8		Estimated By: Nurshafiq
SIZE:	Width	35.43	ft				
	Length	23.62	ft				
	Total	**836.86**	**Sq ft**	**8,201**	**Cu ft**	Estimate for: **SUMMER**	

SOLAR GAIN GLASS				**HEAT GAIN**
ITEM	Area	Sun Gain	Factor	**Btu/hour**
	(Sq ft)	(Btu/h.sq ft)		
N - Glass	0	14	0.56	0
NE - Glass	0	12	0.56	0
E - Glass	0	12	0.56	0
SE - Glass	0	12	0.56	0
S - Glass	0	12	0.56	0
SW - Glass	0	100	0.56	0
W - Glass	0	164	0.56	0
NW - Glass	0.00	123	0.56	0
Sky light(NW)				0
SOLAR & TRANS. GAIN WALLS & ROOF				
ITEM	Area	Eq. temp. diff.	U	
	(Sq ft)	(°F)	(Btu/h.sq ft)	
N - Wall	0	25	0.36	0
NE - Wall	347.214	31	0.36	3,875
E - Wall	0	39	0.36	0
SE - Wall	0	39	0.36	0
S - Wall	0	37	0.36	0
SW - Wall	0	35	0.36	0
W - Wall	0	33	0.36	0
NW - Wall	0.00	27	0.36	0
Roof Sun	0	55	0.12	0
Roof Shaded				0
TRANS. GAIN EXCEPT WALLS & ROOF				
ITEM	Area	Temp. diff.	U	
	(Sq ft)	(°F)	(Btu/h.sq ft)	
All Glass	0	14.4	1.13	0
Partition	0	9.4	0.32	0
Ceiling	837	9.4	0.32	2,517
Floor	0	9.4	0.40	0
INFILTRATION AND OUTSIDE AIR				
Inflatration CFM	°F	BF	FACTOR	
0	14	-	1.08	0
Outdoor Air CFM	°F	BF	FACTOR	
1050	14	0.15	1.08	2,450
INTERNAL HEAT GAIN				
People	40	Nos x	245	9,800
Light	2	1.10	3.41	8
Eq. Load	1674	W x	3.41	5,707
		ROOM SENSIBLE HEAT (RSH)		24,357
Supply duct heat gain +	Supply duct leak. loss +	Heat gain from fan HP(%)	0.0%	0
		Safety factor (%)	5%	1,218
		EFFECTIVE ROOM SENSIBLE HEAT (ERSH)		25,575
LATENT HEAT				
People	40	Nos X	205	8,200
		ROOM LATENT HEAT (RLH)		8,200
Supply duct leakage loss +		Safety factor %	7.5%	615
Inflatration CFM	GR/LB	BF	FACTOR	
0	3.2	-	0.68	0
Outdoor Air CFM	GR/LB	BF	FACTOR	
1050	3.2	0.15	0.68	343
		EFFECTIVE ROOM LATENT HEAT (ERLH)		9,158
		EFFECTIVE ROOM TOTAL HEAT (ERTH)		34,733
OUTSIDE AIR HEAT (SENSIBLE)				
CFM	°F	1 - BF	FACTOR	
1050	14	0.85	1.08	13,883
OUTSIDE AIR HEAT (LATENT)				
CFM	GR/LB	1 - BF	FACTOR	
1050	3.2	0.85	0.68	1,942
		HEAT SUB TOTAL		50,558
Return duct heat gain & leak. loss +	HP Pump +	Dehum. & Pipe loss (%)	3%	1,517
TR	**4.34**	**GRAND TOTAL HEAT**		52,075

CONDITION	**DB (°F)**	**WB (°F)**	**% RH**	**DP (°F)**	**GR/LB**
OUTSIDE	95	75.2			74.7
ROOM	80.6	66.2	55		71.5
DIFFERENCE	**14**	XXXX	XXXX	XXXX	**3.2**

OUTSIDE AIR (VENTILATION)			
25	CFM/Person	CFM VENTILATION	1000
0.06	CFM/Sq. Ft	CFM VENTILATION	50
6	ACPH	CFM VENTILATION	820
		REQUIRED VENTILATION CFM	**1050**

EFF. SENSIBLE HEAT FACTOR (ESHF) =	0.74	
Indicated ADP =	54	°F
Selected ADP =	51	°F
Dehum. temp rise =	25.16	°F
DEHUMIDIFIED CFM =	**941**	

NOTES

Occupancy =	40	Persons
Light =	1.0	Watt/Sqft
Eq. Load =	1.7	KW
Required Fresh a	300	CFM

CHECK FIGURES

CFM/Sq. ft. =	1.12
Sq. Ft./Ton =	193
CFM/Ton =	217

Figure 4.9: System 3 (b) (Technology Enable Classroom)

BTU ESTIMATION SHEET

JOB NAME:	**Politeknik Hulu Terengganu**					Date: 16/5/2019
SPACE FOR:	**SCU4-01 Preparation Area**			**HT.(Ft.)**	9.8	Estimated By: Nurshafiq
SIZE:	Width	10.50	ft			
	Length	16.07	ft			
	Total	**168.74**	**Sq ft**	**1,654**	**Cu ft**	Estimate for: **SUMMER**

SOLAR GAIN GLASS				**HEAT GAIN**
ITEM	Area	Sun Gain	Factor	**Btu/hour**
	(Sq ft)	(Btu/h.sq ft)		
N - Glass	0	14	0.56	0
NE - Glass	0	12	0.56	0
E - Glass	0	12	0.56	0
SE - Glass	0	12	0.56	0
S - Glass	0	12	0.56	0
SW - Glass	0	100	0.56	0
W - Glass	0	164	0.56	0
NW - Glass	0.00	123	0.56	0
Sky light(NW)				0
SOLAR & TRANS. GAIN WALLS & ROOF				
ITEM	Area	Eq. temp. diff.	U	
	(Sq ft)	(°F)	(Btu/h.sq ft)	
N - Wall	0	25	0.36	0
NE - Wall	102.9	31	0.36	1,148
E - Wall	0	39	0.36	0
SE - Wall	0	39	0.36	0
S - Wall	0	37	0.36	0
SW - Wall	0	35	0.36	0
W - Wall	0	33	0.36	0
NW - Wall	0.00	27	0.36	0
Roof Sun	0	55	0.12	0
Roof Shaded				0
TRANS. GAIN EXCEPT WALLS & ROOF				
ITEM	Area	Temp. diff.	U	
	(Sq ft)	(°F)	(Btu/h.sq ft)	
All Glass	0	14.4	1.13	0
Partition	0	9.4	0.32	0
Ceiling	169	9.4	0.32	508
Floor	0	9.4	0.40	0
INFILTRATION AND OUTSIDE AIR				
Inflatration CFM	°F	BF	FACTOR	
0	14	-	1.08	0
Outdoor Air CFM	°F	BF	FACTOR	
210	14	0.15	1.08	490
INTERNAL HEAT GAIN				
People	8	Nos x	245	1,960
Light	2	1.10	3.41	8
Eq. Load	337	W x	3.41	1,151
		ROOM SENSIBLE HEAT (RSH)		5,264
Supply duct heat gain +	Supply duct leak. loss +	Heat gain from fan HP(%)	0.0%	0
		Safety factor (%)	5%	263
		EFFECTIVE ROOM SENSIBLE HEAT (ERSH)		5,528
LATENT HEAT				
People	8	Nos X	205	1,640
		ROOM LATENT HEAT (RLH)		1,640
Supply duct leakage loss +		Safety factor (	7.5%	123
Inflatration CFM	GR/LB	BF	FACTOR	
0	3.2	-	0.68	0
Outdoor Air CFM	GR/LB	BF	FACTOR	
210	3.2	0.15	0.68	69
		EFFECTIVE ROOM LATENT HEAT (ERLH)		1,832
		EFFECTIVE ROOM TOTAL HEAT (ERTH)		7,359
OUTSIDE AIR HEAT (SENSIBLE)				
CFM	°F	1 - BF	FACTOR	
210	14	0.85	1.08	2,778
OUTSIDE AIR HEAT (LATENT)				
CFM	GR/LB	1 - BF	FACTOR	
210	3.2	0.85	0.68	389
		HEAT SUB TOTAL		10,525
Return duct heat gain & leak. loss +	HP Pump +	Dehum. & Pipe loss (%)	3%	316
TR	**0.90**	**GRAND TOTAL HEAT**		10,841

CONDITION	**DB (°F)**	**WB (°F)**	**% RH**	**DP (°F)**	**GR/LB**
OUTSIDE	95	75.2			74.7
ROOM	80.6	66.2	55		71.5
DIFFERENCE	**14**	XXXX	XXXX	XXXX	**3.2**

OUTSIDE AIR (VENTILATION)			
25	CFM/Person	CFM VENTILATION	200
0.06	CFM/Sq. Ft	CFM VENTILATION	10
6	ACPH	CFM VENTILATION	165
		REQUIRED VENTILATION CFM	**210**

EFF. SENSIBLE HEAT FACTOR (ESHF) =	0.75	
Indicated ADP =	54	°F
Selected ADP =	51	°F
Dehum. temp rise =	25.16	°F
DEHUMIDIFIED CFM =	**203**	

NOTES

Occupancy =	8	Persons
Light =	1.0	Watt/Sqft
Eq. Load =	0.3	KW
Required Fresh	300	CFM

CHECK FIGURES

CFM/Sq. ft. =	1.21
Sq. Ft./Ton =	187
CFM/Ton =	225

Figure 4.10: System 4 (Preparation Area)

BTU ESTIMATION SHEET

JOB NAME:	**Politeknik Hulu Terengganu**					Date: 16/5/2019
SPACE FOR:	**SCU5-01 Group Study 6**			**HT.(Ft.)**	9.8	Estimated By: Nurshafiq
SIZE:	Width	13.12	ft			
	Length	10.50	ft			
	Total	**137.76**	**Sq ft**	**1,350**	**Cu ft**	Estimate for: **SUMMER**

SOLAR GAIN GLASS				**HEAT GAIN**
ITEM	Area (Sq ft)	Sun Gain (Btu/h.sq ft)	Factor	**Btu/hour**
N - Glass	0	14	0.56	0
NE - Glass	0	12	0.56	0
E - Glass	0	12	0.56	0
SE - Glass	0	12	0.56	0
S - Glass	0	12	0.56	0
SW - Glass	0	100	0.56	0
W - Glass	0	164	0.56	0
NW - Glass	0.00	123	0.56	0
Sky light(NW)				0
SOLAR & TRANS. GAIN WALLS & ROOF				
ITEM	Area (Sq ft)	Eq. temp. diff. (°F)	U (Btu/h.sq ft)	
N - Wall	0	25	0.36	0
NE - Wall	128.576	31	0.36	1,435
E - Wall	0	39	0.36	0
SE - Wall	0	39	0.36	0
S - Wall	0	37	0.36	0
SW - Wall	0	35	0.36	0
W - Wall	0	33	0.36	0
NW - Wall	0.00	27	0.36	0
Roof Sun	0	55	0.12	0
Roof Shaded				0
TRANS. GAIN EXCEPT WALLS & ROOF				
ITEM	Area (Sq ft)	Temp. diff. (°F)	U (Btu/h.sq ft)	
All Glass	0	14.4	1.13	0
Partition	0	9.4	0.32	0
Ceiling	138	9.4	0.32	414
Floor	0	9.4	0.40	0
INFILTRATION AND OUTSIDE AIR				
Inflatration CFM	°F	BF	FACTOR	
0	14	-	1.08	0
Outdoor Air CFM	°F	BF	FACTOR	
158	14	0.15	1.08	369
INTERNAL HEAT GAIN				
People	6	Nos x	245	1,470
Light	2	1.10	3.41	8
Eq. Load	276	W x	3.41	940
		ROOM SENSIBLE HEAT (RSH)		4,636
Supply duct heat gain +	Supply duct leak. loss +	Heat gain from fan HP(%)	0.0%	0
		Safety factor (%)	5%	232
		EFFECTIVE ROOM SENSIBLE HEAT (ERSH)		4,867
LATENT HEAT				
People	6	Nos X	205	1,230
		ROOM LATENT HEAT (RLH)		1,230
Supply duct leakage loss + Safety factor (			7.5%	92
Inflatration CFM	GR/LB	BF	FACTOR	
0	3.2	-	0.68	0
Outdoor Air CFM	GR/LB	BF	FACTOR	
158	3.2	0.15	0.68	52
		EFFECTIVE ROOM LATENT HEAT (ERLH)		1,374
		EFFECTIVE ROOM TOTAL HEAT (ERTH)		6,241
OUTSIDE AIR HEAT (SENSIBLE)				
CFM	°F	1 - BF	FACTOR	
158	14	0.85	1.08	2,092
OUTSIDE AIR HEAT (LATENT)				
CFM	GR/LB	1 - BF	FACTOR	
158	3.2	0.85	0.68	293
		HEAT SUB TOTAL		8,626
Return duct heat gain & leak. loss +	HP Pump +	Dehum. & Pipe loss (%)	3%	259
TR	**0.74**	**GRAND TOTAL HEAT**		8,885

CONDITION	**DB (°F)**	**WB (°F)**	**% RH**	**DP (°F)**	**GR/LB**
OUTSIDE	95	75.2			74.7
ROOM	80.6	66.2	55		71.5
DIFFERENCE	**14**	XXXX	XXXX	XXXX	**3.2**

OUTSIDE AIR (VENTILATION)		
25 CFM/Person	CFM VENTILATION	150
0.06 CFM/Sq. Ft	CFM VENTILATION	8
6 ACPH	CFM VENTILATION	135
	REQUIRED VENTILATION CFM	**158**

EFF. SENSIBLE HEAT FACTOR (ESHF) =		0.78
Indicated ADP =	54	°F
Selected ADP =	51	°F
Dehum. temp rise =	25.16	°F
DEHUMIDIFIED CFM =	**179**	

NOTES

Occupancy =	6	Persons
Light =	1.0	Watt/Sqft
Eq. Load =	0.3	KW
Required Fresh ai	300	CFM

CHECK FIGURES

CFM/Sq. ft. =	1.30
Sq. Ft./Ton =	186
CFM/Ton =	242

Figure 4.11: System 5 (Group Study 6)

Throughout the above figure, the table of summary of cooling load estimation for each of the area was generated as below.

Table 4.2: Summary of estimated load

No	Unit No.	Location	Width, ft.	Length, ft.	Height, ft.	Area (ft^2)	Volume (ft^3)	Capacity (Btu/hr.)
1	SCU1-01	AV Room	23.62	11.81	9.80	278.99	2734.14	12,498
2	SCU1-02	Head of Unit	23.62	11.81	9.80	278.99	2734.14	12,506
3	SCU1-03	Head of Division	23.62	11.81	9.80	278.99	2734.14	12,498
4	SCU1-04	Stationery Room	17.06	14.76	9.80	251.87	2468.32	12,219
5	SCU1-05	Meeting Room	26.25	12.47	9.80	327.21	3206.71	18,204
6	SCU1-06/07	Seminar Room	29.53	28.87	9.80	852.48	8354.31	38,659
7	SCU1-08	Printing Room	17.06	19.68	9.80	335.83	3291.09	17,982
8	SCU1-09	Discussion Room	17.06	19.68	9.80	335.74	3290.26	16,098
9	SCU1-10	Director Room	24.93	14.76	9.80	368.12	3607.54	16,233
10	SCU1-11	Pantry	11.81	14.44	9.80	170.50	1670.86	8,618
11	SCU1-12	Female Surau	13.12	13.12	9.80	172.22	1687.74	9,350
12	SCU1-13	Male Surau	13.12	13.12	9.80	172.22	1687.74	9,350
13	SCU1-14	Meeting Room	21.65	17.06	9.80	369.41	3620.20	21,847
14	SCU1-15	Head of Department	17.06	17.06	9.80	291.05	2852.28	11,867
15	SCU2-01	VIP Room	13.00	19.50	9.80	253.50	2484.30	12,155
16	SCU2-02	Assist Engineer	11.81	14.44	9.80	170.54	1671.26	8,626
17	SCU2-03	Head of Unit Room	11.81	14.44	9.80	170.54	1671.26	8,618
18	SCU2-04	Discussion Room 1	21.65	25.60	9.80	554.24	5431.55	23,040
19	SCU2-05	Workstation	17.06	19.68	9.80	335.83	3291.09	17,082
20	SCU2-06	Dining Room	17.06	9.19	9.80	156.72	1535.84	12,000
21	SCU2-07	Main Meeting Room	55.50	40.00	9.80	2220.00	21756.00	86,642
22	SCU2-08	Guest Holding Room	17.06	12.00	9.80	204.72	2006.27	12,568
23	SCU2-09	Surau (M/F)	12.80	12.50	9.80	159.94	1567.40	9,499
24	SCU2-10/11	Interpretation Studio	28.71	25.00	9.80	717.68	7033.22	26,797

25	SCU2-12	Massage Room	15.50	23.00	9.80	356.50	3493.70	17,457
26	SCU2-13	Pedicure / Manicure Nail	15.50	23.00	9.80	356.50	3493.70	17,457
27	SCU2-14	SPA & Wellness Training Room	19.70	25.80	9.80	508.26	4980.95	23,417
28	SCU2-15	Lab Assist	10.50	12.90	9.80	135.43	1327.23	7,426
29	SCU3-01	Discussion / Meeting Room	23.40	30.60	9.80	716.04	7017.19	33,838
30	SCU3-02	Video Conference Room	14.44	24.93	9.80	359.94	3527.38	20,664
31	SCU3-03	Shooting Area	25.78	34.12	9.80	879.61	8620.21	36,417
32	SCU3-04	Control / Broadcast Room	13.12	9.19	9.80	120.55	1181.42	8,131
33	SCU3-05	Audio Recording Room	13.80	17.67	9.80	243.85	2389.69	11,505
34	SCU3-06	AV Control Room	9.18	13.12	9.80	120.47	1180.62	7,544
35	SCU3-07	Seminar Room Cisec	28.87	14.44	9.80	416.77	4084.33	28,221
36	SCU3-08	Discussion Room 1	21.00	14.76	9.80	309.99	3037.93	21,041
37	SCU3-09	Discussion Room 1	21.00	14.76	9.80	309.99	3037.93	21,041
38	SCU3-10	Meeting	23.62	14.76	9.80	348.74	3417.67	24,739
39	SCU3-11	Head of Industrial	13.80	17.67	9.80	243.85	2389.69	11,014
40	SCU3-12	Head of Tracer	13.80	17.67	9.80	243.85	2389.69	11,014
41	SCU3-13	Head Of Entrepreneur	13.80	17.67	9.80	243.85	2389.69	11,014
42	SCU3-14	Head of Academic QAC	13.80	17.67	9.80	243.85	2389.69	11,014
43	SCU3-15	Head of Training	13.80	17.67	9.80	243.85	2389.69	11,014
44	SCU3-16	Head of Centre	17.74	23.48	9.80	416.54	4082.04	16,546
45	SCU3-17/18	Technology Enable Classroom 1	35.43	23.62	9.80	836.98	8.00	53,156
46	SCU3-19	Mini Lecturer Theater	55.12	35.80	9.80	1973.20	19337.40	89,314
47	SCU3-20	Control Room	6.56	13.12	9.80	86.11	843.87	5,800
48	SCU3-21/22	Technology Enable Classroom 2	27.56	30.84	9.80	849.90	8329.00	52,139
49	SCU3-23/24	Technology Enable Classroom 3	27.56	30.84	9.80	849.90	8329.00	29,437

50	SCU3-25	Female Surau	13.12	12.12	9.80	159.05	1558.72	9,535
51	SCU3-26	Male Surau	13.12	12.12	9.80	159.05	1558.72	9,535
52	SCU4-01	Preparation Area	10.50	16.07	9.80	168.71	1653.38	11,162
53	SCU4-02	Banquet Store / Preparation Area	16.97	20.83	9.80	353.50	3464.32	16,698
54	SCU4-03	Operation Centre	13.76	17.06	9.80	234.75	2300.51	12,715
55	SCU4-04/05/06/07	Staff Eating Room	33.54	28.70	9.80	962.60	9433.46	39,948
56	SCU4-08	Workstation 3	14.44	10.50	9.80	151.55	1485.21	9,644
57	SCU4-09	Workstation 2	14.44	10.50	9.80	151.55	1485.21	9,644
58	SCU4-10	Head of Psychology	13.80	17.67	9.80	243.85	2389.69	11,014
59	SCU4-11	Officer Workstation 1	14.44	14.44	9.80	208.38	2042.17	9,103
60	SCU4-12	Counselling Individual room	18.37	11.81	9.80	217.00	2126.55	10,923
61	SCU4-13	Counselling Grouping Room	18.58	26.25	9.80	487.66	4779.05	21,662
62	SCU4-14	Reception / Circulation Area	14.44	15.75	9.80	227.33	2227.82	12,596
63	SCU4-15	Male Surau	13.12	13.12	9.80	172.22	1687.74	9,842
64	SCU4-16	Female Surau	13.12	13.12	9.80	172.22	1687.74	9,842
65	SCU4-17/18	Activity / Presentation	18.58	16.70	9.80	310.29	3040.80	14,120
66	SCU4-19	Head of Art Culture	12.77	15.75	9.80	201.13	1971.05	8,653
67	SCU4-20/21	Meeting Room	23.62	15.75	9.80	371.99	3645.52	26,914
68	SCU4-22	Office Secretary	13.12	15.67	9.80	205.64	2015.28	9,817
69	SCU4-23	Head of Department	17.55	24.76	9.80	434.54	4258.47	17,033
70	SCU4-24	Head of Intake & Data Unit	13.12	15.74	9.80	206.51	2023.79	9,843
71	SCU4-25	Head of Sport	13.12	15.74	9.80	206.51	2023.79	9,843
72	SCU4-26	Head of Welfare & Discipline	13.12	15.74	9.80	206.51	2023.79	9,843
73	SCU4-27	Head of Co-Curriculum	13.12	15.74	9.80	206.51	2023.79	9,843
74	SCU4-28	Document Room	13.12	19.68	9.80	258.33	2531.61	11,324
75	SCU4-29	Discussion Room	14.67	23.50	9.80	344.75	3378.50	16,001

76	SCU4-30	Work Area & File Room (Intake & Data)	17.06	19.68	9.80	335.83	3291.09	16,098
77	SCU4-31	(Student Centre) Council Room	26.25	14.76	9.80	387.49	3797.42	19,931
78	SCU5-01	Group Study 6	13.12	10.50	9.80	137.77	1350.19	9,285
79	SCU5-02	Group Study 5	13.12	10.50	9.80	137.77	1350.19	9,285
80	SCU5-03	Group Study 4	13.12	10.50	9.80	137.77	1350.19	9,285
81	SCU5-04	Group Study 3	13.12	10.50	9.80	137.77	1350.19	9,285
82	SCU5-05	Group Study 2	13.12	10.50	9.80	137.77	1350.19	9,285
83	SCU5-06	Group Study 1	13.12	10.50	9.80	137.77	1350.19	9,285
84	SCU5-07	Depository	19.68	13.12	9.80	258.33	2531.61	12,792
85	SCU5-08	Head of Unit	11.81	14.44	9.80	170.50	1670.86	9,106
86	SCU5-09	Media Storage Room	14.44	14.44	9.80	208.38	2042.17	11,071
87	SCU5-10	Multimedia Room	14.44	14.44	9.80	208.38	2042.17	11,071
88	SCU5-11	Multimedia Resource Room	14.44	14.44	9.80	208.38	2042.17	14,113

4.3 Variable Refrigerant Flow System (VRF)

4.3.1 VRF System Design

Throughout deep analysis on the floor plan of the building with consideration of limitations highlighted previously, six system was designed for Polytechnics Hulu Terengganu (PHT). The design of VRF used Hitachi Sigma Free HNCQ model, which is the latest generation and technology offered by Johnson Control Hitachi Air Conditioning Sales Malaysia Sdn. Bhd. in Malaysia. In addition, this system was currently been installed in PHT. Below are the summary of equipment with the technical specification of indoor equipment selected for each area.

Table 4.3 shows the total cooling capacity for system 1 should be 76.2 kW. Hence, this system has been coupled with 28 HP outdoor unit with cooling capacity of 78.5 k. The capacity ratio for system one on 97.2 % which can be refer from table 3.8. Variable refrigerant flow configuration for system 2 can be refer from table 4.4. The total cooling capacity required for system 2 is 100 kW, thus this system has been coupled with 34 HP outdoor of 95 kW cooling capacity given the capacity ration to be 105.26%. Next for system 3 (a), the total cooling capacity required for the system is 38.2 kW. This can be referred from table 4.5. Hence, the system was pair with 14 HP outdoor unit with 40 kW rated cooling capacity. The capacity ratio for this system was 95.5%.

On top of that, for system 3 (b), from table 4.5 state that the total cooling capacity required was 140 kW. Thus, this system was then been paired with 48 HP outdoor unit with 136 kW rated cooling capacity. The capacity ratio for this system is 102.94%. System 4 require total cooling capacity of 156.4 kW and this can be refer from table 4.6. This system is coupled with 54 HP outdoor unit with 151.5 kW, give the capacity ration of 103.23%. Lastly, for system 5, the required total cooling capacity is 35.1 kW which can be refer from table 4.7. Hence this system was pair with 12 HP outdoor unit with 33.5 kW rated cooling capacity, which give out the capacity ratio of 104.78%.

Figure 4.12 to figure 4.17 show the schematic drawing that was design for each system. This drawing is an actual drawing based on the floor plan analysis.

Table 4.3: System design for System 1

Room Number/ Location		Cooling DB, ºC	Cooling WB, ºC	Relative Humidity	Room Number	Selected Model	Qty	Cooling Capacity, kW	Cooling Sensible Capacity, kW	Circulatory Airflow, m^3/min
SYS1[RAS-28HNBCMQ]										
SCU1-01	AV Room	27.0	19.6	50%	SCU1-01	RPK-1.3FSNQS	1	3.3	2.6	9.2
SCU1-02	Head of Unit	27.0	19.6	50%	SCU1-02	RPK-1.3FSNQS	1	3.3	2.6	9.2
SCU1-03	Head of Division	27.0	19.6	50%	SCU1-03	RPK-1.3FSNQS	1	3.3	2.6	9.2
SCU1-04	Stationery Room	27.0	19.6	50%	SCU1-04	RPK-1.3FSNQS	1	3.3	2.6	9.2
SCU1-05	Meeting Room	27.0	19.6	50%	SCU1-05	RCI-2.0FSKDNQ	1	5.1	4.6	17
SCU1-06	Seminar Room	27.0	19.6	50%	SCU1-06	RCI-4.0FSKDNQ	1	10.3	7.9	31
SCU1-07	Seminar Room	27.0	19.6	50%	SCU1-07	RCI-4.0FSKDNQ	1	10.3	7.9	31
SCU1-08	Printing Room	27.0	19.6	50%	SCU1-08	RPK-2.0FSNQS	1	5.1	3.5	12
SCU1-09	Discussion Room	27.0	19.6	50%	SCU1-09	RCI-2.0FSKDNQ	1	5.1	4.6	17
SCU1-10	Director Room	27.0	19.6	50%	SCU1-10	RCI-3.0FSKDNQ	1	7.7	6.2	23
SCU1-11	Pantry	27.0	19.6	50%	SCU1-11	RPK-1.0FSNQS	1	2.6	2	8.5
SCU1-12	Female Surau	27.0	19.6	50%	SCU1-12	RPK-1.0FSNQS	1	2.6	2	8.5
SCU1-13	Male Surau	27.0	19.6	50%	SCU1-13	RPK-1.0FSNQS	1	2.6	2	8.5
SCU1-14	Meeting Room	27.0	19.6	50%	SCU1-14	RCI-3.0FSKDNQ	1	7.7	6.2	23
SCU1-15	Head of Department	27.0	19.6	50%	SCU1-15	RCI-1.5FSKDNQ	1	3.9	3.4	17
Total								76.2	60.7	233.3

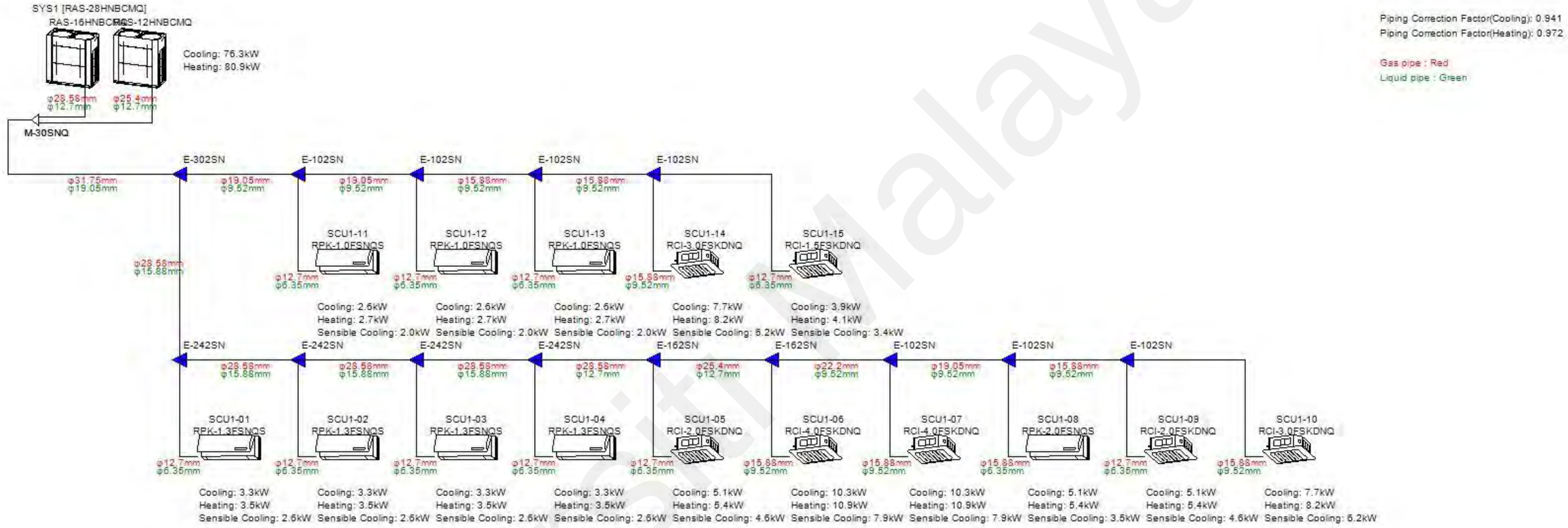


Figure 4.12: Schematic Drawing System 1

Table 4.4: System Design for System 2

Room Number/ Location		Cooling DB, ºC	Cooling WB, ºC	Relative Humidity	Room Number	Selected Model	Qty	Cooling Capacity, kW	Cooling Sensible Capacity, kW	Circulatory Airflow, m^3/min
SYS2[RAS-34HNBCMQ]										
SCU2-01	VIP Room	27.0	19.6	50%	SCU2-01	RCI-1.5FSKDNQ	1	4.3	3.8	17
SCU2-05	Assist Engineer	27.0	19.6	50%	SCU2-05	RCI-2.0FSKDNQ	1	5.7	5.1	17
SCU2-02	Head of Unit Room	27.0	19.6	50%	SCU2-02	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU2-03	Discussion Room 1	27.0	19.6	50%	SCU2-03	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU2-04	Workstation	27.0	19.6	50%	SCU2-04	RCI-2.5FSKDNQ	1	7.1	6.2	23
SCU2-06	Dining Room	27.0	19.6	50%	SCU2-06	RCI-1.5FSKDNQ	1	4.3	3.8	17
SCU2-07	Main Meeting Room	27.0	19.6	50%	SCU2-07	RPI-10FSN3Q	1	28.5	20	72
SCU2-08	Guest Holding Room	27.0	19.6	50%	SCU2-08	RCI-1.5FSKDNQ	1	4.3	3.8	17
SCU2-09	Surau (M/F)	27.0	19.6	50%	SCU2-09	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU2-10	Interpretation Studio	27.0	19.6	50%	SCU2-10	RCI-2.5FSKDNQ	1	7.1	6.2	23
SCU2-11	Interpretation Studio	27.0	19.6	50%	SCU2-11	RCI-2.5FSKDNQ	1	7.1	6.2	23
SCU2-12	Massage Room	27.0	19.6	50%	SCU2-12	RCI-2.0FSKDNQ	1	5.7	5.1	17
SCU2-13	Pedicure / Manicure Nail	27.0	19.6	50%	SCU2-13	RCI-2.0FSKDNQ	1	5.7	5.1	17
SCU2-14	SPA & Wellness Training Room	27.0	19.6	50%	SCU2-14	RCI-3.0FSKDNQ	1	8.6	6.9	23
SCU2-15	Lab Assist	27.0	19.6	50%	SCU2-15	RPK-1.0FSNQS	1	2.9	2.2	8.5
Total								100	81	300

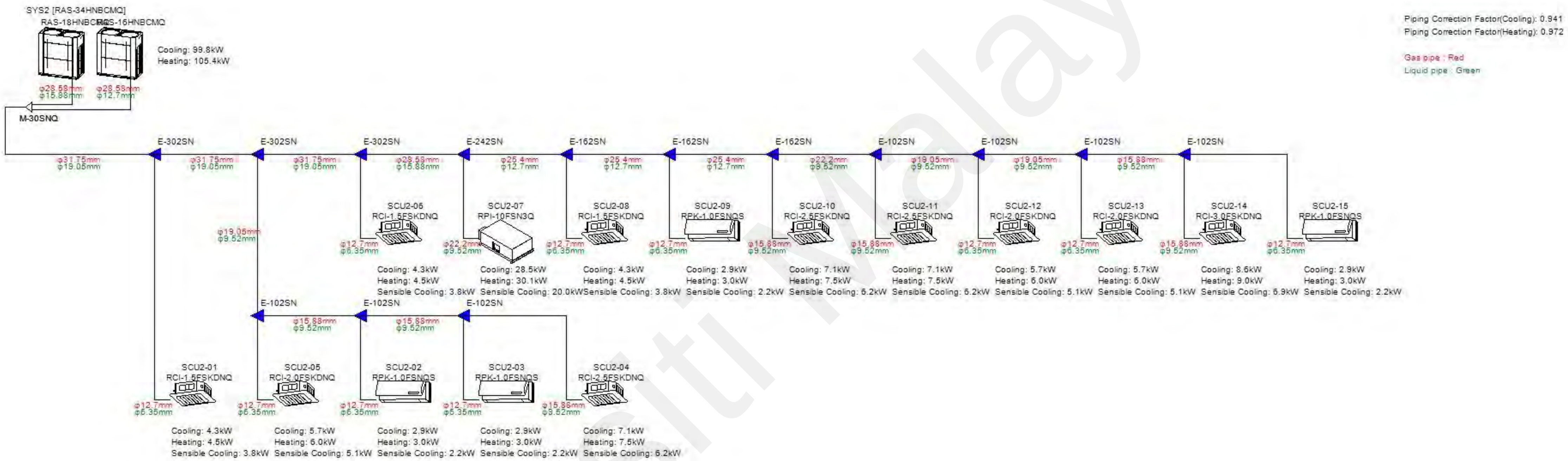


Figure 4.13: Schematic Drawing for System 2

Table 4.5: System Design for System 3 (a) & (b)

Room Number/ Location		Cooling DB, ºC	Cooling WB, ºC	Relative Humidity	Room Number	Selected Model	Qty	Cooling Capacity, kW	Cooling Sensible Capacity, kW	Circulatory Airflow, m^3/min
SYS3(a)[RAS-14HNBCMQ]										
SCU3-01	Discussion / Meeting Room	27.0	19.6	50%	SCU3-01	RCI-4.0FSKDNQ	1	11.5	8.8	31
SCU3-02	Video Conference Room	27.0	19.6	50%	SCU3-02	RPK-2.0FSNQS	1	5.7	4	12
SCU3-03	Shooting Area	27.0	19.6	50%	SCU3-03	RPC-4.0FSN3	1	11.5	8.8	30
SCU3-04	Control / Broadcast Room	27.0	19.6	50%	SCU3-04	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU3-05	Audio Recording Room	27.0	19.6	50%	SCU3-05	RPK-1.3FSNQS	1	3.7	2.9	9.2
SCU3-06	AV Control Room	27.0	19.6	50%	SCU3-06	RPK-1.0FSNQS	1	2.9	2.2	8.5
Total								**38.2**	**28.9**	**99.2**
SYS3(b)[RAS-48HNBCMQ]										
SCU3-07	Seminar Room Cisec	27.0	19.6	50%	SCU3-07	RCI-3.0FSKDNQ	1	8.4	6.8	23
SCU3-08	Discussion Room 1	27.0	19.6	50%	SCU3-08	RCI-2.5FSKDNQ	1	7	6.1	23
SCU3-09	Discussion Room 1	27.0	19.6	50%	SCU3-09	RCI-2.5FSKDNQ	1	7	6.1	23
SCU3-10	Meeting	27.0	19.6	50%	SCU3-10	RCI-2.5FSKDNQ	1	7	6.1	23
SCU3-11	Head of Industrial	27.0	19.6	50%	SCU3-11	RPK-1.3FSNQS	1	3.7	2.9	9.2
SCU3-12	Head of Tracer	27.0	19.6	50%	SCU3-12	RPK-1.3FSNQS	1	3.7	2.9	9.2
SCU3-13	Head Of Entrepreneur	27.0	19.6	50%	SCU3-13	RPK-1.3FSNQS	1	3.7	2.9	9.2

SCU3-14	Head of Academic QAC	27.0	19.6	50%	SCU3-14	RPK-1.3FSNQS	1	3.7	2.9	9.2
SCU3-15	Head of Training	27.0	19.6	50%	SCU3-15	RPK-1.3FSNQS	1	3.7	2.9	9.2
SCU3-16	Head of Centre	27.0	19.6	50%	SCU3-16	RPK-2.0FSNQS	1	5.6	3.9	12
SCU3-17	Technology Enable Classroom 1	27.0	19.6	50%	SCU3-17	RCI-3.0FSKDNQ	1	8.4	6.8	23
SCU3-18	Technology Enable Classroom 1	27.0	19.6	50%	SCU3-18	RCI-3.0FSKDNQ	1	8.4	6.8	23
SCU3-19	Mini Lecturer Theater	27.0	19.6	50%	SCU3-19	RPI-10FSN3Q	1	28.2	19.7	72
SCU3-20	Control Room	27.0	19.6	50%	SCU3-20	RPK-0.8FSNQS	1	2.3	1.8	8.5
SCU3-21	Technology Enable Classroom 2	27.0	19.6	50%	SCU3-21	RCI-3.0FSKDNQ	1	8.4	6.8	23
SCU3-22	Technology Enable Classroom 2	27.0	19.6	50%	SCU3-22	RCI-3.0FSKDNQ	1	8.4	6.8	23
SCU3-23	Technology Enable Classroom 3	27.0	19.6	50%	SCU3-23	RCI-3.0FSKDNQ	1	8.4	6.8	23
SCU3-24	Technology Enable Classroom 3	27.0	19.6	50%	SCU3-24	RCI-3.0FSKDNQ	1	8.4	6.8	23
SCU3-25	Female Surau	27.0	19.6	50%	SCU3-25	RPK-1.0FSNQS	1	2.8	2.2	8.5
SCU3-26	Male Surau	27.0	19.6	50%	SCU3-26	RPK-1.0FSNQS	1	2.8	2.2	8.5
Total								**140**	**110.2**	**385.5**

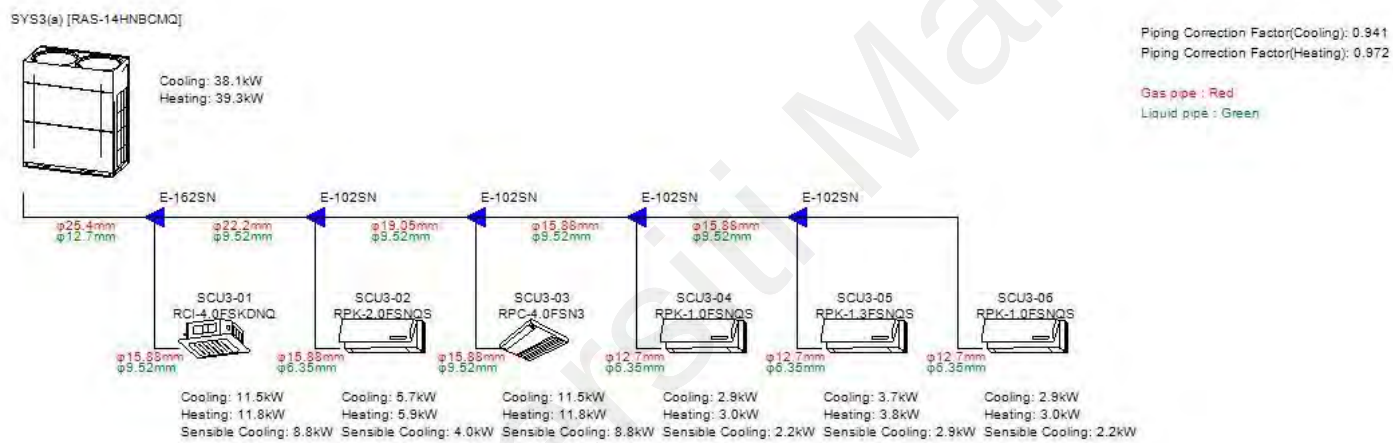


Figure 4.14: Schematic Drawing for System 3 (a)

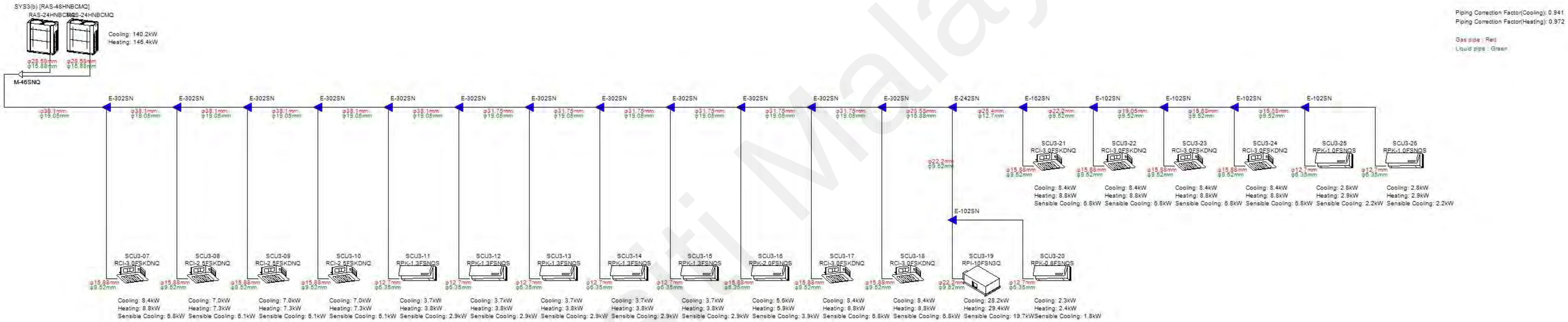


Figure 4.15: Schematic Drawing for System 3 (b)

Table 4.6: System design for System 4

Room Number/ Location		Cooling DB, ºC	Cooling WB, ºC	Relative Humidity	Room Number	Selected Model	Qty	Cooling Capacity, kW	Cooling Sensible Capacity, kW	Circulatory Airflow, m^3/min
SYS4[RAS-54HNBCMQ]										
SCU4-01	Preparation Area	27.0	19.6	50%	SCU4-01	RPK-1.3FSNQS	1	3.7	2.9	9.2
SCU4-02	Banquet Store / Preparation Area	27.0	19.6	50%	SCU4-02	RPC-2.0FSN3	1	5.7	4.2	15
SCU4-03	Operation Centre	27.0	19.6	50%	SCU4-03	RPK-1.3FSNQS	1	3.7	2.9	9.2
SCU4-04	Staff Eating Room	27.0	19.6	50%	SCU4-04	RCI-4.0FSKDNQ	1	11.4	8.8	31
SCU4-05	Staff Eating Room	27.0	19.6	50%	SCU4-05	RCI-4.0FSKDNQ	1	11.4	8.8	31
SCU4-06	Staff Eating Room	27.0	19.6	50%	SCU4-06	RCI-4.0FSKDNQ	1	11.4	8.8	31
SCU4-07	Staff Eating Room	27.0	19.6	50%	SCU4-07	RCI-4.0FSKDNQ	1	11.4	8.8	31
SCU4-08	Workstation 3	27.0	19.6	50%	SCU4-08	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU4-09	Workstation 2	27.0	19.6	50%	SCU4-09	RPK-1.3FSNQS	1	3.7	2.9	9.2
SCU4-10	Head of Psychology	27.0	19.6	50%	SCU4-10	RPK-1.3FSNQS	1	3.7	2.9	9.2
SCU4-11	Officer Workstation 1	27.0	19.6	50%	SCU4-11	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU4-12	Counselling Individual room	27.0	19.6	50%	SCU4-12	RPK-1.3FSNQS	1	3.7	2.9	9.2
SCU4-13	Counselling Grouping Room	27.0	19.6	50%	SCU4-13	RCI-2.5FSKDNQ	1	7.1	6.2	23
SCU4-14	Reception / Circulation Area	27.0	19.6	50%	SCU4-14	RPK-1.5FSNQS	1	4.3	3.1	10

SCU4-15	Male Surau	27.0	19.6	50%	SCU4-15	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU4-16	Female Surau	27.0	19.6	50%	SCU4-16	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU4-17	Activity / Presentation	27.0	19.6	50%	SCU4-17	RPK-1.5FSNQS	1	4.3	3.1	10
SCU4-18	Activity / Presentation	27.0	19.6	50%	SCU4-18	RPK-1.5FSNQS	1	4.3	3.1	10
SCU4-19	Head of Art Culture	27.0	19.6	50%	SCU4-19	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU4-20	Meeting Room	27.0	19.6	50%	SCU4-20	RCI-2.0FSKDNQ	1	5.7	5.1	17
SCU4-21	Meeting Room	27.0	19.6	50%	SCU4-21	RCI-2.0FSKDNQ	1	5.7	5.1	17
SCU4-22	Office Secretary	27.0	19.6	50%	SCU4-22	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU4-23	Head of Department	27.0	19.6	50%	SCU4-23	RCI-2.0FSKDNQ	1	5.7	5.1	17
SCU4-24	Head of Intake & Data Unit	27.0	19.6	50%	SCU4-24	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU4-25	Head of Sport	27.0	19.6	50%	SCU4-25	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU4-26	Head of Welfare & Discipline	27.0	19.6	50%	SCU4-26	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU4-27	Head of Co-Curriculum	27.0	19.6	50%	SCU4-27	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU4-28	Document Room	27.0	19.6	50%	SCU4-28	RPK-1.3FSNQS	1	3.7	2.9	9.2
SCU4-29	Discussion Room	27.0	19.6	50%	SCU4-29	RPK-1.8FSNQS	1	5.1	3.5	12
SCU4-30	Work Area & File Room (Intake & Data)	27.0	19.6	50%	SCU4-30	RPK-1.8FSNQS	1	5.1	3.5	12
SCU4-31	(Student Centre) Council Room	27.0	19.6	50%	SCU4-31	RPK-2.3FSNQS	1	6.6	4.6	13.7
Total								**156.4**	**121.2**	**420.9**

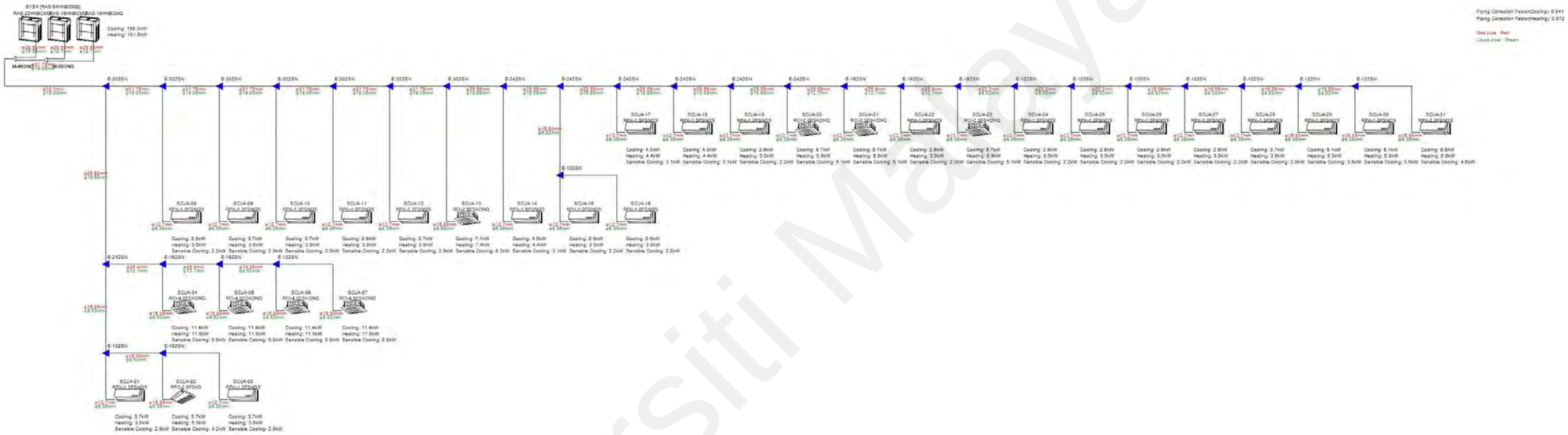


Figure 4.16: Schematic Drawing for System 4

Table 4.7: System Design for System 5

Room Number/ Location		Cooling DB, ºC	Cooling WB, ºC	Relative Humidity	Room Number	Selected Model	Qty	Cooling Capacity, kW	Cooling Sensible Capacity, kW	Circulatory Airflow, m^3/min
SYS5[RAS-12HNBCMQ]										
SCU5-01	Group Study 6	27.0	19.6	50%	SCU5-01	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU5-02	Group Study 5	27.0	19.6	50%	SCU5-02	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU5-03	Group Study 4	27.0	19.6	50%	SCU5-03	RPK-0.8FSNQS	1	2.3	1.8	8.5
SCU5-04	Group Study 3	27.0	19.6	50%	SCU5-04	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU5-05	Group Study 2	27.0	19.6	50%	SCU5-05	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU5-06	Group Study 1	27.0	19.6	50%	SCU5-06	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU5-07	Depository	27.0	19.6	50%	SCU5-07	RPK-1.3FSNQS	1	3.7	2.9	9.2
SCU5-08	Head of Unit	27.0	19.6	50%	SCU5-08	RPK-1.0FSNQS	1	2.9	2.2	8.5
SCU5-09	Media Storage Room	27.0	19.6	50%	SCU5-09	RPK-1.3FSNQS	1	3.7	2.9	9.2
SCU5-10	Multimedia Room	27.0	19.6	50%	SCU5-10	RPK-1.3FSNQS	1	3.7	2.9	9.2
SCU5-11	Multimedia Resource Room	27.0	19.6	50%	SCU5-11	RPK-1.5FSNQS	1	4.3	3.1	10
Total								**35.1**	**26.8**	**97.1**

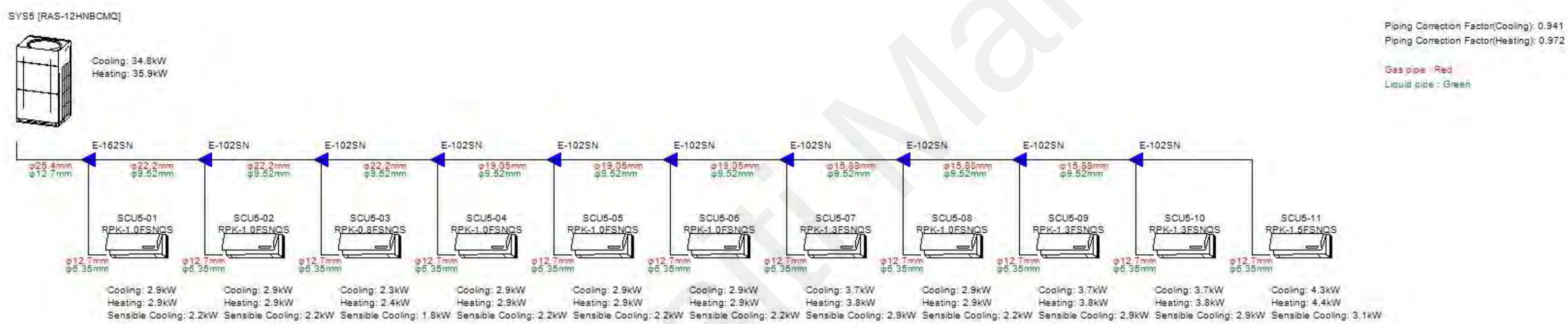


Figure 4.17: Schematic Drawing for System

Table 4.8: VRF Outdoor Unit Summary

Outdoor Name	Outdoor Unit Model (Power Supply)	Outdoor Working Condition		Actual Cooling Capacity, kW	Capacity Ratio %
		Cooling DB, ºC	RH		
SYS1	RAS-28HNBCMQ (380~415V/3Ph/50Hz)	35.0	51%	76.3	97.2%
SYS2	RAS-34HNBCMQ (380~415V/3Ph/50Hz)	35.0	51%	100	105.26%
SYS3(a)	RAS-14HNBCMQ (380~415V/3Ph/50Hz)	35.0	51%	38.2	95.50%
SYS3(b)	RAS-48HNBCMQ (380~415V/3Ph/50Hz)	35.0	51%	140	102.94%
SYS4	RAS-54HNBCMQ (380~415V/3Ph/50Hz)	35.0	51%	156.4	103.23%
SYS5	RAS-12HNBCMQ (380~415V/3Ph/50Hz)	35.0	51%	35.1	104.78%
Total				**545.5**	

In conclusion, the total number of system design for Polytechnics Hulu Terengganu was six system with the total cooling capacity need to achieve was 545.5 kW which almost equivalent to 150 tons. The total outdoor condensing unit offered for the whole system was 534.5, which makes the total capacity ratio of the whole variable refrigerant flow system to be 102.06%. In common nature of VRF system, the capacity ration is allowed to be diverse from 50% to 130%. Hence, with 102.06%, this VRF system will still work under suitable and optimum condition.

4.3.2 VRF System Performance Analysis

This part will discussed on the part load performance of each system. The coefficient of performance of the system was further analyses using the part load data.

Table 4.9: Performance Data for VRF System 1

Part Load (%)	Actual Total Cooling Capacity, kW	Part Load Total Capacity. kW	Part Load Power Input, kW	Capacity Offered HP	Actual COP	Part Load COP
System 1						
100	76.2	78.5	22.37	28	3.41	3.51
90	76.2	76.5	22.71	30	3.36	3.37
80	76.2	76	22.95	34	3.32	3.31
75	76.2	75.8	21.78	36	3.50	3.48
70	76.2	74.6	21.78	38	3.50	3.43
60	76.2	77.7	23.91	46	3.19	3.25
50	76.2	75.8	22.14	54	3.44	3.42

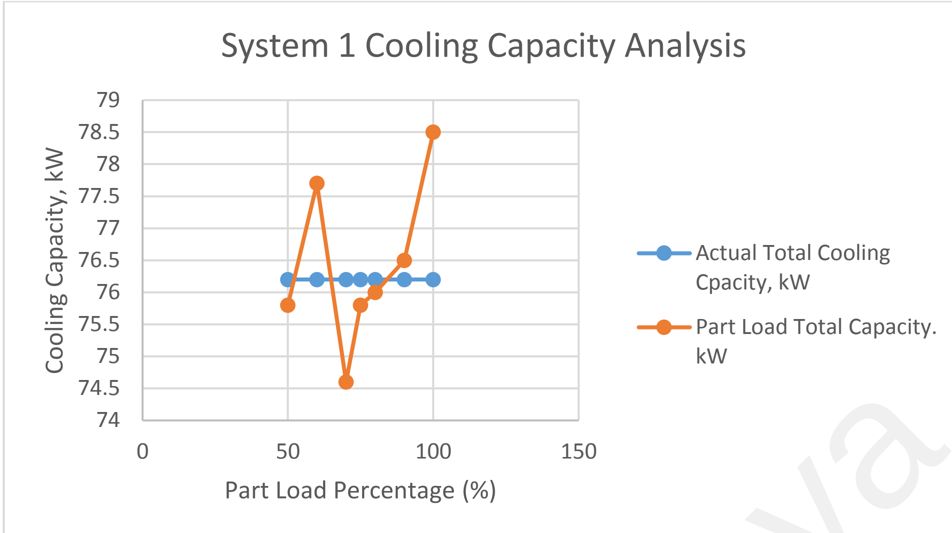


Figure 4.18: Cooling Capacity Analysis for System 1

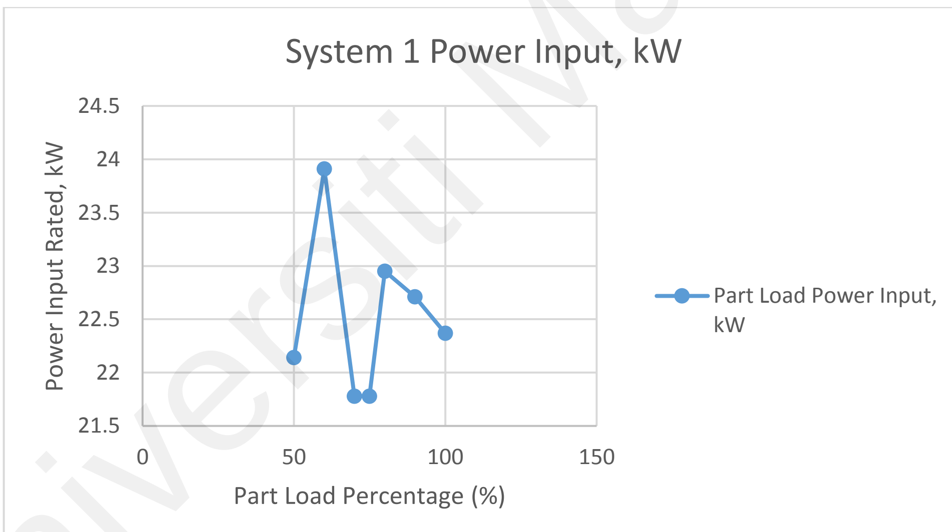


Figure 4.19: Power Input Data for System 1

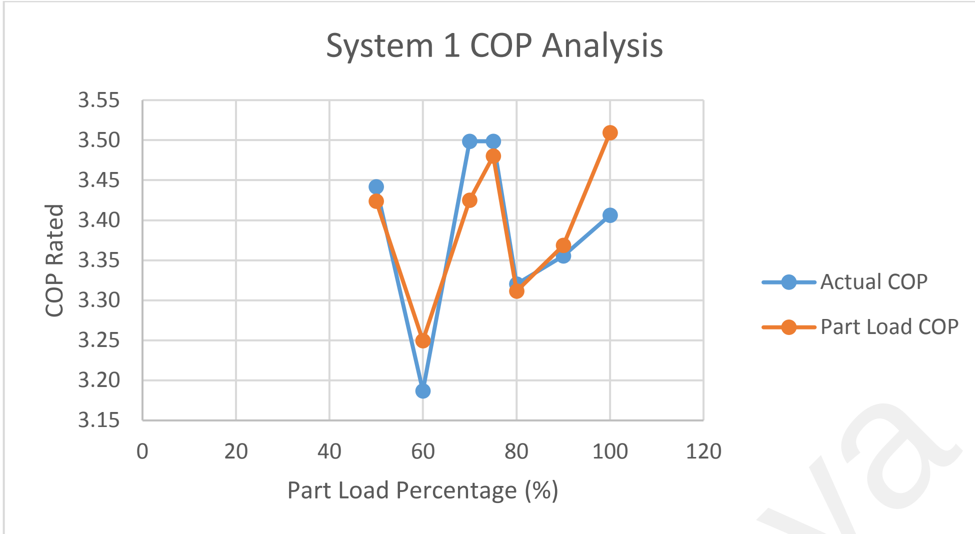


Figure 4.20: COP Analysis for System 1

Table 4.9 show the summary of part load total capacity, the power input generated by each part load and also the coefficient of performance for both actual system and part load data. From figure 4.18 we can see that the optimum total part load cooling capacity to be consider was at 50%, 75%,80% and 90% from the total capacity ratio generated for the system. This is because the rated capacity nearly matched with the actual required capacity which is 76.2 kW. Figure 4.19 show that the lowest power input was at the part load percentage of 75% and 70%. This low rated power input will influenced the coefficient of performance of a system. Hence, figure 4.20 realize the COP reading of the system. In actual COP where we take the actual total cooling capacity of system 1 which is 76.2 kW to divide with the part load power input, we can see that at part load percentage of 70% and 75%, the COP rated the highest among other by 3.50. On the other hand, when we compared each offered outdoor cooling capacity (or part load total capacity) with the power input generated by the system, part load percentage at 100% show the highest reading of COP which is 3.51. Thus, in conclusion of performance for system 1, 3 suitable outdoor capacity can be offered in a way to achieve high performance which is at 100% part load of 28 HP, 75% part load at 36HP and 70% part load at 38 HP of outdoor total cooling capacity.

Table 4.10: Performance Data for VRF System 2

Part Load (%)	Actual Total Cooling Capacity, kW	Part Load Total Capacity. kW	Part Load Power Input, kW	Capacity Offered HP	Actual COP	Part Load COP
System 2						
100	100	95	29.8	34	3.36	3.19
90	100	101.7	32.32	40	3.09	3.15
80	100	99.2	30.68	44	3.26	3.23
75	100	97.1	29.99	46	3.33	3.24
70	100	98	28.97	50	3.45	3.38
60	100	97.8	30.12	58	3.32	3.25
50	100	98.8	30.55	70	3.27	3.23

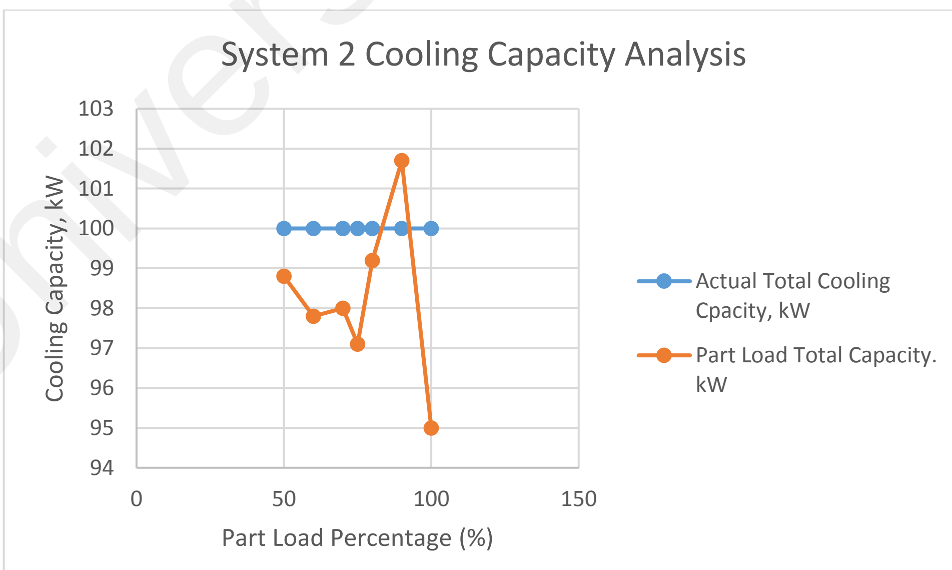


Figure 4.21: Cooling Capacity Analysis for System 2

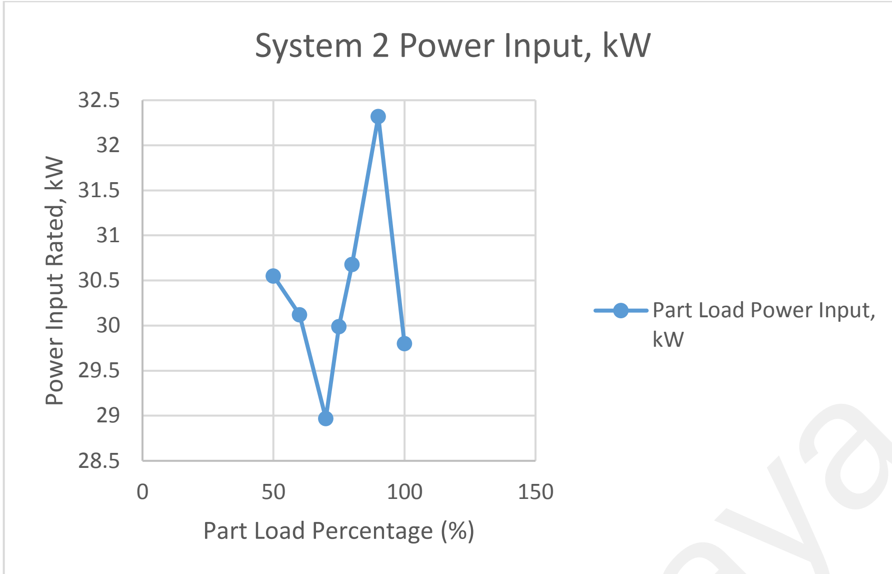


Figure 4.22: Power Input Data for System 2

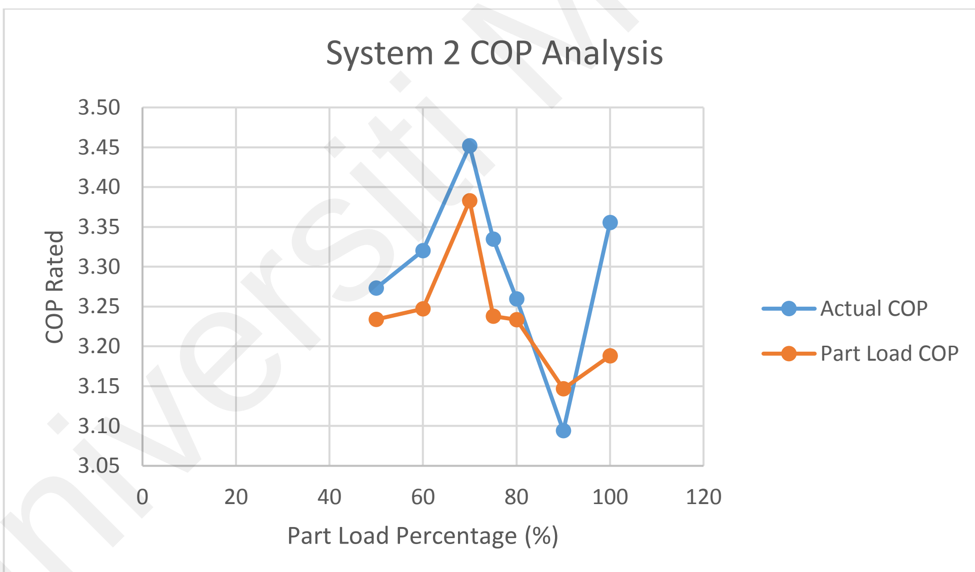


Figure 4.23: COP Analysis for System 2

With an actual total cooling capacity of 100 kW, system 2 has come out with a summary of performance that can be refer from table 4.10. From Figure 4.21, we can see that the optimum part load percentage of total cooling capacity offered was at 80% and 90% which is 99.2 kW and 101.7 kW. Even though this two part load condition give an optimum capacity which is near to the actual capacity of 100 kW, the power input generated from both percentage is quite high. Hence both of this unit in the end offer quite low coefficient of performance to be compared with other percentage. From figure 4.22, we can see that at 70%, 75% and 100% of the part load capacity offer low input power generation of the system. Hence, we can see that this three part load percentage rated among the highest COP generated for system 2. In conclusion, for system 2, the suitable capacity that can be offer in a way to achieve a high efficient in performance was at 100% part load at 34 HP, 75% part load at 46 HP and 70% part load at 50 HP of total outdoor cooling capacity.

Table 4.11: Performance Data for VRF system 3 (a)

Part Load (%)	Actual Total Cooling Capacity, kW	Part Load Total Capacity. kW	Part Load Power Input, kW	Capacity Offered HP	Actual COP	Part Load COP
System 3 (a)						
100	38.2	40	12	14	3.18	3.33
90	38.2	40.5	12.27	16	3.11	3.30
80	38.2	40	12.09	18	3.16	3.31
75	38.2	37.5	11.23	18	3.40	3.34
70	38.2	39.2	10.92	20	3.50	3.59
60	38.2	36.9	10.77	22	3.55	3.43
50	38.2	39.3	10.29	28	3.71	3.82

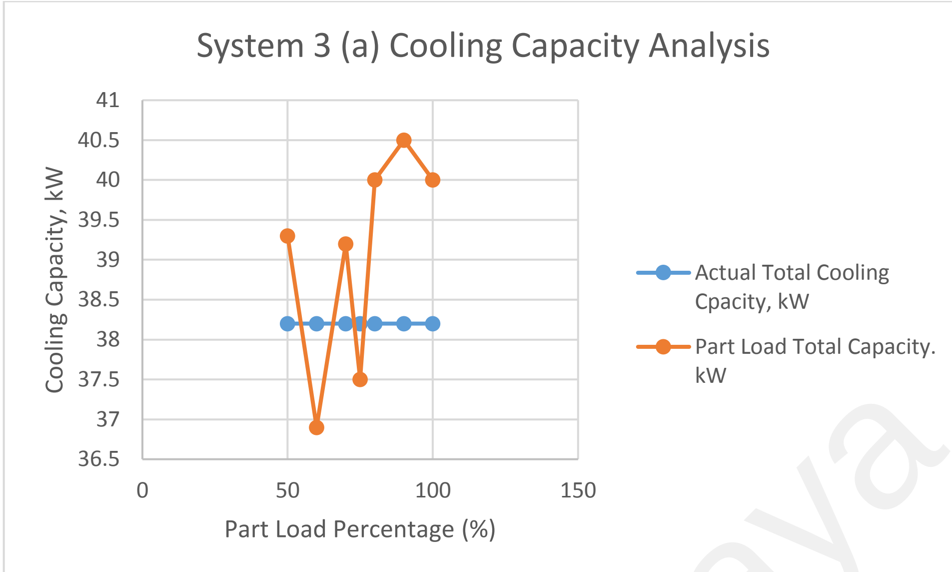


Figure 4.24: Cooling Capacity Analysis for System 3 (a)

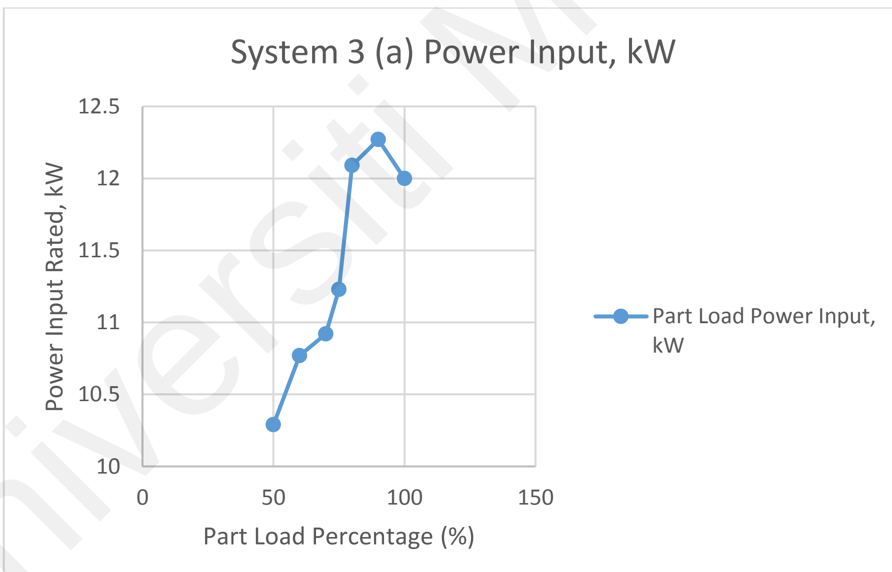


Figure 4.25: Power Input Data for System 3 (a)

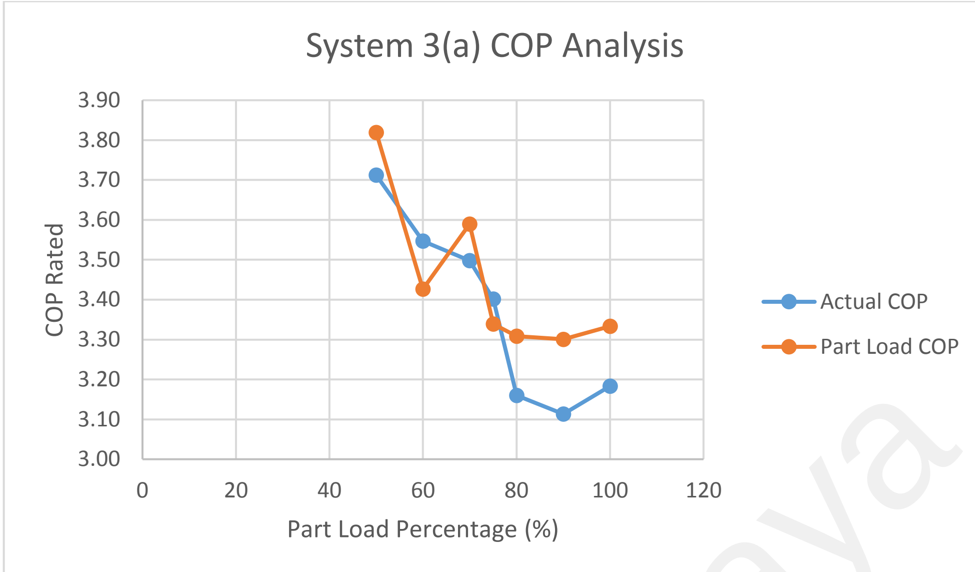


Figure 4.26: COP Analysis for System 3 (a)

System 3 (a) was designed with 38.2 kW of total cooling capacity for its system. Table 4.11 show the summary of the performance data for system 3 (a). From Figure 4.24, we can see that the optimum par load percentage that could take into our consideration was at 70% and 75%. This is because the capacity offered was near to the actual cooling capacity required by the system which is 38.2 kW. On the other hand, when we refer from figure 4.25, the power input rated at 50% part load percentage give the minimum amount compared to others with 10.29 kW. Due to this rated input power results, the rated COP at 50% marks the highest compared to others with 3.71. Hence, the suitable capacity that can be offer in a result to have a high efficient system was at 50% part load condition of 28 HP outdoor cooling capacity.

Table 4.12: Performance Data for VRF System 3 (b)

Part Load (%)	Actual Total Cooling Capacity, kW	Part Load Total Capacity. kW	Part Load Power Input, kW	Capacity Offered HP	Actual COP	Part Load COP
System 3 (b)						
100	140	136	46.1	48	3.04	2.95
90	140	136.4	41.45	54	3.38	3.29
80	140	139.6	43.16	62	3.24	3.23
75	140	139.5	44.19	66	3.17	3.16
70	140	138.3	42.9	70	3.26	3.22
60	140	137.7	38.12	82	3.67	3.61
50	140	136	43.33	96	3.23	3.14

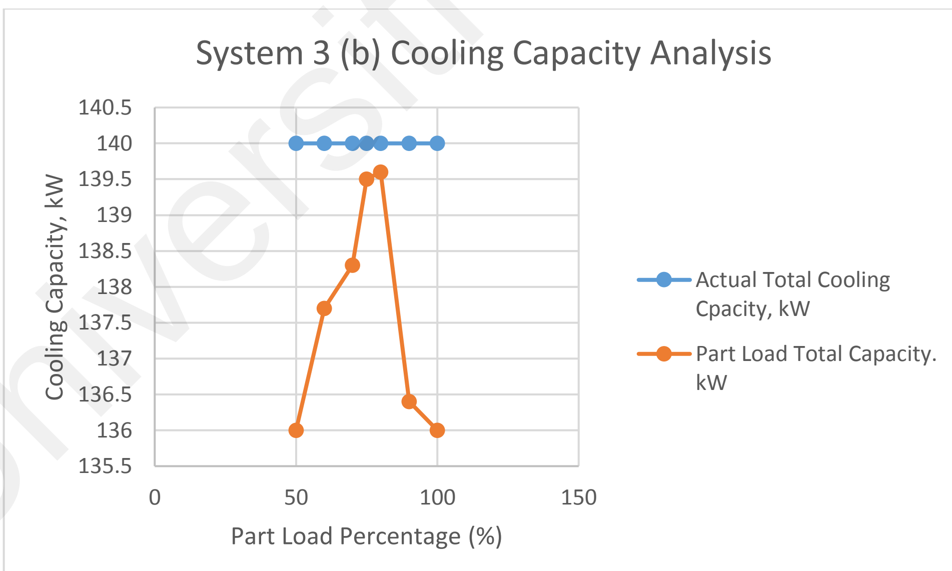


Figure 4.27: Cooling Capacity Analysis for System 3 (b)

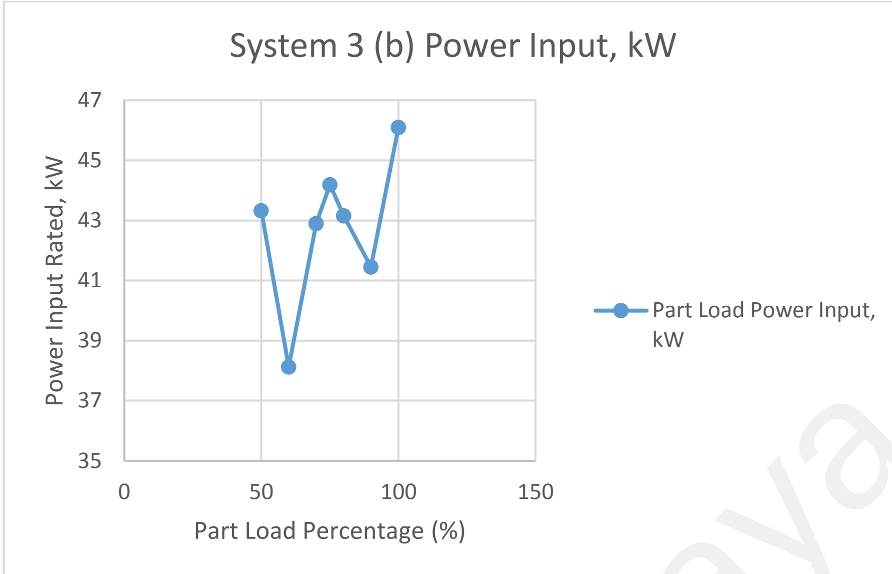


Figure 4.28: Power Input Data for System 3 (b)

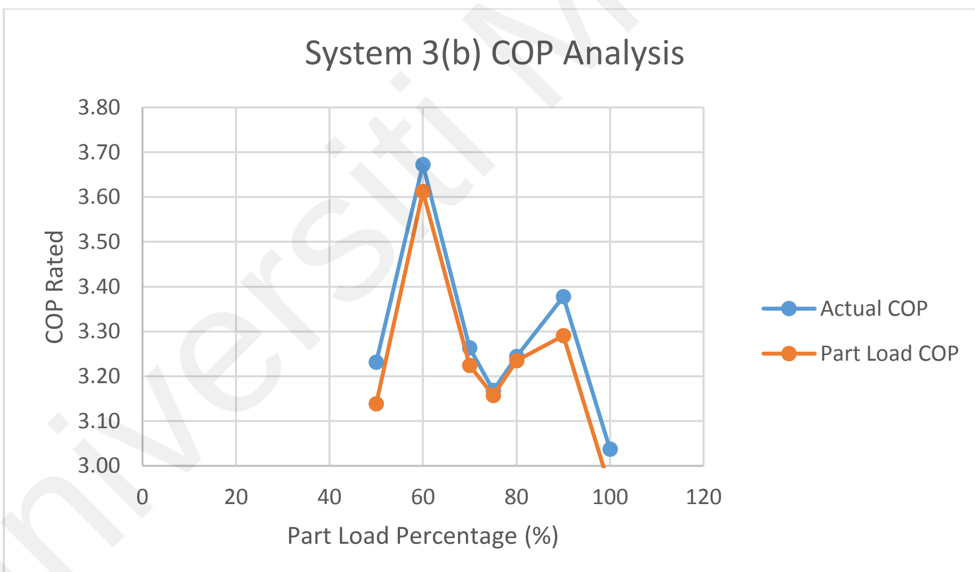


Figure 4.29: COP Analysis for System 3 (b)

Table 4.12 show the summary of performance data generated while designed this system. System 3(b) was designed with 140 kW actual cooling capacity that need to achieve by the outdoor condensing unit. Thus, from figure 4.27 in the cooling capacity analysis for different part load condition or percentage, it shows that the optimum condition was at 80% and 75% of capacity ratio (part load percentage). By referring to figure 4.28, the power input rated at 60% of part load percentage give the lowest reading among others. This input power is a main influencer of the coefficient of performance for a system. The lower the input power generated by the system, the higher the COP of the system. This is the reason, from figure 4.29, we can see that the highest COP reading was at 60% of indoors and outdoors total cooling capacity or part load data. Hence the most suitable condition in fulfilling the high performance of system 3(b) was at 60% part load condition with 82 HP of outdoor total cooling capacity.

Table 4.13: Performance Data for VRF System 4

Part Load (%)	Actual Total Cooling Capacity, kW	Part Load Total Capacity. kW	Part Load Power Input, kW	Capacity Offered HP	Actual COP	Part Load COP
System 4						
100	156.4	151.5	47.1	54	3.32	3.22
90	156.4	152.1	47.48	60	3.29	3.20
80	156.4	153.6	38.43	68	4.07	4.00
75	156.4	156.8	47.87	72	3.27	3.28
70	156.4	156.8	44.35	80	3.53	3.54
60	156.4	156	48.99	90	3.19	3.18
50	156.4					

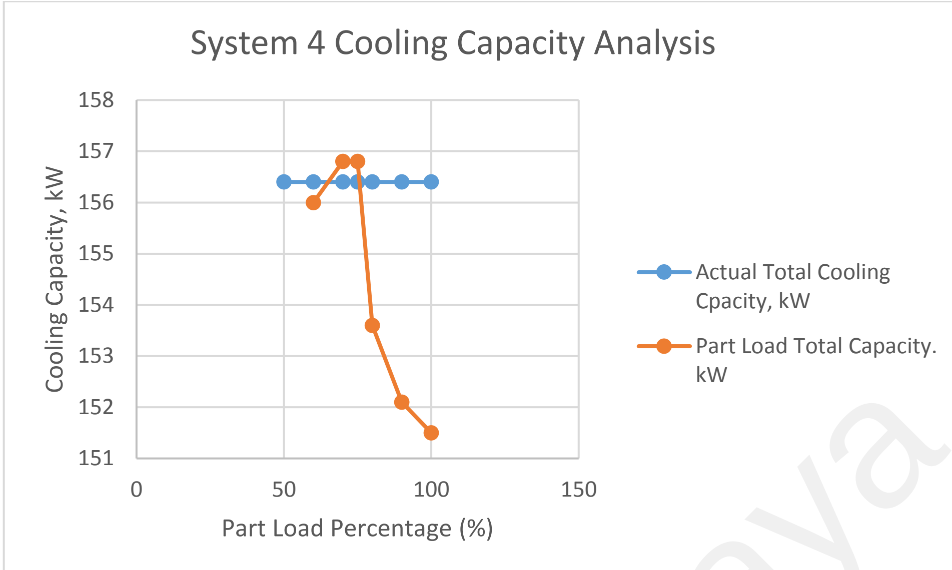


Figure 4.30: Cooling Capacity Analysis for System 4

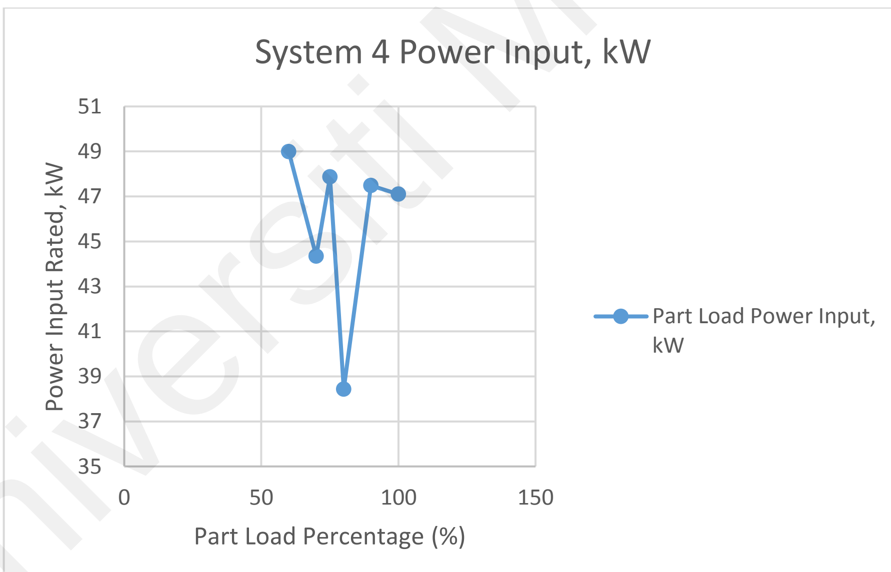


Figure 4.31: Power Input Data for System 4

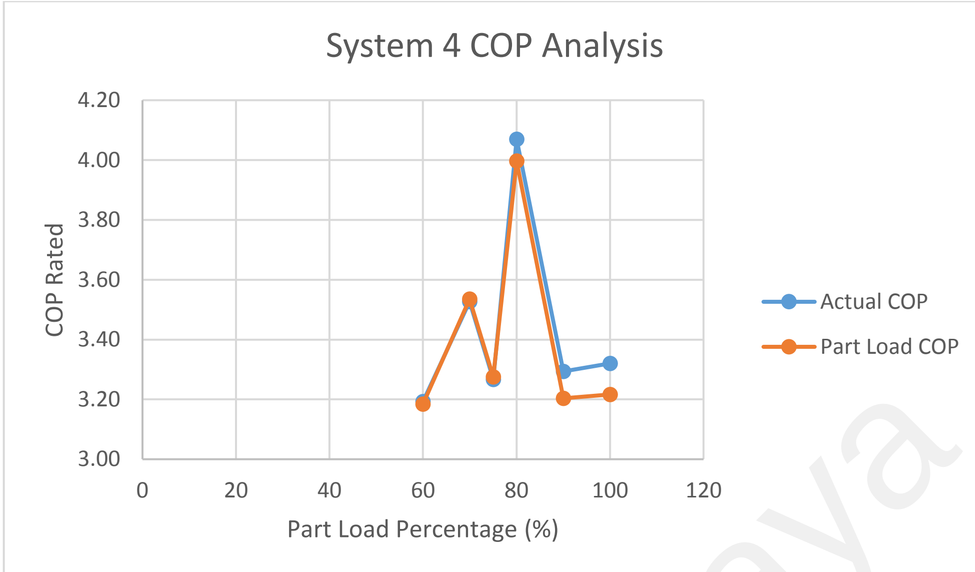


Figure 4.32: COP Analysis for System 4

In addition, the summary of performance data for system 4 at different part load condition can be refer from table 4.13. Due to limitation of capacity ranged offered by Hitachi VRF outdoor condensing unit line up, the data for 50% part load cannot be obtained. From figure 4.30 we can see the most optimum part load condition was at 60%, 70% and 75% of outdoor cooling capacity. The three range of this part load percentage shows the rated cooling capacity near or almost to the actual capacity demand for system 4 which is 156.4 kW. On top of that, from figure 4.31, the power input rated at 80% of part load percentage shows the lowest power input generated among others with 38.43 kW. As been discussed previously, the input power generated by a system will give a great impact with the performance of the system itself. Due to this condition, we can see that the most suitable capacity ration between actual indoor capacity and total outdoor cooling capacity was at 80% part load percentage. This give a reading of COP of the system at 80% part load percentage is 4.07.

Table 4.14: Performance Data for VRF System 5

Part Load (%)	Actual Total Cooling Capacity, kW	Part Load Total Capacity. kW	Part Load Power Input, kW	Capacity Offered HP	Actual COP	Part Load COP
System 5						
100	35.1	33.5	8.27	12	4.24	4.05
90	35.1	36	10.44	14	3.36	3.45
80	35.1	32	9.12	14	3.85	3.51
75	35.1	33.8	9.94	16	3.53	3.40
70	35.1	35	10.36	18	3.39	3.38
60	35.1	33.6	9.24	20	3.80	3.64
50	35.1	34	10.83	24	3.24	3.14

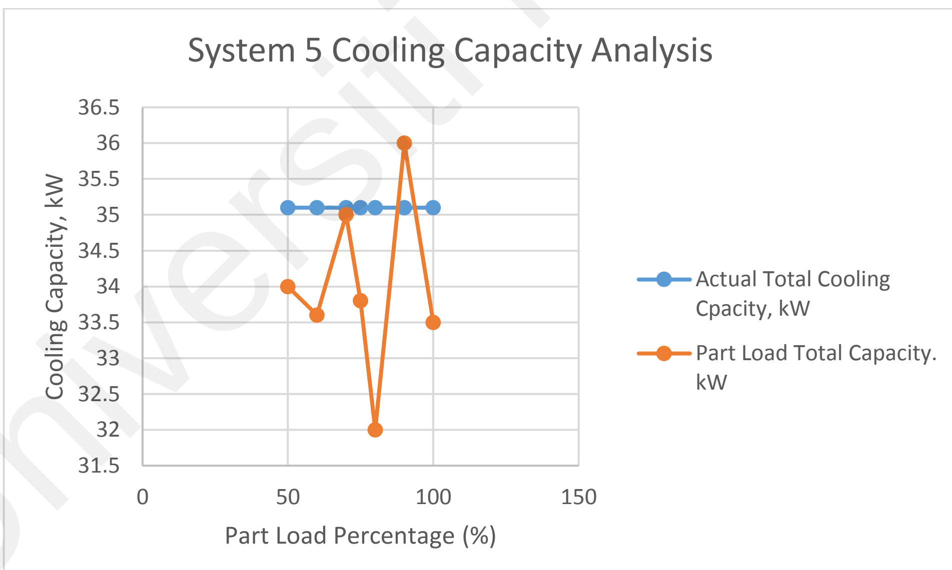


Figure 4.33: Cooling Capacity Analysis for System 5

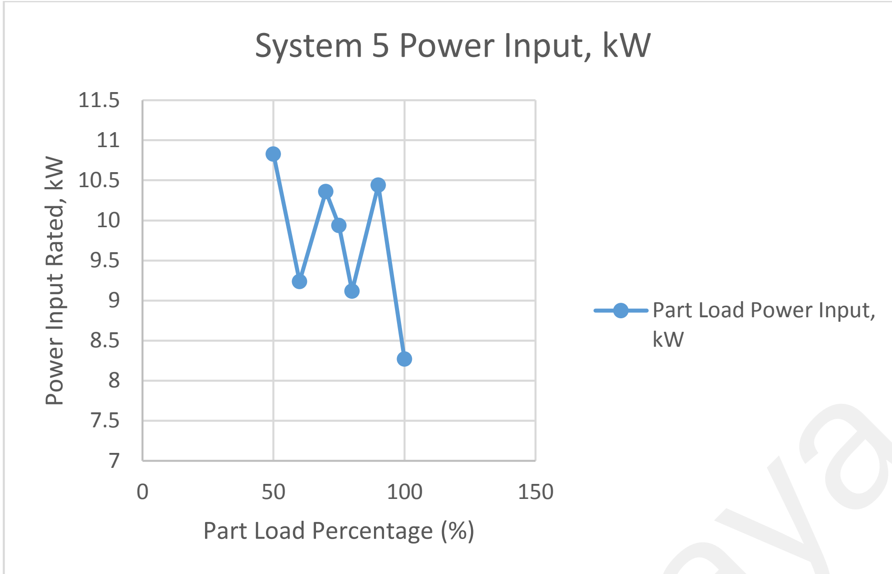


Figure 4.34: Power Input Data for System 5

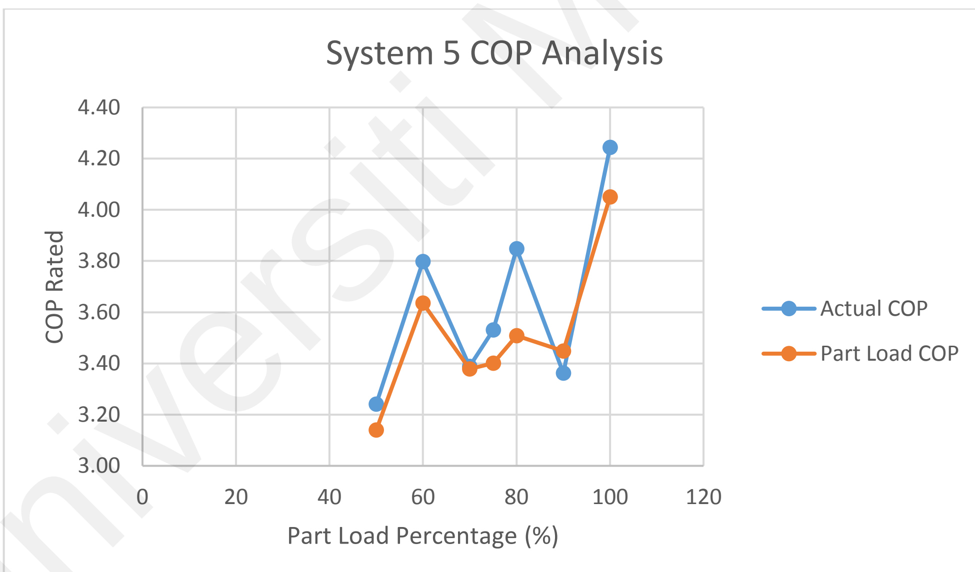


Figure 4.35: COP Analysis for System 5

Lastly, table 4.15 shows the performance data summary at different part load for system 5. Throughout this data in table 4.14, the figure 4.33 of cooling capacity analysis was generated. From this figure we can see that the optimum capacity ratio or part load condition was at 70 %. This part load generate 35 kW cooling capacity which is much closed with the demand of total cooling capacity required by actual system which is 35.1 kW. When we analyze the power input distribution among the part load percentage offered for system 5, the input power generated at 100% capacity ratio between indoor and outdoor condition show the lowest reading at 8.27 kW. Hence, the most suitable part load for this system was at 100% at 12 HP outdoor cooling capacity that will give the highest performance reading of .24.

In conclusion, each of the variable refrigerant flow system was been analyzed under different percentage of capacity ration or part load percentage. Hitachi VRF system is a full inverter system, thus the allowable efficient range for this system to be diverse are from 50% to 100%. However, the diversity range can be increase up to 130% in maximum if the cost of equipment was the upmost consideration of this analysis. The part load data can be refer from appendix 1

4.4 Chilled Water System

4.4.1 Equipment Design

Chilled water cooling system is another system used of cooling and heating of an area. After several survey has been done among 3 main key player of chiller industry in Malaysia which is Daikin, York and Hitachi, the most significance water chiller type used for a project with 150 tons and below are the screw chiller type. Hence, three data of three different composition of screw chiller was designed for Polytechnics Hulu Terengganu. The list as below.

1. Chiller 1 : Daikin Air Cooled Single Screw Chiller
2. Chiller 2 : York YVWE (Variable Water Cooled Screw Chiller High Efficiency)
3. Chiller 3 : York YGWE (Water Cooled Screw Chiller High Efficiency)

Several other equipment also involved for a complete chiller system such as cooling tower for water type of condensing unit chiller system, exhaust fan for air cooled type chiller system, pumps and air handling unit ducting, and diffusers.

4.4.1.1 Design of Air Handling Unit

To match with the variable refrigerant flow system, sin number of AHU was designed for chilled water system. Table 4.15 shows the summary of AHU that was designed for this system. In designing the AHU, several factor was take into a consideration. The factors are as below.

1. The water velocity pass through the coil should not exceed 2.5 m/s
2. The maximum number of rows per system should be 6, this to avoid water trap between the rows that will cause freezing and effect the performance of the system
3. The maximum no of fins per inch should be 14, and the reason is the same as 6 rows required per system.

Table 4.15: AHU Performance Summary

	Total Cooling Capacity, kW	ESP	Water Velocity, m/s	Airflow, CFM	Rows	FPI
System 1	80.5	570	1.19	11600	6	8
System 2	94.13	570	0.67	13800	6	12
System 3 (a)	40.3	250	1.77	5800	4	14
System 3 (b)	137.76	570	1.16	19700	6	8
System 4	151.87	570	1.25	22000	6	8
System 5	33.42	250	0.88	5000	5	12

The AHU performance report and drawing can be refer to appendix 2

4.4.1.2 Design of chiller unit

As been informed previously there are 3 type of chiller that was designed for this project.

1. Chiller 1: Daikin Air Cooled Single Screw Chiller
2. Chiller 2 : York YVWE (Variable Water Cooled Screw Chiller High Efficiency)
3. Chiller 3 : York YGWE (Water Cooled Screw Chiller High Efficiency)

Table 4.16 below show the pre-requisite of chiller specification that need to comply by each chiller system. The chiller full performance report and drawing can be refer from appendix 3.

Table 4.16: Pre-requisite to Design Chiller

Evaporator Leaving Water Temperature	44 ºF
Condenser Leaving Water Temperature	85 ºF ~ 90 ºF
Evaporator Fouling Factor	0.000100
Condenser Fouling Factor	0.000250
Fluid Type	Water
Total Tonnage requires for Chiller	150 Tons
Refrigerant used	R134A

4.4.2 Chiller System Performance Analysis

Table 4.17: Performance Data for Chiller 1

Chiller 1	**Daikin Air Cooled Single Screw Chiller**				
	Actual Total Cooling Capacity, kW	Part Load Total Capacity. kW	Part Load Power Input, kW	Actual COP Chiller 1	Part Load COP Chiller 1
100	537.98	503.28	88.39	6.09	5.69
90	537.98	453	79.28	6.79	5.71
80	537.98	402.7	71.12	7.56	5.66
70	537.98	352.4	63.5	8.47	5.55
60	537.98	302	56.21	9.57	5.37
50	537.98	251.68	48.83	11.02	5.15

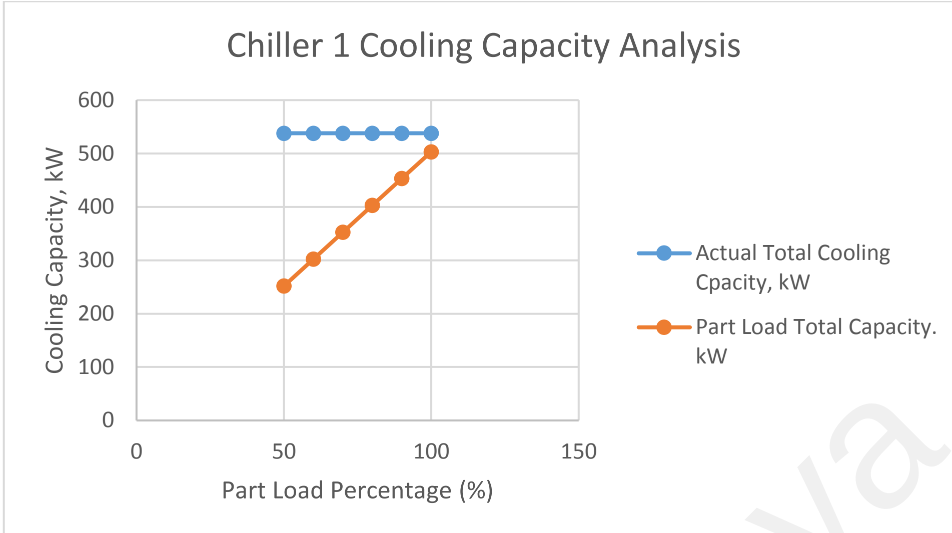


Figure 4.36: Cooling Capacity Analysis for Chiller 1

Chiller 1 is an air type condensing chiller which used air source as a cooling medium for its condensing unit. The part load percentage was rated from 100 % to 50 %, can be seen from table 4.17 was the same as the analysis that has been done for variable refrigerant flow system. Chiller 1 system design was obtain from Daikin brands. The cooling capacity analysis for this system that can be refer from figure 4.36 show the optimum condition design for part load performance was at 100 % capacity ratio. This project required to supply 150 tons of cooling capacity for the whole system. Thus, at 100% part load condition, the capacity rated is around 503.28 kW which is nearest to the total cooling capacity required by the system which is 537.98. In this condition, at 100% part load percentage give among the highest part load coefficient of performance compared to other. If the capacity ration between indoor equipment (air handling unit) and the chiller itself around 50%, it will reached e very efficient system with 11.02 COP rated for its actual system. This is due to the fact that lesser power input was generated by the chiller system.

Table 4.18: Performance Data for Chiller 2

Chiller 2	**York YVWE (Variable Water Cooled Screw Chiller High Efficiency)**				
	Actual Total Cooling Capacity, kW	Part Load Total Capacity. kW	Part Load Power Input, kW	Actual COP Chiller 1	Part Load COP Chiller 1
100	537.98	527.55	93.4	5.76	5.65
90	537.98	474.8	80.87	6.65	5.87
80	537.98	422.04	68.55	7.85	6.16
70	537.98	369.28	58.66	9.17	6.30
60	537.98	316.53	48.99	10.98	6.46
50	537.98	263.77	41.34	13.01	6.38

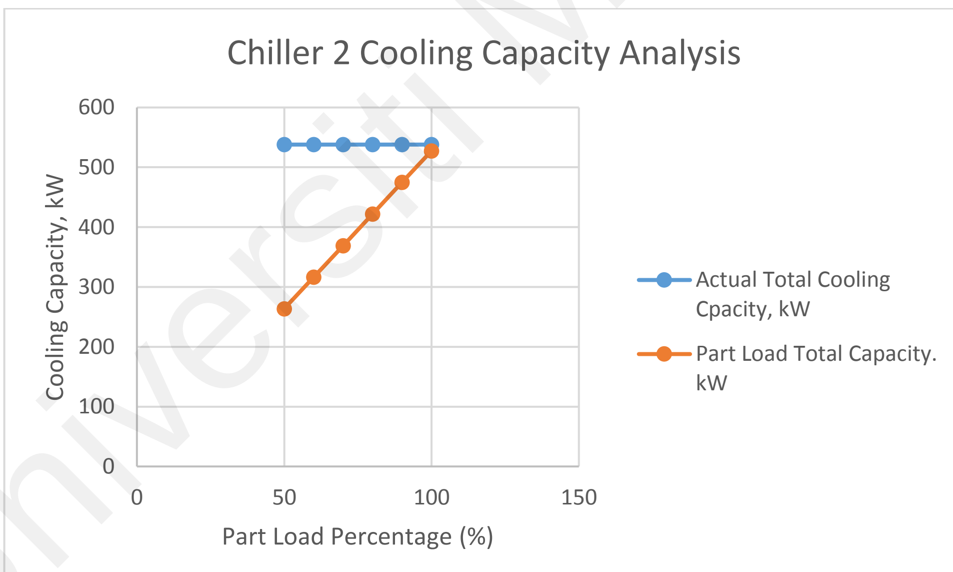


Figure 4.37: Cooling Capacity Analysis for Chiller 2

Chiller 2 is a water type condensing chiller system or known as water cooled chiller system. The system operate by using water source as its cooling medium for its condensing unit. This is the reason, most of this type of system will be equipped with a cooling tower. Chiller 2 is a water cooled screw chiller with high efficiency performance supply by York Sales & Services Malaysia Sdn. Bhd. Table 4.19 show the summary of performance data for chiller 2. This table shows the part load variance from 50 % to 100% of the total capacity ration between chiller and indoor equipment. Figure 4.37 show that the optimum condition for this chiller to operate was at 100 % part load condition. This is because the total cooling capacity generated by the system at 100 % very near to the capacity demand of an actual system which is 150 ton ~ 537.98 kW. However, as the part load percentage or capacity ratio between indoor cooling capacity and total outdoor cooling capacity generated, the input power of the chiller also decrease. This is a significant reason on the increment of actual coefficient of performance as the part load percentage decrease.

Table 4.19: Performance Data for Chiller 3

Chiller 3	**York YGWE (Water Cooled Screw Chiller High Efficiency)**				
	Actual Total Cooling Capacity, kW	Part Load Total Capacity. kW	Part Load Power Input, kW	Actual COP Chiller 3	Part Load COP Chiller 3
100	537.98	527.55	95.79	5.62	5.51
90	537.98	474.8	85.46	6.30	5.56
80	537.98	422.04	75.25	7.15	5.61
70	537.98	369.28	68.12	7.90	5.42
60	537.98	316.53	61.17	8.79	5.17
50	537.98	263.77	54.25	9.92	4.86

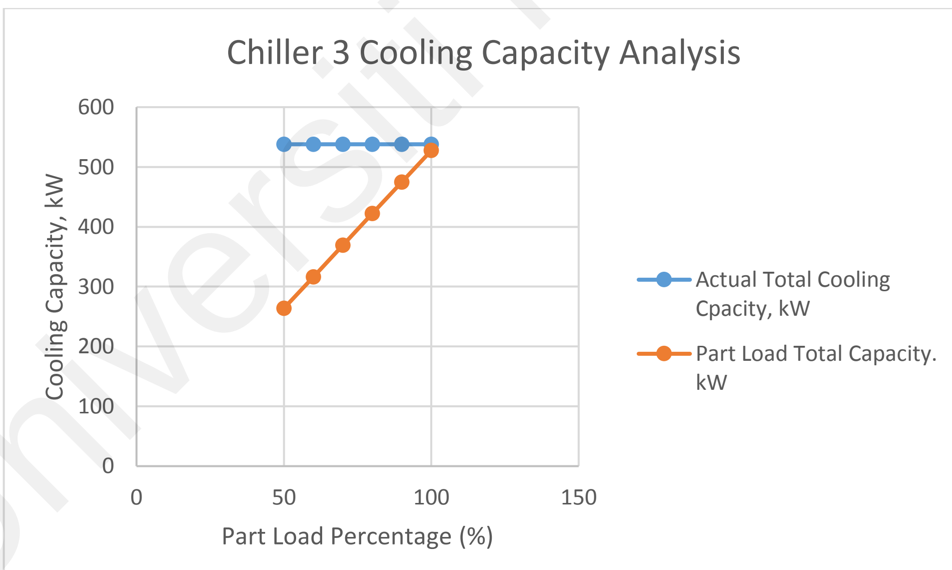


Figure 4.38: Cooling Capacity Analysis for Chiller 3

Next is chiller 3. This chiller is also a water cooled chiller which used water source as its condensing cooling medium. This chiller was also supply by York Malaysia Sales & Services Malaysia Sdn. Bhd. From table 4.19, we can see the summary of performance data for chiller 3. Basically chiller 3 have the same composition with chiller 2, but the system was design without variable speed drive. Which this give a meaning that, chiller 3 will operate at a fixed rated input power without any variables. Figure 4.38 show that the optimum part load percentage to be matched with the required cooling capacity for the whole system which is 537.98 was at 100% with 527.55. This case of performance run very similar as chiller 2. As the part load percentage decrease, the coefficient of performance increase. This due to the reduction of power input generated as the system part load or capacity ratio decrease.

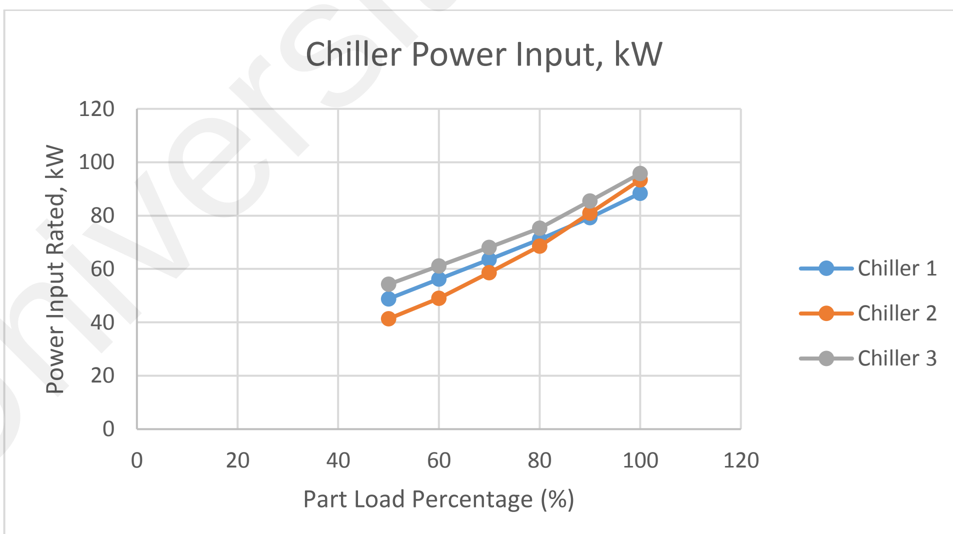


Figure 4.39: Power Input Data Comparison between Chiller

After all the discussion on performance of each chiller, the comparison of input power data between those three chillers can be seen from figure 4.39. From this figure it show that chiller 2, which is a variable water cooled screw chiller marked to have the lowest input power reading compared to other. This is due to the fact that chiller 2 was equipped with a variable speed drive motor in its system that enhance in the variability of the input power throughout the part load or capacity ratio of the whole system. Air-cooled screw chiller which is the chiller 1 marked to generate high input power of the system. Other than the fact that this chiller 1 operate at a constant speed without any variable speed drive implemented in this system, it also because by using air as a cooling medium is not a very efficient way of cooling method. Unlike water, air really depend on ambient and surrounding temperature and also the capacity and type of fan used in condenser.

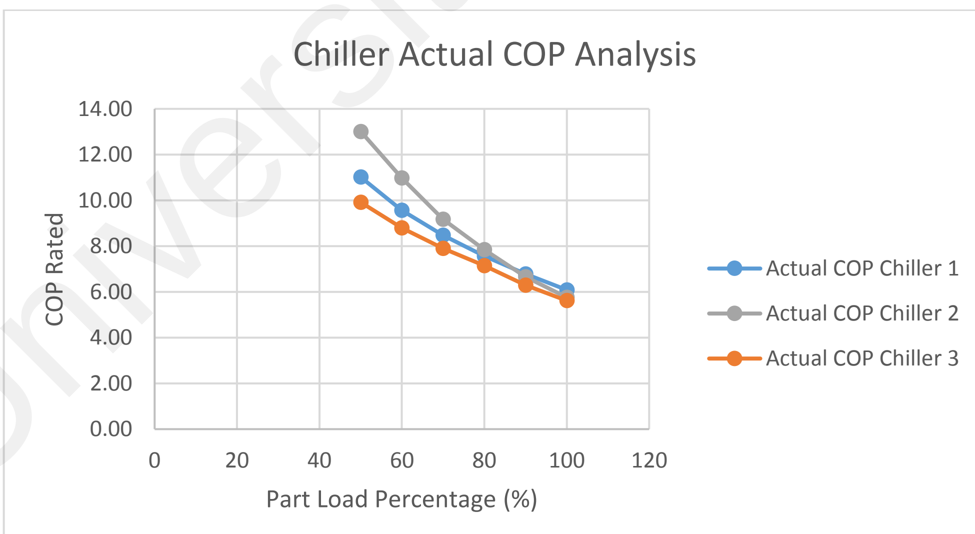


Figure 4.40: Actual COP Comparison between Chillers

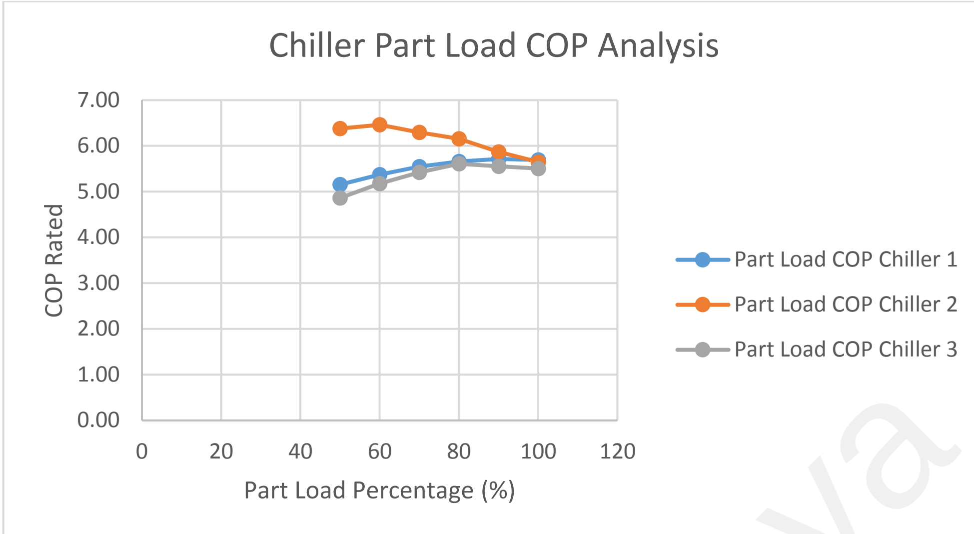


Figure 4.41: Part Load COP Comparison between Chillers

On top of that, the comparison between the coefficient of performance of actual load and part load generated by this three type of chiller can be refer from figure 4.40 and figure 4.41. From both figure, we can see that chiller 2 which is York variable water cooled screw chiller was rated as the most efficient chiller compared to an air cooled screw chiller and standard water cooled screw chiller. Despite the reason that chiller 2 was equipped with a variable speed drive motor, the medium of cooling that was used in this system which is water source have very much give a great impact for this system. It a behavior of water itself that can easily expands and also been controlled was additional reason why water source condensing system remarks the great coefficient of performance for a system.

In conclusion, three different type of chiller was designed to fulfill the cooling capacity demand of the building which is 537.98 kW which equivalent to 150 tons. Throughout the analysis of the chillers system performance, chiller 2 which is a variable water screw chiller system has been marked as the best among others. This chiller offer higher coefficient of performance compare to other two chiller

4.5 Cost and Economic Evaluation on System

4.5.1 Operational Cost Analysis

Model	Capacity, kW	Qty	Input Power, kW	Operating hours	Day, kWh	Month, kWh	Bill, RM/ month	Bill, RM/year	Bill, RM/ 10 year
RAS-28HNBCMQ (380~415V/3Ph/50Hz)	78.5	1	22.37	12	268.44	5,368.80	2,692.50	32,310.01	323,100.10
RAS-34HNBCMQ (380~415V/3Ph/50Hz)	95	1	29.8	12	357.60	7,152.00	3,612.63	43,351.58	433,515.84
RAS-14HNBCMQ (380~415V/3Ph/50Hz)	40	1	12	12	144.00	2,880.00	1,408.28	16,899.36	168,993.60
RAS-48HNBCMQ (380~415V/3Ph/50Hz)	136	1	46.1	12	553.20	11,064.00	5,631.22	67,574.69	675,746.88
RAS-54HNBCMQ (380~415V/3Ph/50Hz)	151.5	1	47.1	12	565.20	11,304.00	5,755.06	69,060.77	690,607.68
RAS-12HNBCMQ (380~415V/3Ph/50Hz)	33.5	1	8.27	12	99.24	1,984.80	946.36	11,356.28	113,562.82
Total							**20,046.06**	**240,552.69**	**2,405,526.91**
Chiller 1	503.28	1	88.39	12	1,060.68	21,213.60	**10,868.42**	**130,421.01**	**1,304,210.11**
Chiller 2	527.55	1	93.4	12	1,120.80	22,416.00	**11,488.86**	**137,866.27**	**1,378,662.72**
Chiller 3	527.55	1	95.79	13	1,245.27	24,905.40	**12,773.39**	**153,280.64**	**1,532,806.37**

Table 4.20: Operational Cost Summary

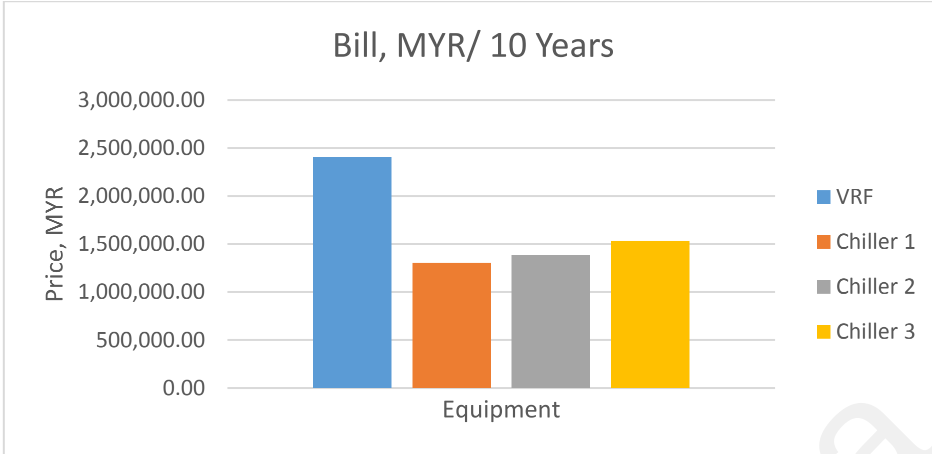


Figure 4.42: Bill Generated for 10 Years of Consumption

Table 4.20 show the summary of operational cost for each system that was designed for this project. The operating hours of the system was fixed for 12 hours every day. The costing of the operation cost for each system was refer from the fixed charge rated by *Tenaga Nasional Berhad (TNB).* Variable refrigerant flow remarks to have the highest input power generated throughout the system compared to chiller system with 170.64 kW. This reading was taken at the 100 % part load data. Although this system is fully inverter system and there will be a variable in cooling capacity supplied by the outdoor or condensing unit, the rated input power that take into consideration still at its 100% load. As result from this, the total bill monthly for variable refrigerant system was RM 20,056.06 and in a year, the total bill that will generate by variable refrigerant flow system was RM 240,552.69. IN 10 years the total operational cost for this VRF system will be at RM 2,505,526.91.

On the other hand, let us see the summary for chilled water system. For chiller 1, the operational cost in a month of this system is RM 10,868.42 and in a year of RM 130,421.01. The generated cost for 10 years of consumption for chiller 1 will be RM 1,304,210.11. Next from chiller 2 operational cost data that can be refer from table 4.21, the cost in a month generate by this system is RM 11,488.86 and by a year, the total operation cost will be RM 137,866.27. In 10 years, the total operational cost generated by chiller 2 will be RM 1,378,662.72. On top f that, foe chiller 3, the monthly operation cost will be RM 12,773.39 and in a year it will generate an amount of RM 153,280.64. In 10 years, the total operational cost for chiller 3 will be at RM 1,532,806.37.

In conclusion, we can see that the operational cost generated by the chiller system much lower to be compared to variable refrigerant system. Figure 4.42 show the comparison of operational cost generated by the four system offered in this project.

4.5.2 Equipment Cost

Table 4.21: Equipment First Cost

Model	Indoor Equipment, MYR	Outdoor Equipment, MYR	Accessories, MYR	Total, MYR
VRF	364,960.00	425,500.00	49,670.00	840,130.00
Chiller 1	184,018.00	270,000.00	564,352.50	1,018,370.50
Chiller 2	184,018.00	383,751.00	564,352.50	1,132,121.50
Chiller 3	184,018.00	392,640.00	564,352.50	1,141,010.50

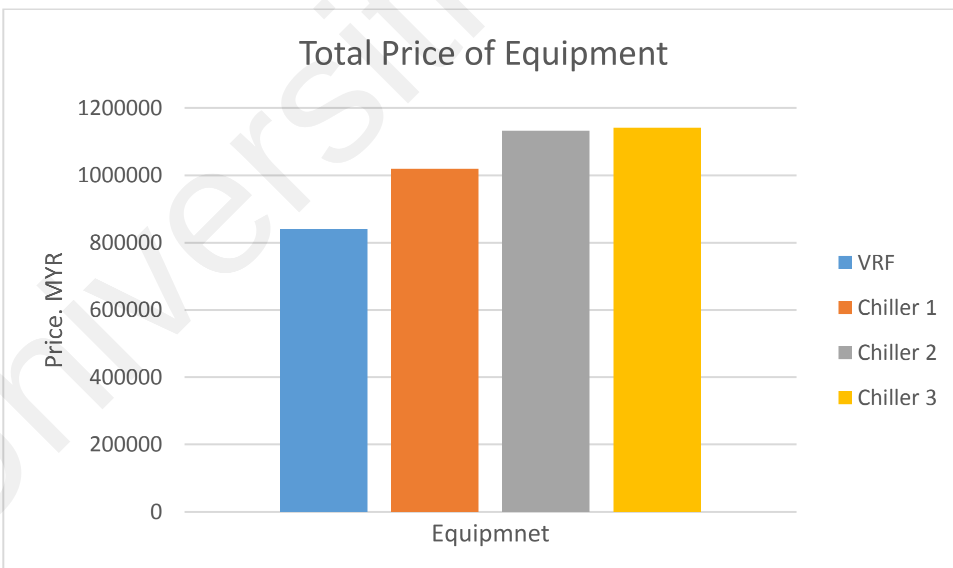


Figure 4.43: Total Price of Equipment

Table 4.21 show the equipment cost for four system that has been design for Polytechnic Hulu Terengganu. The total price or first cost for variable refrigerant flow system is RM 840,130.00. The accessories of VRF system consist of the y-joint connection on indoor unit, outdoor connection unit and also the control system. For chiller 1, the total cost of equipment is RM 1,018,370.50. Chiller 2 by York give a cost of RM 1,132,121.50 and chiller 3 is RM 1,141,010.50. The accessories cost for chiller is a lump sum cost per tonnage that include cooling tower, water pump and piping.

In conclusion, we can see that VRF offer the lowest starting cost compared to chiller system, and this is the reason, currently Polytechnic Hulu Terengganu installing variable refrigerant system. Figure 4.43 show the comparison of total price of the equipment.

4.5.3 Payback Period

Table 4.22: Payback Period Summary

	Annual Savings, MYR	First Cost. MYR	Payback Period	Years
VRF vs Chiller 1	110,131.68	1,018,370.50	9.25	9 Years 3 Month
VRF vs Chiller 2	102,686.42	1,132,121.50	11	11 Years
VRF vs Chiller 3	87,272.05	1,141,010.50	13	13 Years

As currently Polytechnic Hulu Terengganu installing the VRF system, this study was further extend to compare the existing system with the chillers system. The annual savings was obtained from formula below.

Annual Savings (MYR) = Annual Energy Cost of Standard Unit (VRF operational cost) – Annual Energy Cost of High Efficiency Unit (chiller system)

Payback Period = First Cost (MYR) / Annual Savings (MYR)

CHAPTER 5

CONCLUSION & RECOMMENDATIONS

This study mainly highlight and objective to study the performance of a variable refrigerant air conditioning system and also chilled water system. Hence Polytechnics Hulu Terengganu has been selected as the study area for this project.

This project was start by analyzing and estimate the capacity required for the entire building. As a result from this estimation, Polytechnic Hulu Terengganu need to cover with 150 tons of cooling capacity. The variable refrigerant flow (VRF) was deign using Johnson Control VRF software. With a deep analyze on the floor plan of the building, I have proposed 6 individual system in total to fulfill the cooling capacity demand by the building. The performance of VRF system was evaluated throughout the part load data offered in achieving the required cooling capacity per system. The first cost of VRF equipment marks as the lowest compared to chiller system with RM 840,130.00. This is the reason currently, Polytechnics Hulu Terengganu installing the VRF system on its building.

On top of that, this project was further been study by designing a chiller system with a capacity of 150 tons. There were three type of chillers has been design which is first an air cooled screw chillers, second is a variable water type screw chiller and also a standard water type screw chiller. As the tonnage requires just 150 tons, the only suitable chiller for this range of capacity is screw type. Among these three chiller, the variable water cooled screw chiller have marks as the most efficient chillers compared to other. This is because, this chiller was equipped with a variable speed drive that help in regulated the variability of input power generated by this system. on the other hand, the reason of this chiller give a high efficiency performance is because of the medium used by the system in its condenser cooling mechanism which is by water. The nature of water that easily expand and controlled have given such a great plus point for this type of chillers. The operational cost of each chiller and first cost has been well explained in the previous chapter.

Since currently, the system that been installed in Polytechnics Hulu Terengganu is a VRF system, a further analysis on payback period if we implement the chillers system was done. The duration of payback period for chiller system should be around nine year to thirteen year depend on type of chiller used. In conclusion, the chiller system have given the most reliability efficiency in performance to be compared with the VRF system. However, VRF system offer lower starting cost compares to the chiller system In addition, the operational cost of each system was well discussed in previous chapter.

Further recommendation that can be done in improving this research first by changing the site of study to the building with bigger capacity, advisable to be more than 300 tons. This is because there is more type of chiller such as centrifugal can be analyze and compared too. Second, to study the implementation of other primary source such as solar with the system to analyze the savings on the operational cost. Third, this research also can be focused more on the variability of refrigerant used such as R410A, R32, R407C and R134A. This is because each refrigerant has its own properties that eventually will affect the performance of the system itself. Last, to really further study on the part load condition of each systems.

REFERENCES

ABB, (2011) . "Guide to variable speed drives", technical guide n°4

Carrier, (2013). Variable Refrigerant Flow (VRF) System- Flexible solution for comfort. Carrier Corporation Syracuse, New York. 2013

Carrier. "Operation and application of variable frequency drive (VFD) technology" New York, Carrier Corporation Syracuse (2005) available on line.

Chua KJ, Chou SK, Yang WM, Yan J. (2013). Achieving better energy-efficient air conditioning – a review of technologies and strategies. Appl Energy 2013; 104:87–104.

Cooling Post, Global AC growth expected this year, 2017 [accessed 17.08.31]. http://www.coolingpost.com/world- news/global- ac- growth- expectedthis

Department of energy and climate change, energy consumption in the UK. London: DECC; 2013.

Department of Statistic, 2019. Electricity consumption in Malaysia. Global Economic, Data, Indicators, Charts & Forecast, (CEIC)

Doo Yong Park, Gyeong Yun, Kang Soo Kim, (2017). Experimental evaluation and simulation of a variable refrigerant- flow (VRF) air-conditioning system with outdoor air processing unit, Energy and Buildings 146 (2017) 122–140

E. Zahi , A.E. Abu O Lu , A.H. Kutlar , O.U. Da C , (2017). A comparative thermodynamic and economic analysis and assessment of a conventional HVAC and a VRF system in a social and cultural center building, Energy Build. 140 (2017) 196–209.

Fernando H, Venezia J, Rigdon C, Verma P, 2008 Capturing King Coal: Deploying Carbon Capture and Storage Systems in the U.S. at Scale (World Resources Institute, Washington, DC)

Goetzler W., 2017. Variable refrigerant flow systems. ASHRAE Journal. April, 2007

Hydraulic Institute, Euro Pump and U.S. Department of Energy, (2004). "Variable speed pumping - A Guide to Successful Applications"

IEA, 2009 World Energy Outlook (OECD/ International Energy Agency, Paris)

IPCC. (2001). the scientific basis. WMO/UNEP Intergovernmental Panel on Climate Change, Third Assessment Report, VI. Climate change

John Buie, 2011. Evolution of Water Baths and Chillers. Lab Manager.com. https://www.labmanager.com/lab-product/2011/10/evolution-of-water-baths-and-chillers#.XS8rgSIzapp

Karunakaran R, Iniyan S, Goic R. (2010). Energy efficient fuzzy based combined variable refrigerant volume and variable air volume air conditioning system for buildings. Appl Energy 2010; 87:1158–75. http://dx.doi.org/10.1016/j.apenergy.2009.08.013.

McNeil MA, Letschert VE. 2007. Future air-conditioning energy consumption in developing countries and what can be done about it: the potential of efficiency in the residential sector. In: Proceedings of ECEE2007 summer study workshop.

Mechanical and Electrical Information, 2017. Report of central air conditioning market in China (2017).

Paul Evans, 2008. Chiller Types and Application Guide. The Engineering Mindset.com https://theengineeringmindset.com/chiller-types-and-application-guide/

Shaharuddin Ahmad. 1997. Urbanization and human comfort in Kuala Lumpur-Petaling Jaya, Malaysia. Ilmu Alam 23: 171-189.

Siti Zakiah Mohammed. 2004. The influence of urban heat towards pedestrian comfort and the potential use of plants and water as heat ameliorator in Kuala Lumpur (MSc thesis). Universiti Pertanian Malaysia.

Suruhanjaya Tenaga. 2010. Maklumat prestasi dan statistik 2010, Industri Pembekalan Elektrik di Malaysia, Suruhanjaya Tenaga.

Suruhanjaya Tenaga. 2018. "Malaysia Energy Statistic Handbook 2017", Suruhanjaya Tenaga.

T.N. Aynur, Variable refrigerant flow systems: a review, 42 (2010). Energy Build. 1106–1112.

Technomedia Solution Private Ltd 2019. Training manual for technical staff of Technomedia Solution Private Ltd. Technical Literature, India

Trane, 2009. Chiller System Design and Control. Trane

X. Lin, H. Lee, Y. Hwang, R. Radermacher, (2015). A review of recent development in variable refrigerant flow systems, Sci. Technol. Built. Environ. 21 (2015) 917–933.

Yun GY, Lee JH, Kim HJ, (2016). Development and application of the load responsive control of the evaporating temperature in a VRF system for cooling energy savings. Energy Build 2016; 116:638–45.